# INVESTIGATIONS ON PERFORMANCE OF A DIESEL ENGINE OPERATED WITH RAW CARDANOL AND KEROSENE BLENDS

Thesis

Submitted in partial fulfillment of the requirements for the degree of

DOCTOR OF PHILOSOPHY

by

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# D E C L A R A T I O N

I hereby *declare* that the Research Thesis entitled **"INVESTIGATIONS ON PERFORMANCE OF A DIESEL ENGINE OPERATED WITH RAW CARDANOL AND KEROSENE BLENDS"** which is being submitted to the **National Institute of Technology Karnataka, Surathkal** in partial fulfilment of the requirements for the award of the Degree of **Doctor of Philosophy** in **Department of Mining Engineering** is a *bonafide report of the research work carried out by me.* The material contained in this Research Thesis has not been submitted to any University or Institution for the award of any degree.

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**Date: 01/02/2019**

# C E R T I F I C A T E

This is to *certify* that the Research Thesis entitled **"INVESTIGATIONS ON PERFORMANCE OF A DIESEL ENGINE OPERATED WITH RAW CARDANOL AND KEROSENE BLENDS"**submitted by **Mr. Ravindra** (**Register Number: 155029MN15F05**) as the record of the research work carried out by him, is *accepted as the Research Thesis submission* in partial fulfilment of the requirements for the award of degree of **Doctor of Philosophy.**

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# ABSTRACT

Worldwide diesel engines are the main source of power for heavy duty equipments in mines and other applications. Since the world crude oil reserves are depleting very fast, there is a need for alternative source. The biodiesel originated from animal fats or vegetable oils are the easier alternatives for diesel fuel, which can be utilised without much engine alterations. However, increased cost of the biodiesel due to the esterification process involved in the production of biodiesel is a limiting factor for vast usage of this alternative.

In this research work, raw Cardanol extracted from cashew nut shell is tested as a diesel engine fuel without esterification. To reduce the viscosity of Cardanol, it was blended with kerosene. Experiments were carried out in a 3.5 kW four stroke single cylinder diesel engine using different blends of Cardanol and kerosene, such as BK10 (10% kerosene and 90% Cardanol), BK20% (20% kerosene and 80% Cardanol), BK30 (30% kerosene and 70% Cardanol), BK40% (40% kerosene and 60% Cardanol) as a fuel. Performance parameters such as brake thermal efficiency, brake specific fuel consumption, exhaust gas temperature and the exhaust emissions of unburned hydrocarbon, carbon monoxide, oxides of nitrogen and smoke were measured and compared with diesel fuel. Effect of the engine operating parameters like, compression ratio, injection pressure and injection timing on the engine performance are also investigated. Using the taguchi method, the experimental results were optimised and BK30 blend proved as the most favourable blend for optimum engine performance with minimum emissions, under the following operating conditions: compression ratio - 18:1; injection pressure - 220 bar; injection timing - 24.5°BTDC; load - 12 kg. A fuel cost reduction by about 22% could be observed upon using BK30 biofuel blend as a replacement to diesel fuel in mine machineries. Invention of this novel biofuel blend increases the effective utilisation of Cardanol as a biofuel.

Key words: Cardanol; kerosene; performance; compression ratio; brake thermal efficiency.

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## LIST OF SYMBOLS AND ABBREVATIONS

ASTM- American society for testing and materials

BK10 - 10% kerosene and 90% Cardanol blend by volume

BK20 - 20% kerosene and 80% Cardanol blend by volume

BK30 - 30% kerosene and 70% Cardanol blend by volume

BK40 - 40% kerosene and 60% Cardanol blend by volume

BSFC- Brake specific fuel consumption

BTDC - Before top dead centre

BTE- Brake thermal efficiency

CB15- Coconut oil 15% and 85% diesel blend by volume

CI- Compression ignition

CNSL-Cashew nut shell liquid

CO- Carbon monoxide

$CO_2$- Carbon dioxide

CPO- Crude palm oil

CR- Compression ratio

DEE – Di ethyl ether

DMC-Dimethyl carbonate

EGR- Exhaust gas recirculation

EGT- exhaust gas temperature

FFA-Free fatty acid

HC- Unburned hydrocarbon

HEMM -Heavy earth moving machinery

HSU-Hartridge smoke unit

kg- Kilograms

kW- Kilowatts

ml-Milli litre

MT- Metric ton

NaOH- Sodium hydroxide

NOx- Oxides of nitrogen

Pa-Pascal

PB15- Palm oil biodiesel 15% and 85% diesel blend by volume

PM- Particulate matter

ppm – Parts per million

rpm- Revolutions per minute

SFC – Specific fuel consumption

SVO- Straight vegetable oil

# CHAPTER 1

# INTRODUCTION

## 1.1 INTRODUCTION TO DIESEL ENGINE

Heat engines are serving the society since two centuries. The device that transforms heat energy of a fuel into mechanical energy are named as heat engines. The concept of heat engine was utilised by many scientists in developing an engine. During 1765- 1769 James Watt established a steam engine, where the heat energy was utilised for steam production and the engine was operated by steam. In 1816, Robert Sterling a Scottish scientist developed a hot air operated engine called as sterling engine. After that a French engineer Nicolas Leonard Sadi Carnot, in the year 1824 suggested the thermodynamic theory of heat engine and developed the concept of Carnot engine. The idea of burning compressed coal air mixture was initially suggested by Jean Joseph Etienne Lenoir (1822-1900). Later Nicolaus A. Otto during 1867, improved the J.J.E.Lenoir's concept and used the pressure rise due to burning of fuel air mixture for the movement of piston. Influenced by N.A.Otto's inventions Rudolf diesel, a German scientist during the year 1892, originated diesel engine. He adapted the theory of combustion of a fuel in a closed chamber with air as surrounding media. The atmospheric air drawn in during suction is compressed and the fuel is sprayed into compressed air. The heat produced during compression of air initiates combustion process of the atomised fuel. Therefore, the diesel engines are generally designated as compression ignition engines.

Diesel engines play a significant role as power source for the heavy duty vehicles and equipments such as heavy trucks, buses, marine propulsion and earth moving equipments. This is because of its high torque, rigidness, durability, fuel economy and low carbon monoxide emissions.

## 1.2 DIESEL ENGINES IN MINES

In mines, various Heavy Earth Moving Machinery (HEMM) are commonly used to excavate the coal, minerals and overburden rocks and to transport them from one place to another. The dumpers having lesser capacity are driven by means of diesel engine with the help of torque converters whereas higher capacity dumpers are driven by electric drives. These trucks consume heavy amount of diesel and lubricants that increases the pollution. A dumper that has a capacity of 55 MT can consume as high as 110 litres of fuel approximately per hour when it is operational. An excavator of 405 HP power consumes 45 litres of diesel per hour of its operation.

## 1.3 USAGE OF FOSSIL FUELS

The problems of climate change have been catching world-wide attention over last few decades. The pollution problems due to the increased usage of non-renewable sources of energy has been a key contributor for the same. The demand for the use of non-renewable sources of energy, in particularly fossil fuels has been increasing every day, even though there is a shortfall (Figure 1.1). One of the primary issues with regard to the usage of fossil fuels is the problems associated with emissions.

The non-renewable sources of energy, due to the increased demand will be exhaustible in the near future. As per the analysis carried out by Shahriar and Topal (2009) the oil, coal and gas are expected to be exhausted within 35, 107 and 37 years, respectively. More than 70% of the crude petroleum oil is imported by India and an increased trend is observed every year (Natesan 2013). The world oil production rate has been reached to its peak and started declining (Figure 1.2). It will be exhausted for the next generation.

## 1.4 PROBLEMS WITH FOSSIL FUELS

Some of the environmental problems due to continued use of fossil fuels are listed as follows.

- Greenhouse effect- A greenhouse gas is an atmospheric gas that absorbs and emits radiation within the thermal infrared range.

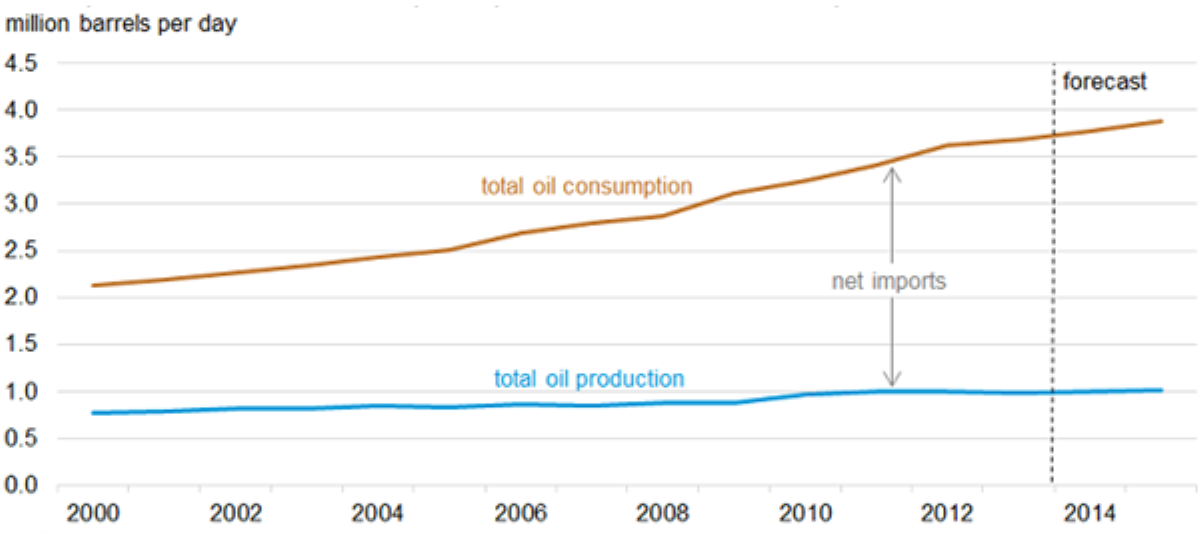


Fig. 1.1 Indian oil production and consumption (Source: eia)

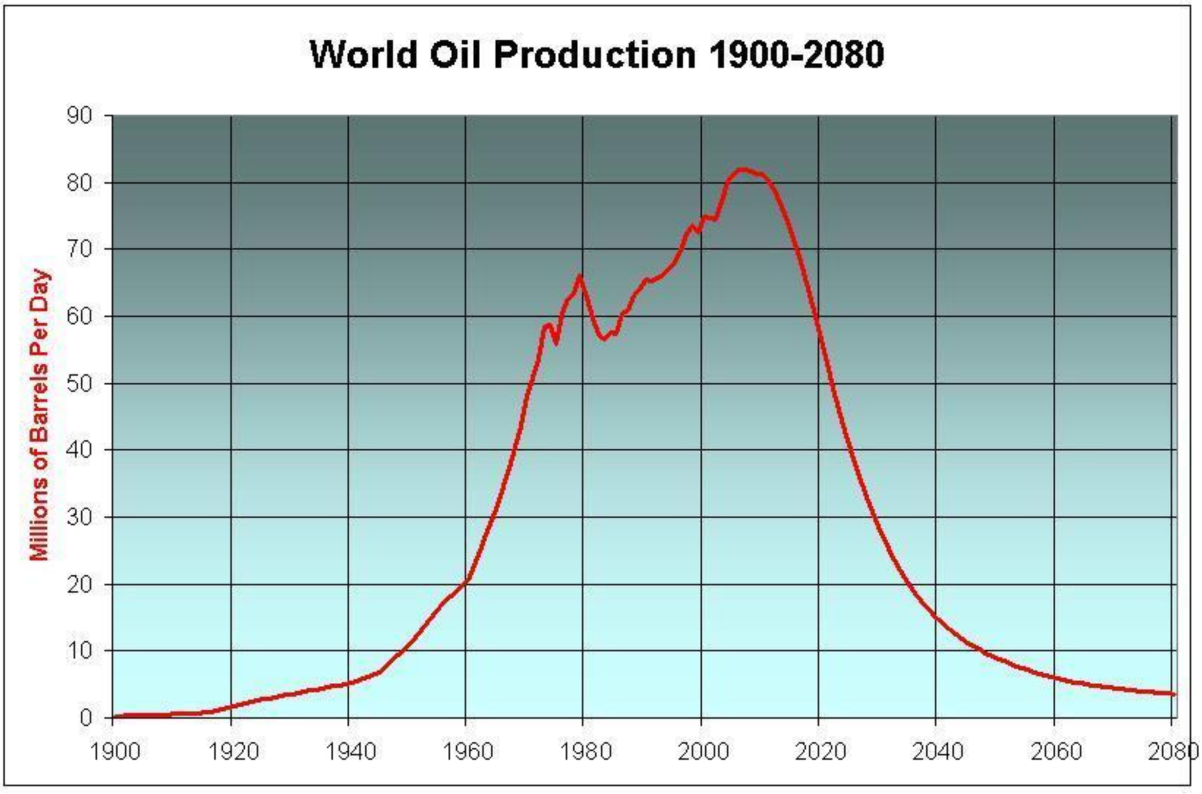


Fig.1.2 World oil production (Source: [http://www.paulchefurka.ca](http://www.paulchefurka.ca))

This is the fundamental cause of the greenhouse effect. Over population has resulted in change of land use and deforestation in turn causing draughts are the primary reasons for greenhouse effect.

- Global warming- This takes place primarily due to increasing levels of atmospheric greenhouse gases. An increase in surface temperature of earth due to greenhouse gases, which traps the heat in atmosphere. As this phenomenon is occurring slowly the human beings are prone to these effects. In the recent years many cities in India have recorded highest temperatures which is mainly due to global warming.
- Climatic changes- Frequent climatic changes have also resulted due to the increased dependence on the fossil fuels. Increased pollution, forest fires, deforestation, draughts, shifts in the rainy seasons are primarily due to these effects.
- Rise in sea level- The climatic changes has melted most of the glaciers. Warmed up water will result in destruction of many plant and animal species, thereby posing a great threat to wipe out the islands in the oceans and flooding of cities located on the sea shore.
- Acid rain- The emissions during combustion of fossil fuel contains oxides of sulphur and nitrogen. These get carried away by the water vapour and form clouds. The resulting rain has corroded several monuments, for instance, the black spots on the iconic Taj Mahal.
- Human health- There are more than 35 case studies of lung cancer incidents and mortality due to diesel emissions have been published till date (Attfield et al., 2012). Pollution in mines, especially in underground mines, is very harmful as the particulate matter gets entrapped in human respiratory system for a very long time and can cause serious health problems for the workers.

## 1.5 ALTERNATIVES TO FOSSIL FUELS

In olden days people used to burn wood for cooking and to keep their surrounding warm so as to protect from cold. The energy obtained from plants and its products are called as biomass energy. Even today wood is one of the biggest biomass energy resource. Other biomass sources are food crops, grass, agriculture waste, algae with oil, organic municipal waste, industrial waste etc. These biomass can be converted into fuels called as biofuels like alcohol or biodiesel. Alcohol is produced by gasification of biomass. During the process synthesis gas ($H_2$ +CO) is produced that can be converted in to alcohols by a chemical process in the presence of a catalyst.

Biodiesel is produced by combining alcohol with plant oil or animal fats in presence of a catalyst. Initially free fatty acid (FFA) content in the oil is tested. If it is higher than 1.5, then acid esterification is carried out to reduce FFA. After that it is mixed with sodium hydroxide (catalyst) and alcohol (methanol or ethanol) to carry out trans-esterification. The methyl or ethyl ester of the oil so obtained is called as biodiesel, which is a long chain alkyl ester.

Due to the environmental problems associated with the use of fossil fuels, now many researchers are focusing upon the usage of alternate energy sources such as biofuels instead of using natural oil resources and its associated derivatives. Using biofuels as an alternate fuel will ensure a balance between the agriculture, economic development and environment (Saddu and Kivade 2014). The Government of India also has considered the biofuels as supplementary to the existing fuels, as they not only provide employment opportunities to the rural people but also guarantee the farmers and poor a living (Subramanian et al. 2005; Natesan 2013).Therefore, biofuels derived from different vegetable and animal sources is one of the substitute for conventional fossil fuel.

Many researchers have demonstrated the production of biofuels from jatropha, pongemia, palm oil, sun flower oil, coconut oil, cottonseed oil, waste cooking oil, cardanol, mustard oil, fish oil etc., which are used in diesel engines.

## 1.6 CARDANOL AS A FUEL

Cardanol is extracted from Cashew Nut Shell Liquid (CNSL), which is a by-product during cashew nut processing. Cashew trees (Anacardium accidenate) are extensively grown in Africa, Philippines, Tanzania, Brazil, Madagascar and other tropical countries. In India it is cultivated in Karnataka, Kerala, Andhra Pradesh, West Bengal, Assam, Maharashtra, Goa, Tamil Nadu and Orissa. It is cultivated in 0.70 million hectares of area in India, with a total production of 0.4 million tons of cashew nuts (Mallikappa et al. 2011).Cashew shell is separated during the process of cashew kernel production. From these shell, Cashew Nut Shell Liquid is separated by different methods.

**Extraction of Cashew Nut Shell Liquid (CNSL):** There are different methods of separating CNSL from cashew shell.

**i. Thermal extraction**

Cashew nuts are roasted in drums to extract the oil. During roasting the CNSL comes out and the nut becomes brittle that makes cracking easy. The nuts are roasted at 185°C which extracts about 90% of CNSL.

**ii. Hot oil bath process**

The raw cashew nuts are filled in a cylinder through which steam at 250ºC is passed for three minutes. Due to the heat of steam CNSL oozes out from the shells. The process is repeated to extract maximum liquid.

**iii. Mechanical pressing process**

During this process the cashew nut shells are pressed at high pressure in a screw press or hydraulic press. CNSL comes out of the shell due to high pressure. But complete extraction of oil is not possible in this method.

**iv. Solvent extraction**

In this method the cashew nut shell is cut to small piece and added to organic solvent. The CNSL comes out of the shell and is added to the solution. The CNSL is separated by boiling the solution. The solvent is condensed and reused.

By the distillation of CNSL at 200°C to 240°C under reduced pressure Cardanol is produced. Double distillation is carried out to improve the purity of Cardanol. Cardanol diesel blends have been tested as fuel in diesel engine and obtained good performance with reduction in exhaust emissions (Velmurugan et al. 2011; Mallikappa et al. 2012; Radhakrishnan et al. 2014; Santhanakrishnan and Ramani 2015). To reduce viscosity, cashew nut shell liquid (CNSL) has been blended with camphor oil and the blend was tested in diesel engine by Kasiraman et al. (2012). The results of this study revealed that 70% CNSL and 30% camphor oil blend shows nearly similar behaviour to diesel fuel in terms of emissions and performance. Vedharaj et al. (2015) have explored the performance of compression ignition engine fuelled with pre-heated methyl ester of cashew nut shell liquid. The authors reported that pre heated CNSL methyl ester may be economically used as compression ignition (CI) engine fuel with improved performance and combustion. Jagadish et al. (2012) have mixed 10% methanol with cardanol to reduce the flash point during the test in diesel engine and also resolved that during trans-esterification of Cardanol glycerol was not produced.

### 1.7 ORIGIN OF WORK

The inventor of diesel engine Rudolf Diesel in 1900 operated his engine with peanut oil. Since petroleum diesel with superior property is easily available, people preferred to use petroleum diesel in diesel engines. Now the situation has changed, the fossil fuel resources are vanishing due to its increased excavation and the attention is towards invention of an alternative to diesel fuel that is biofuel. Due to their higher viscosity usage biofuels in diesel engines prone to problems like choking of fuel injector, gumming etc. To overcome these snags, biofuels are tested in diesel engines in different ways like preheating the biofuel, addition of methanol or ethanol to biofuel, blending of camphor oil etc. Cotton seed oil, honne oil, mustard oil canola oil and ox tallow have been experimented in diesel engine by blending with kerosene and favourable outcome of performance and emissions were reported (Aydin et al. 2015; Venkanna et al.2011; Azad et al. 2013; Roy et al. 2014; Silva et al.2014). Azad et al. (2012) have demonstrated that there is a reduction in brake specific fuel consumption (BSFC) and increase in thermal efficiency by using raw mustard oil blended with diesel.

The domestic usage of kerosene is reduced owing to various policies of the Government. Although its production will not reduce as long as petroleum oil is available, since it is a by-product obtained during the fractional distillation process of crude oil. Cardanol is abundantly available in India at lower cost. The Cardanol kerosene blends have not yet been tested as fuel for diesel engine. The performance and emissions of a diesel engine with any fuel mainly depends on compression ratio, injection pressure and injection timing. Hence, keeping in view of these aspects, the present research work is focussed to study the performance of raw cardanol oil blended with kerosene in a compression ignition engine with the following objectives.

1. To determine the properties (viscosity, flash and fire point, density and calorific value) of cardanol-kerosene blends.

2. To study the performance and emission characteristics of different blends in a four stroke naturally aspirated diesel engine.

3. To find the effect of compression ratio, injection pressure and injection timing on the performance and emission characteristics of different blends.

## 1.8 SCOPE OF THE WORK

The scope for biofuel is increasing due to possible future oil crisis. Some of the diesel engine manufacturers have already permitted biodiesel to some extent. So testing of a new blend of biofuel (i.e. raw Cardanol and kerosene) in diesel engine as a fuel, will improve the scope of biofuel, usage of raw oil also cuts down the cost of esterification. This dissertation offers a new blend to the field of biofuel.

## 1.9 THESIS OUT LINE

This thesis consists of ten chapters. The first chapter covers the general introduction followed by scope and objectives of the work. The second chapter gives the brief literature review related to the objectives of the work. The third chapter gives the information about experimental set-up and instrumentation used in the study. Chapter four explains performance and emission analysis of cardanol kerosene blends in diesel engine. Chapter five elaborates the effect of compression ratio on performance and emission of diesel engine operated with Cardanol kerosene blend. Chapter six discusses the effect of injection pressure on performance and emission of diesel engine operated with Cardanol kerosene blend. Chapter seven explains the effect of injection timing on performance and emission of diesel engine operated with Cardanol kerosene blend. Chapter eight covers the optimisation of performance and emission parameters. Chapter nine briefs economic analysis of the biofuel blend with respect to mining sector. Chapter ten includes the conclusions of the present research work and the scope for future work.

# CHAPTER 2

# LITERATURE REVIEW

## 2.1 INTRODUCTION

Edible and non-edible vegetable oil and animal fats are the feed stocks for biofuel. The idea of using vegetable oil as diesel engine fuel credits to Rudolph Diesel, who exhibited the operation of diesel engine with peanut oil in 1900 at Paris. At that time researchers did not give much attention on biofuels, since crude oil was amply available at low price. After the crude oil crisis during 1970's the cost of fossil fuel started increasing randomly. After that the scientists thought of developing an alternative to fossil fuel and possibility of usage of biofuels.

Many researchers have conducted experiments on biofuels as an alternative to diesel fuel. The existing works on usage of biofuel in diesel engine have been referred in two categories.

i. Use of straight vegetable oils and its blends
ii. Use of biodiesel and its blends

The earlier research related to the effects of operating parameters like compression ratio, injection pressure and injection timing on performance of diesel engine with biofuel operation has been presented in this section.

## 2.2 STUDIES ON STRAIGHT VEGETABLE OILS AND ITS BLENDS

The oil extracted from vegetable seeds or fats without any modification in its chemical structure is called as straight vegetable oil (SVO). The waste vegetable oil from various origins and the oil obtained from various feed stocks are included under SVO. Many investigators have used SVO as diesel engine fuel either in pure form or by blending with diesel fuel.

### 2.2.1 Pure Vegetable Oils as Engine Fuel

Vegetable oils and waste vegetable oils can be used as an alternative fuel in compression ignition engine without any chemical modification. Numerous scientists have tested diesel engine with oils like rapeseed oil, sunflower oil, palm oil, peanut oil, jatropha oil and orange peel oil as fuel without any blending.

Altin et al. (2001) evaluated performance of diesel engine run with some vegetable oils like raw sunflower oil, raw cotton seed oil, raw soybean oil, refined corn oil, opium poppy oil and refined rape seed oil. Their test results supported use of vegetable oil as an alternate fuel with respect to its performance. They also reported the problems of cold flow, atomisation and particulate emission due to higher viscosity and poor cold flow properties.

Bari et al. (2002) have used crude palm oil (CPO) as diesel engine fuel. At room temperature they observed that the viscosity of oil was ten times higher than diesel fuel. They have preheated the oil above 60°C to reduce the viscosity. Performance of engine with CPO was comparable with diesel fuel. They also reported that carbon monoxide (CO) emission was 9.2% higher and oxides of nitrogen (NOx) emission 29.3% higher than diesel fuel.

Pugazhradivu and Jeyachandran (2005) experimented on preheated waste frying oil as an alternate fuel in compression ignition engine. Viscosity of oil was reduced by preheating. The waste frying oil was preheated up to 135°C in order to moderate its viscosity to that of diesel fuel at 30°C. The authors resolved that waste frying oil preheated up to 135°C can be used as a fuel in diesel engine with improved performance and reduced CO and smoke emission.

Cetin and Yuksel (2007) investigated performance of Mercedes-Benz OM 616 pre-chamber diesel engine operated with hazelnut oil. Higher specific fuel consumption was reported for hazelnut oil than diesel fuel. Emissions of CO, unburned hydrocarbon (HC) and smoke were higher than diesel emission.

Deepak and Avinash (2007) conducted experiments on agricultural diesel engine operated with jatropha oil. They reduced the viscosity by preheating the oil. Preheated oil up to 100°C had viscosity comparative with diesel fuel. For the jatropha oil operation emissions of HC, CO smoke and NOx were higher than diesel fuel. Exhaust gas temperature and specific fuel consumption of unheated oil was higher than the preheated oil.

Purushothaman and Nagarajan (2009) have operated a constant speed diesel engine with orange oil as an alternative fuel and obtained brake thermal efficiency of 31.7% and 29.3% for orange oil and diesel fuel respectively at peak load operation. HC, CO and smoke reduced with orange oil compared to diesel fuel operation but NOx increased.

Mani et al. (2011) analysed performance of a constant speed Kirloskar make diesel engine operated with waste plastic oil. They observed that pure waste plastic oil can be used to run the engine. By using this oil as fuel in diesel engine, the thermal efficiency was higher than diesel fuel operation up to 80% load. Emissions of NOx increased by 25%, CO raised by 5%, smoke by 40% and hydrocarbon by 15%.

Yilmaz and Morton (2011) explored performance of a Yanmar generator diesel engine using preheated sunflower oil, peanut oil and canola oil. At full load condition brake thermal efficiencies developed by these oils were 27%, 28% and 26% for sunflower oil, peanut oil and canola oil respectively, but for diesel fuel it was 27%. All these oils were preheated up to 90°C during the test. NOx emission increased for the oils compared to diesel fuel.

Paulsen et al. (2011) have operated an agricultural tractor engine with rape and camelina oil and their mixture. The tests were carried for 1000 hours of operation in the field. Results indicated higher NOx emission than diesel fuel and more CO emission at lower loads. Emissions of HC, particulate and carbon dioxide ($CO_2$) were comparable with diesel fuel. The authors suggested for checking of the injectors for deposits and chocking

Hossain and Davies (2012) studied on multi cylinder CI engine operated with pre heated non-edible oils, such as karanja and jatropha. The results revealed that the specific fuel consumption was 3% higher for the oils compared to that of diesel fuel. The NOx emission increased by 8% and 3% increase in cylinder pressure related to diesel fuel. They recommended that after preheating the karanja and jatropha oils can be used as fuel for diesel engine.

Golimowski et al. (2013) have tested performance of a tractor engine operated with raw rapeseed oil. They operated the engine initially for 230 hours and then for 1000 hours.

During the test 12% to 14% power drop was observed for rapeseed oil compared to diesel fuel. NOx emission increased because of higher oxygen content in the biofuel. The efficiency of engine was 34.4% and 33.3% for vegetable oil and diesel fuel operation respectively.

### 2.2.2 Pure Vegetable Oil-Diesel Blends as Engine Fuel

High viscosity of pure vegetable oils leads to poor atomisation, dispersion and deposits in combustion chamber. To overcome these problems many researchers prepared blends of straight vegetable oil with diesel fuel and obtained good results during the test in diesel engine as a fuel.

Nwafor and Rice (1996**)** studied “Performance of rapeseed oil blends in a diesel engine” and found that the higher viscosity of raw rapeseed oil reduced the atomisation and combustion rate. They have tested diesel blends with rapeseed oil and noticed power loss with raw rapeseed oil. At lower loads they observed knocking (abnormal combustion which gives high rate of pressure rise, may be due to fuel properties or engine parameters. It can be identified by excessive audible noise and engine vibration.) with raw oil blends because of lower cylinder temperature and pressure which leads to long delay period. There was reduction in unburned hydrocarbon emission up to 50% raw oil blends when compared to the base line experiments.

Kalam et al. (2003) have tested a four cylinder compression ignition (CI) engine operated with blends of diesel and pure coconut oil. All the emissions were reduced with increase in percentage of coconut oil. Up to 30% coconut oil blend the performance was better than diesel fuel with lower emissions. Problems of injector deposits were not reported during the tests. The coconut oil blends have the similar brake power up to 30% blending.

He and Bao (2003) have tested pure rapeseed oil in S195 diesel engine. Due to higher viscosity of the oil the authors faced problems of high carbon deposit in the engine. To reduce the viscosity the oil has been mixed with diesel. It has been reported that with increase in percentage of oil in the blend the specific fuel consumption (SFC) increased. For 30% oil and 70% diesel blend minimum SFC was reported, which is 3% to 5% more than the diesel fuel.

Pramanik, K. (2003) explored the properties and application of jatropha oil as CI engine fuel. Various blends of Jatropha oil and diesel fuel were prepared and tested in diesel engine. Results were compared with pure diesel fuel performance. Effect of temperature on viscosity of oil were investigated. Viscosity of blends having 50% to 60% olive oil were close to diesel fuel at 60°C. Up to 50% oil blend thermal efficiency was comparable with diesel with lower emissions.

Forson et al. (2004) explored the performance of a direct injection diesel engine run with raw jatropha oil and its blends with diesel fuel. They have tested 50% diesel 50% jatropha oil, 80% diesel 20% jatropha oil and 97.4% diesel 2.6% jatropha oil blends. Reported that pure jatropha oil and blends of oil with diesel revealed similar performance and emission compared to diesel fuel. Jatropha oil acts like ignition accelerator and also reduced the exhaust gas temperature. 97.4% diesel and 2.6% jatropha oil blend showed comparable results with pure diesel fuel performance and emission.

Huzayyin et al. (2004) evaluated the performance of diesel engine with jojoba oil and diesel fuel blends. By the study of physical and chemical properties of jojoba oil the authors decided that the oil can be a good alternative fuel to diesel engine. They tested blends of jojoba oil with gas oil (diesel fuel with red dye) by blending 20%, 40% and 60% oil. The viscosity of 60% oil and 40% gas oil was comparable with diesel fuel. The reports of chemical analysis indicates that the jojoba oil has a straight chain chemical structure without any glycerine content, which reduces the problems of long term usage. The test results pointed out a reduction in NOx and soot formation with a trivial power loss compared to gas oil performance

Ramadhas et al. (2005) investigated performance of a four stroke diesel engine operated with rubber seed oil. The experimental results revealed that highest thermal efficiency was obtained for 80:20 (rubber seed oil: diesel). The thermal efficiency of 60:20 blend was very nearer to the diesel fuel. The smoke emissions from 50% and 60% oil blends were lower than diesel. The authors resolved that 50 to 80 percent rubber seed oil blend with diesel can be conveniently used as fuel in CI engine without any major variations in engine.

Hebbal et al. (2006) have investigated on the use of deccan hemp oil as an alternate fuel for diesel engines. They found that up to 25% raw oil can be blended with diesel fuel without any preheating, but for using 100%, 75% and 50% oil in the blend preheating up to 95°C, 80°C and 70°C is required. For 50% blend performance and emissions were better than the other blends. They resolved that without modification of engine up to 50% blends with diesel can be used as a fuel.

Deepak et al. (2008) analysed the performance of stationary diesel engine operated with diesel blends of linseed oil, mahua oil and rice bran oil. Various blend of oil with 10%, 20% and 30% mahua and rice bran oil and up to 50% linseed oil with diesel fuel were tested in the engine. They reported that 50% linseed oil blend had higher thermal efficiency than diesel fuel but the smoke was higher than diesel. 30% mahua oil blend showed performance similar to diesel fuel. 20% rice bran oil blend reported minimum specific fuel consumption.

Altun et al. (2008) investigated emissions and performance of CI engine run with sesame oil and diesel blends. A blend of 50% sesame oil with diesel was prepared for the test. The emissions of CO, HC and NOx were lower for sesame oil diesel blend compared to pure diesel operation. They concluded that the sesame oil diesel blend can be used in diesel engine without much alteration.

Agarwal and Rajamanoharan (2009) evaluated performance of an agricultural diesel engine operated with karanja oil blends with diesel fuel. Various blends were prepared by mixing 10%, 20%, 50% and 75% oil with diesel fuel. The oil was preheated using waste heat from exhaust gas. Improvement in performance and emission were observed compared to diesel fuel. Nitrogen oxide emission were lower than diesel with unheated and preheated oil blends. Up to 50% karanja oil blends can be used in diesel engine with improvement in performance and lower emissions.

Haldar et al. (2009) experimented putranjiva roxburghii oil as a fuel in a variable compression ratio engine. Up to 30% oil blends with diesel fuel emissions of CO, HC and smoke were lower than diesel fuel with comparable brake thermal efficiency, but further increase in oil percentage the thermal efficiency deteriorated. The authors concluded that up to 30% putranjiva roxburghii oil blends with diesel fuel can be used as fuel for diesel engine with minimum emissions.

Devan and Mahalakshmi (2009) analysed performance of Kirloskar TAF1 engine operated with poon oil and its blends with diesel fuel. 20%, 40% and 60% poon oil blends were tested during the trial. Results revealed that NOx emission of poon oil blend were 32% lower than diesel fuel. HC and CO emission were higher for the poon oil blends, but smoke was 3% lower than neat diesel. Brake thermal efficiency reduced with increase in percentage of poon oil in the blend it was lower than diesel fuel at all loads.

Bajpai et al. (2009) studied performance of a constant speed diesel engine run with pure karanja oil blends with diesel fuel. Tests were conducted with various blends having 20%, 15%, 10% and 5% oil with diesel. Better performance was reported with respect to thermal efficiency, smoke emission, and NOx emission than diesel fuel because of self-lubricity of oil and the in build oxygen content.

Haldar et al. (2009) investigated performance and emission of a variable compression ratio diesel engine operated by jatropha oil, karanja oil and putranjiva oil blends with diesel fuel. The authors revealed that viscosity can be economically reduced by degumming process. Among the three oils jatropha oil is better than the other two with respect to performance and emission. Up to 20% blend can be easily used in diesel engine without any modifications.

Hazar and Aydin (2010) analysed the performance of diesel engine fuelled with raw rapeseed oil and diesel blends. They tested two blends, 50% oil blend and 20% oil blend. The raw oil has been preheated up to 100°C in order to reduce its viscosity. They reported a reduction in mass of fuel consumed with preheating. The emissions of CO and smoke were reduced with preheating. NOx emission was lower than diesel fuel. The vegetable oils can be used as fuel in diesel engine by preheating, which reduces emission.

Rakopoulos et al. (2011) investigated performance of a bus engine run with four straight vegetable oils. Olive oil, corn oil, cotton seed oil and sunflower oil blends with diesel fuel were tested in the engine. 10% and 20% blends of all these oils were mixed to get different blends of fuel. Smoke emissions were reduced for all the blends compared to diesel fuel. NOx, HC and CO emissions increased with increase in oil percentage in the blend. Thermal efficiency of all the blends were compatible with diesel fuel. Performance of olive oil blend was better than the other oil blends.

Acharya et al. (2011) studied the performance of a diesel engine operated by rice bran oil. To reduce viscosity the oil was preheated and blended with diesel fuel. Performance of blends were very close to that of diesel fuel. Exhaust emissions were lower for the blends compared to diesel. The authors suggested that the crude rice bran oil can be used in diesel engines in rural areas to operate agricultural equipment.

Leevijit and Prateepchaikul (2011) investigated performance of a turbocharged diesel engine run by crude palm oil-diesel blends. Blends were prepared by mixing 20%, 30% and 40% palm oil with diesel. Tests were conducted for 1000 hours of operation. It was reported that specific fuel consumption increased by 7%, brake thermal efficiency reduced by 5%, smoke reduced by 40% and CO reduced by70% with the palm oil blends than the diesel fuel. NOx emission slightly increased compared to diesel fuel operations.

Ndayishimiye and Tazerout (2011) studied on various techniques to use palm oil as diesel engine fuel. They tested preheated palm oil, diesel blends with palm oil and palm oil methyl ester with waste cooking oil. Results indicated that specific fuel consumption increased for preheated palm oil compared to that of diesel. There was reduction in HC emission with the pre heated palm oil. NOx emissions were slightly higher for methyl ester than the pure palm oil.

Kalam et al. (2011) explored performance of a multi cylinder diesel engine run with blends of diesel with waste cooking oil. Two blends were prepared by mixing 5% waste coconut oil and 95% diesel and 5% waste palm oil and 95% diesel. Results revealed that the brake power reduced by 1.2% and 0.7% for palm oil and coconut oil blends. HC and CO emissions reduced for both the blends but NOx was increased for palm oil blend and reduced for the coconut oil blend.

Ndayishimiye and Tazerout (2011) tested palm oil and diesel fuel blends in diesel engine to find performance and emissions characteristics. Initially experiments were carried with diesel fuel and then with biofuel blends. It was noticed that heating value of blends were lower than diesel fuel and increasing palm oil percentage in the blend reduces the heating value. Increase in brake thermal efficiency was noticed for palm oil blends. CO emissions was increased for the blends. NOx emissions increased for palm oil blends with increase in oil ratio in the blend.

Misra and Murthy (2011) tested blends of soap nut oil with diesel in a constant speed diesel engine at various loads. Blends were prepared by mixing 10%, 20%, 30% and 40% pure soap nut oil with diesel. During the test 10% and 20% blends revealed lower CO emission than the other blends. The 40% soap nut oil blend performed very well and had 35% reduction in NOx emission compared to diesel fuel. Performance and emissions of 10% soap nut oil blend was very close to diesel fuel.

Labecki et al. (2012) analysed performance of multi cylinder diesel engine operated with rapeseed pant oil and its blends with diesel fuel. The authors revealed that higher viscosity of plant oil adversely affects performance and durability of engine for long run operation. They tested 50% and 30% oil blends with diesel at various injection pressure (IP) and injection timing (IT). For increased IP and retarded IT, 30% blend shown soot emission equivalent to diesel fuel with reduction in NOx emission.

Daho et al. (2013) investigated performance and emissions of a diesel engine (model Hatz ID80) run with cottonseed oil and its blend with diesel fuel. The results reveal that specific fuel consumption (SFC) increases with rise in percentage of cottonseed oil in the blend. At peak load for pure cottonseed oil the SFC was 21% higher than diesel fuel. CO emission was higher and NOx emission was lower than diesel fuel.

Huseyin Aydin (2013) investigated performance of a coated diesel engine operated with pure sunflower oil and cottonseed oil blends with diesel fuel. In the study 15% oil and 35% oil blends with diesel fuel were tested. The blends produce less emissions than diesel fuel. Performance of sunflower oil blends were superior to the cottonseed oil with respect to power and emissions.

## 2.3 STUDIES ON BIODIESEL AND ITS BLENDS

Use of straight vegetable oils as fuel in diesel engine is prone to deposits of carbon and gum present in the oil. The oil is converted into biodiesel to remove its gum content and then used as a fuel in engines.

### 2.3.1 Biodiesel Production Process

High viscosity of vegetable oils and animal fats acts as major barrier to use them as biofuel in diesel engine. Viscosity of vegetable oils and fats are reduced by converting into biodiesel through trans-esterification process. Trans-esterification is a reaction between triglyceride present in the oils/fats and an alcohol in presence of catalyst (sodium hydroxide) to form esters and glycerol. The ester so formed is called as biodiesel. The chemical reaction is shown in Figure 2.1.

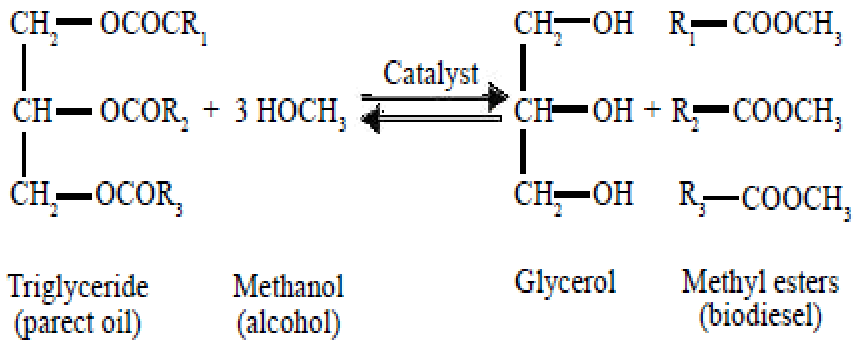


Fig.2.1 Trans-esterification reaction

Usually methyl alcohol or ethyl alcohol are used for the reaction because of their availability, chemical advantage and low cost. Also sodium hydroxide (NaOH) and tri-glycerides present in oil dissolves very easily in methanol and ethanol.

#### 2.3.1.1 Trans-esterification process

To produce biodiesel from any oil or fat, initially one litre oil is heated to remove moisture and then added to inner vessel of trans-esterification reactor. Temperature inside the reactor is maintained at 60°C.

In a separate vessel about 6 grams of NaOH pellets are dissolved in 200 ml methyl alcohol. The mixture is stirred well to ensure complete dissolution of sodium hydroxide. This mixture is called as sodium methoxide.

The sodium methoxide solution is added to the oil in the reactor vessel and stirred continuously at constant temperature of 60°C. The stirring is continued for one hour until the reaction completes and separation of glycerol is started.

The mixture is transferred to a separator funnel and allowed to settle down by gravity separation. The separation process requires minimum of eight hours or preferably overnight. Two separate layers are formed. Dark coloured semi- liquid glycerine at bottom and honey coloured biodiesel at top as shown in Figure 2.2.

Fig. 2.2 Separation funnel

Without disturbing the funnel, the bottom layer glycerine is drained carefully. The layer present in the funnel is methyl ester of the oil. The methyl ester is transferred to another vessel and washed with warm distilled water to clean the remaining glycerine and other impurities. The water washing process is repeated three times to ensure removal of all impurities. The methyl ester of oil so obtained is ready to be uses as a biodiesel.

The biodiesel is one of the renewable fuel used in diesel engines. Some scientists tested pure biodiesel as fuel in CI engine. To reduce viscosity of biodiesel many researchers have prepared blends of biodiesel and diesel. These blends were tested in diesel engine to study its performance and emission characteristics.

### 2.3.2 Pure Biodiesel as Engine Fuel

Dorado et al. (2003) studied performance of a Perkins diesel engine run with waste olive oil methyl ester at various operating conditions. Characteristics of neat diesel and neat olive oil methyl ester were tested. The test results revealed that emissions of CO, $SO_2$, smoke and NO were reduced for olive oil biodiesel compared to diesel. Specific fuel consumption slightly higher than diesel fuel. Combustion efficiency of olive oil and diesel were similar.

Karabektas et al. (2008) investigated performance of diesel engine with pre heated cottonseed methyl ester. The ester was preheated to various temperatures (30°C, 60°C, 90°C and 120°C). Higher brake thermal efficiency was recorded for the pre heated cottonseed methyl ester compared to that of diesel. At 90°C preheating maximum thermal efficiency was obtained. CO emissions reduced and NOx emissions increased with cotton seed methyl ester compared to diesel fuel emission.

Banapurmath et al. (2008) explored performance of CI engine operated with jatropha oil, sesame oil and honge oil methyl esters. Thermal efficiency with esters were lower than diesel. The thermal efficiency of 30.4%, 29.5%, 29% and 31.25% were reported for sesame oil, honge oil, jatropha oil methyl esters and diesel fuel respectively. Emissions of HC, CO and smoke were little higher for the esters compared to diesel. Among the three esters tested sesame oil methyl ester was better than the other two with respect to performance and emissions.

Kapilan Natesan (2013) has preheated mahua oil biodiesel using exhaust gas and tested in a diesel engine. By preheating brake thermal efficiency increased and HC, CO, smoke emissions reduced. NOx emissions increased due to high temperature of fuel. He concluded that preheated mahua oil can be used as fuel for CI engine.

#### 2.3.2.1 Advantages and disadvantages of biodiesel

Biodiesel is a natural, renewable source of energy which is an alternative to petroleum diesel. Use of biodiesel as CI engine fuel has many advantages and disadvantages.

Advantages:

i. It is easily available in nature.
ii. It can be used in diesel engines very easily with minor engine modification.
iii. It is nontoxic and non-inflammable.
iv. It reduces the exhaust emissions.
v. Low sulphur content.
vi. It is biodegradable.
vii. Renewable in nature.
viii. It gives support to agricultural growth.
ix. It can be grown at any place where it is required.
x. High flash point reduces the risk of fire hazard at storage.

Disadvantages:

i. It has high viscosity which leads to clogging problems in the injection system.
ii. Nitrogen oxides emissions increases.
iii. Its calorific value is low.
iv. Engine wear is more.
v. It produce low torque due to lower energy content.
vi. Volatility is lower.

### 2.3.3 Biodiesel-Diesel Blends as Engine Fuels

Raheman and Phadatare (2004) investigated on use of karanja methyl-ester and its blends with diesel fuel as fuel in single CI engine. They reported that by using karanja methyl-ester and diesel blends, emissions such as CO, smoke and NOx reduced. Brake power increased by 6% up to 40% blending of the ester. The authors concluded that 40% karanja methyl-ester blends with diesel fuel can be used in CI engine without any major alteration in the engine.

Ramadhas et al. (2005) experimented on the use of rubber seed oil biodiesel in 5.5 kW naturally aspirated diesel engine operated at 1500 rpm. They noticed reduction in CO, $CO_2$ and smoke emissions for biodiesel blends than diesel fuel. For 10% biodiesel blend brake thermal efficiency was 3% more than diesel. With increase in biodiesel percentage in the blends the exhaust gas temperature increased slightly, but for 100% biodiesel operation the exhaust gas temperature was very high.

Nabi et al. (2006) conducted experiments on neem oil methyl ester as a fuel in CI engine at various loading conditions. Various blends having 5%, 10% and 15% neem oil methyl ester were tested. Compared to diesel fuel the neem oil blends had lower emissions of CO and smoke. Highest thermal efficiency was recorded at 1000 rpm at which NOx emission was 330 ppm and 310 ppm for 15% neem oil blend and diesel fuel respectively. The smoke emissions were 7.7 FSN for diesel and 7.3 FSN for 15% neem oil blend at the same operating condition.

Sahoo et al. (2007) analysed methyl-ester of polanga seed oil and its blends with diesel as an alternate fuel in CI engine. Various percentages of polanga seed oil were blended with diesel. Among all the blends, 60% polanga seed oil ester blend had lowest smoke emission, which was lower than diesel fuel emission also. 100% polanga oil operation exhibited least exhaust gas temperature and lower NOx emissions.

Bueno et al. (2011) studied performance of CI engine operated with soybean ethyl ester and its blends with diesel fuel. During the test up to 30% soybean ethyl ester was blended with diesel. Highest brake thermal efficiency was obtained for 20% blend compared to other blends, which was 4.2% higher than diesel. Power output of 10% and 20% blends were higher than pure diesel operation.

Muralidharan and Vasudevan (2011) investigated on performance of variable compression ratio diesel engine operated with methyl-ester of waste cooking oil and its blends with diesel. The tests were conducted at various compression ration varying from 18:1 to 22:1. The brake thermal efficiency of 40% oil blend at 21:1 compression ratio was higher than diesel fuel at the same operation condition. Exhaust gas temperature of blends were higher than diesel fuel at lower compression ratio and at higher compression ratio it was lower than diesel fuel. HC and NOx emission were higher for the biodiesel blends compared to diesel fuel operation.

Harikude and Padalkar (2012) explored the performance and emission of a 3.7 kW single cylinder diesel engine run on waste fried oil methyl-ester and diesel fuel blends.

They reported thermal efficiency of 50% biodiesel blend was 6.8% higher, smoke was 47% lower and CO emission were 45% lower than diesel fuel. The specific fuel consumption and exhaust gas temperature increased with increase in percentage of waste fried oil methyl-ester in the blend.

Ravikumar et al. (2013) tested mahua biodiesel blends in a coated diesel engine and found that coating with thermal barrier improved thermal efficiency and specific fuel consumption. B25 blend showed performance similar to diesel HC, $CO_2$ and smoke emissions are reduced but NOx increased due to higher temperature inside combustion chamber.

Celikten et al. (2013) analysed performance of a constant speed 46 kW diesel engine operated with hazelnut and rapeseed oil methyl ester blended with diesel fuel. Blends were prepared by mixing 50% diesel and 50% rapeseed oil, 50% diesel and 50% hazelnut oil, and 50% diesel and 25% hazelnut oil, 25% rapeseed oil. Lowest specific fuel consumption was reported for 50% diesel and 50% rapeseed oil blend. For all the blends emissions of HC, CO and smoke were lower than diesel fuel, but NOx emission was little higher.

Silitonga et al. (2013) have tested performance of a diesel engine fuelled with ceiba pentandra and concluded from their study that Ceiba pentandra biodiesel when blended up to 50% with diesel, the HC, CO and smoke was reduced but nitrogen oxides (NOx) emissions were increased. Also they observed that for blending ratio of 10% there was a good performance.

Ahmed et al. (2014) found that when mustard biodiesel was blended with diesel up to 20%, there was a reduction in hydrocarbon (HC) and carbon monoxide (CO) emissions during the test in diesel engine as a fuel.

Pali et al. (2014) Sal methyl ester was tested in a diesel engine and concluded that there was a reduction in brake thermal efficiency and increase in brake specific energy consumption (BSEC) with increase in biodiesel in the diesel blend. CO, HC and smoke emissions were less at higher loads but NOx emissions increased.

Ong et al. (2014) Jatropha curcas, Ceiba pentandra and Calophyllum inophyllum biodiesel was tested in a diesel engine and for 10% blends of biodiesel with diesel gave higher torque, brake thermal efficiency and power. All the blends reduced the CO and HC emission but increased NOx emissions. 10% Jatropha biodiesel produced more torque than the other two biodiesels.

How et al. (2014) used ethanol as an additive to coconut oil biodiesel blend with diesel in a diesel engine and found that BSFC increased with addition of ethanol and improvement in brake thermal efficiency. NOx emission is more for coconut oil blend than pure diesel with addition of ethanol there was a reduction in NOx .Smoke emission and CO emission was reduced with addition of ethanol. A reduction in Heat release rate was observed with addition of ethanol due to cooling effect of alcohol and its lower calorific value.

Habibullah et al. (2014) have produced biodiesel from palm oil and coconut oil. The blends of these biodiesel with diesel are tested in a diesel engine. They tested 30% blends and mixture of two biodiesel (PB15CB15) .During test they observed a reduction in brake torque and increase in BSFC compared to neat diesel ,decrease in HC,CO emissions and increase in NOx emission. Performance of PB15CB15 is better than PB30 and CB30.

Nabi et al. (2015) have blended Licella biofuel with diesel and used in a diesel engine. For blends up to 20% there was no change in indicated and brake power. HC and NO emissions were more than diesel but particulate matter (PM) emissions is less.

## 2.4 STUDIES ON BIOFUEL BLENDS WITH KEROSENE AS DIESEL ENGINE FUEL

Many scientists conducted experiments on biodiesel blends with kerosene and diesel blends with kerosene, as fuel in compression ignition engine. By mixing kerosene to biodiesel cold flow properties improved with reduction in viscosity and flash point of the blend.

Yadav et al. (2005) have tested adulteration of diesel with kerosene and reported that the density does not change with increase in kerosene but viscosity reduced. The smoke opacity also reduced with increase in kerosene percentage.

Aydin et al. (2010) conducted experiments on air cooled 4-S diesel engine with cotton seed oil and kerosene blend and found that BK20 shows better performance than other blends. Nox and CO emissions are lower compared to other blends and brake thermal efficiency was high for this blend.

B.K.Venkanna et al. (2011) investigated the performance of a diesel engine operated with diesel, hone oil, kerosene and dimethyl carbonate (DMC) blends and observed increase in BTE with addition of kerosene and further improvement with the addition of DMC. All the exhaust emissions except NOx and smoke density were reduced with kerosene and DMC addition but NOx increased .The percentage increase of NOx is less than the percentage reduction in smoke density with increase in DMC. At 75%load net heat release increased with kerosene blend compared to diesel, addition of DMC increased peak cylinder pressure.

Azad et al. (2013) have studied performance of a diesel engine with mustard oil kerosene blends .They observed an increase of BSFC with increase percentage of biofuel. Maximum brake thermal efficiency (BTE) was observed for M30 blend. Highest exhaust gas temperature was noted for M100 and lower exhaust gas temperature for M40.The mean effective pressure increased with load.

Silva et al. (2013) tested a Rover IS/60 turbo shaft engine fuelled by ox tallow ethyl ester and kerosene blends. They found that brake specific fuel consumption (BSFC) decreased with increase in load, $CO_2$ concentration increased with load but CO emissions reduced. When percentage of ox tallow ethyl ester increased in the blend all the emissions were reduced.

Roy et al. (2014) conducted test on a 4 stroke 2 cylinder diesel engine using canola oil biodiesel blended with diesel, and wintronxc30 additive, and a blend of biodiesel with kerosene. Reported that for all blends bsfc increased with increase of biodiesel proportion. The fuel conversion efficiency has increased by 2.5% for pure biodiesel and with additive, but with kerosene biodiesel efficiency reduced with increase of kerosene percentage due to higher friction. NOx is reduced in kerosene blend compare to the other blends.in all load HC emission reduced with increase of biodiesel but with kerosene blends HC decreased for heavy load only.

Patil et al. (2014) studied effect of di ethyl ether (DEE) and kerosene blending with diesel on a 4-S diesel engine. An improvement in brake thermal efficiency was observed with addition of DEE to diesel up to 15% higher efficiency is obtained for DEE15K5D than other blends. The opacity reduced with DEE up to 15% then increased, by addition of kerosene smoke is reduced .NOx and CO reduced but HC increased with kerosene addition.

Uddin et al. (2015) investigated on 4stroke diesel engine fuelled with mustered oil and kerosene blend and reported that m20 (20% kerosene and 80% mustard oil) and m30 (30% kerosene and 70% mustard oil) can be suitably used in diesel engines. Brake thermal efficiency of 32.96% was observed for m30 blend.

## 2.5 CARDANOL

In present work Cardanol a by-product from cashew industry is used as a fuel for diesel engine. Cardanol is produced from cashew nut sell liquid, which is extracted from cashew nut shells. In the following section details about cashew nut, cashew nut shell liquid and Cardanol are discussed.

### 2.5.1 Introduction to Cashew Nut Trees

Cashewnut trees (Anacardium Occidentale) belongs to the Anacardiaceae family of plants. Native of the tree is Brazil, but now it has spread to Mexico, south and Central America and West Indies. During 1600s, to prevent soil erosion Portuguese introduced the cashew tree into India and Africa. It is widely planted for its nuts and fruits in the coastal regions of South Africa, Philippines, Tanzania, Sri Lanka and Madagascar. Now in India it is one of the major export earning crop. Cashew nut trees are grown in most of the states in India like Kerala, Karnataka, Goa, Maharashtra, Tamil Nadu, Andhra Pradesh, West Bengal, Jharkhand, Odisha, Gujarat, Assam, Pondicherry, Manipur, Meghalaya and Nagaland. During 2008-09 in India cashew trees plantations were made at about 893 thousand hectare area, but during 2016-17 plantations area has been increased to 1040 thousand hectares. Table 2.1 gives the state wise area and yield of cashew nut for the year 2016-17. Maharashtra state occupies large area (186.2 thousand hectares). Konkan region of Maharashtra having Ratnagiri, Thane and Sindhudurg districts are the major part of the state where cashew is cultivated. The cashew trees are tropical evergreen, drought resistant, which grows up to 12 metres high and spread symmetrically up to 25 metres surrounding. It protrudes reddish cluster of flowers, after 30 to 40 day kidney shaped fruit development starts. Fully grown cashew apple are red or yellow in colour. Figure 2.3 shows various stages of growth of cashew nut. At the end of each cashew apple a kidney-shaped nut, with a hard shell is present. It grows at the locations from sea level up to 1000 metres altitude and in regions where annual rainfall of 500 mm to 3750 mm. Maximum yield obtained at locations where annual rainfall lies between 889 mm to 3048 mm with red soil or sandy loam soil. Cashew trees bear fruits in the third or fourth year in good soil and in favourable weather and soil condition reaches to matured yield in seventh year. A matured tree produces an average of 7 kg to 11 kg nuts per year. The cashew trees can live up to 60 years but their yield drops drastically after 20 years.

Table 2.1 Area and Production of cashew 2016-17

| **State** | **A (000 ha)** | **P (000MT)** | **APY (kg/ha)** |
|---|---|---|---|
| Kerala | 90.866 | 83.98 | 962 |
| Karnataka | 127.86 | 85.147 | 672 |
| Goa | 58.18 | 32.659 | 561 |
| Maharashtra | 186.2 | 256.61 | 1378 |
| Tamil Nadu | 141.58 | 67.65 | 478 |
| Andhra Pradesh | 185.57 | 111.39 | 600 |
| Odisha | 183.319 | 93.895 | 513 |
| West Bengal | 11.36 | 12.96 | 1140 |
| Jharkhand | 14.83 | 5.83 | 393 |
| Chhattisgarh | 13.7 | 9.33 | 681 |
| Gujarat | 7.22 | 6.5 | 900 |
| Pondicherry | 5 | 2.16 | 432 |
| Assam | 1.05 | 1.08 | 1028 |
| Tripura | 4.25 | 3.45 | 812 |
| Meghalaya | 8.5 | 5.83 | 686 |
| Manipur | 0.9 | 0.324 | 360 |
| Nagaland | 0.5 | 0.54 | 1080 |
| **Total** | **1040.89** | **779.335** | **753** |

Source: Directorate of cashew nut and cocoa development.
(http://www.dgciskol.nic.in/)

Fig. 2.3 Various stages of cashew fruit growth

### 2.5.2 Importance of Cashew Tree

Cashew tree is very useful plant. Its tender leaves are sometimes used for treatment of diarrhea and piles. Old and dry leaves can be decomposed to compost and used as good fertiliser in agriculture. The stem can be used as wood. A grown up tree yields cashew fruits mainly during summer season. A cashew fruit consists of cashew apple and cashew nut shown in Figure 2.4. The cashew nut is separated from the fruit and the remaining cashew apple is a famous fruit with rich vitamin C. The cashew apple can be processed in different ways to have products like fruit juice, jelly, alcoholic drink, wine, pickle and syrup. The brown coloured hard nuts separated from the cashew apple are shown in Figure 2.5.

Fig. 2.4 Cashew apple with nut

Fig. 2.5 Cashew nut

Fig. 2.6 Split Cashew nut

The cashew nuts are split into two halves as shown in Figure 2.6 and inner cashew kernel is separated. These kernels are rich source of fats and proteins. Raw kernels or the roasted kernels are used as good quality food. From the kernel edible oil can also be extracted. From the remaining outer shell oil is separated which is called as cashew nut shell liquid (CNSL). About 30% to 35% oil is present in the cashew nut shell (Dinesha, P. and Mohanan, P., 2015). This oil is very useful in production of varnishes, brake liners and paints. The CNSL comprises 90% anacardic acid and 10% Cardanol

(Jagadish et al., 2012). When CNSL is exposed to distillation at low pressure (265 Pa) and about 235° C temperature anacardic acid is converted to cardanol (Mallikappa et al., 2012). The cardanol so obtained is a phenolic liquid which is used in production of medicines and chemical industries.

### 2.5.3 Availability of Cardanol

In India cashew cultivation has spread to 0.77 million hectares area with an average of 0.55 million metric tons of cashew nut production. The cashew nut shell oil constitutes about 30% weight of the nuts. The CNSL produced in India is exported to various countries. About 10938 metric ton of CNSL is exported during the year 2015, which contributes to Rs 55.81crores of revenue. The Table 2.2 shows the quantity of CNSL export during the year 2014- 2015.

Table 2.2 Export of cashew nut shell liquid from India

| | 2013-2014 | | 2014-2015 | |
|---|---|---|---|---|
| Countries | QTY (MT) | VALUE (Rs. in crores) | QTY (MT) | VALUE (Rs. in crores) |
| Korea Rep. | 1915 | 11.52 | 4695 | 28.60 |
| China | 2820 | 8.31 | 2721 | 9.21 |
| USA | 2987 | 8.17 | 1622 | 5.85 |
| Japan | 341 | 1.52 | 261 | 1.22 |
| Slovenia | 413 | 2.16 | 260 | 1.42 |
| Taiwan | 95 | 0.62 | 219 | 1.59 |
| Singapore | 221 | 1.71 | 187 | 1.51 |
| United Kingdom | 112 | 0.63 | 159 | 0.92 |
| Indonesia | 0 | 0.00 | 0 | 0.00 |
| Others | 576 | 3.97 | 814 | 5.49 |
| Total | 9480 | 38.61 | 10938 | 55.81 |

(Source: http://www.dgciskol.nic.in/)

### 2.5.4 Use of Cardanol as Diesel Engine Fuel

Cardanol and its blends with diesel were tested by many researchers as a fuel in compression ignition engine.

Mallikappa et al. (2011) have conducted a test on double cylinder 4S diesel engine with cardanol blends and found that brake thermal efficiency is less for the blends than diesel .NOx increased with increase in blending. HC and CO emissions are very high beyond 20% blend he concluded that only 20% blend can be used in engine.

Velmurugan et al. (2011) have used cardanol and diesel blend in a diesel engine .They found that B20 blend shown similar performance to that of diesel. At $19^{\circ}$BTDC injection timing and 22Mpa injection pressure they reported optimum performance.

Kasiraman et al. (2012) studied performance of CNSL with camphor oil blends in a four stroke diesel engine and obtained a brake thermal efficiency of 23.1%, 30.14% and 29.1% for CNSO, diesel and CMPRO30 respectively. Performance of CMPRO30 is very close to diesel. By blending camphor oil with CNSO emission were reduced.

Jagadish et al. (2012) carried out tests on a diesel engine with cardanol and methanol blend. Tested B20 with different blends of alcohol and found that with addition of alcohol brake thermal efficiency increased, it is more than diesel at 75% load. All emissions and opacity reduced by the additive. They also reported that Cardanol does not produce glycerol during esterification.

Mallikappa et al. (2012) studied the performance and emission characteristics of diesel engine operated with cardanol and concluded that up to 20% cardanol blend with diesel can be used in an engine beyond this HC and CO emissions increased.

Dinesha et al. (2014) operated a diesel engine by cardanol diesel blend with supply of extra oxygen to intake air and observed an increase in heat release rate with 7% addition of oxygen to B20M10 blend. There was an increase in BTE and NOx emission with increase in percentage of oxygen in the intake air, but CO, HC and smoke emissions were reduced with increase in enrichment of oxygen.

Radhakrrishnan et al. (2014) evaluated the performance of a diesel engine run with cardanol diesel blend and showed that B20 blend was performing very close to diesel.

They operated the engine with different injection timing and found that 19°BTDC was the optimum for that engine with Cardanol diesel blend as fuel.

Santhanakrishnan et al. (2015) have tested a four stroke diesel engine with CNSL blends and found that the maximum brake thermal efficiency was limited to 20.4% with cnsl and 30.8% with diesel. It was observed that SFC, exhaust gas temperature, CO, HC, NOx and smoke emissions were increased with increase in CNSL in the blend. The engine performed good with 20% blend than other blends.

Dinesha et al. (2015) tested cardanol in a CI engine with addition of oxygen to intake air and found that brake thermal efficiency increased with oxygen addition, a maximum of 33.98% efficiency was obtained for 7% enrichment. HC and CO emission reduced but NOx increased with oxygen enrichment.

## 2.6 STUDIES ON THE EFFECT OF ENGINE VARIABLES ON PERFORMANCE

The performance of any diesel engine varies with the variation in the operating variables like compression ratio, injection pressure, injection timing. Compression ratio is the ratio of the volume of combustion chamber of an engine at the beginning of compression stroke to the volume of combustion chamber at the end of compression. By changing the compression ratio the pressure and temperature inside the engine combustion chamber alters, which affects the combustion process. The pressure at which the fuel is injected into the combustion chamber is known as injection pressure. The fuel spray characteristics, which decides the quality of combustion, depends on the injection pressure. Injection timing (IT) is the crank angle position with respect to top dead centre at which fuel is injected in to the combustion chamber. With the variation of IT the speed of combustion changes due to variations in pressure and temperature of combustion chamber at the time of fuel injection. Many scientists have studied the performance of diesel engine operated with various biofuels and its blends with diesel fuel by altering the operating variables.

Raheman and Ghadge (2008) have experimented on Ricardo E6 engine operated with mahua oil biodiesel and its blends by varying the compression ratio and injection timing. They reported that the brake thermal efficiency increased by 33% when the compression ratio was increased from 18:1 to 20:1 and BSFC reduced by 11%. The exhaust gas temperature decreased with increase in compression ratio and injection timing.

Jindal et al. (2010) tested jatropha biodiesel on diesel engine with direct fuel injection system at different CR and IP. The test results shown that performance in terms of BTE and BSFC improved when the CR and IP were increased. At higher IP emissions of NOx, HC and smoke were reduced, whereas CO emission and EGT increased. When the CR was increased the HC and EGT were increased whereas the smoke and CO emission were reduced. But NOx emission was unaltered at higher IP

Ganapathy et al. (2011) have studied the impact of injection timing on the performance and emission of a diesel engine operated with jatropha biodiesel. From the results they resolved that the CO, smoke, HC and BSFC were reduced by 2.5%, 8.5%, 1.2% and 5% with the advancement of injection timing, but NOx increased. The trend was reversed when injection timing was retarded.

Gumus et al. (2011) investigated the effect of IP on exhaust emission on CI engine operated by diesel biodiesel blend. The investigations revealed that when IP was increased there was a reduction in the emissions of smoke, CO and HC, whereas the NOx, $CO_2$ and $O_2$ emissions increased.

Muralidharan K. and Vasudevan D (2011) have tested waste cooking oil methyl ester and its blends with diesel in a single cylinder four stroke diesel engine. They reported that 40% blend (B40) performance was good at 21:1 compression ratio with reduction of carbon monoxide, unburned hydrocarbon and increase in NOx.

Kannan and Anand (2012) studied performance of CI engine with direct injection using waste cooking oil biodiesel at various Injection timing (IT) and injection pressure (IP). They observed an enhancement in BTE and decrease in emissions of NOx and smoke, when the IP was increased to 280 bar and IT was advanced to 25.5° BTDC.

Hirkude J. and Padalkar A.S. (2014) have investigated experimentally the effect of compression ratio on performance of compression ignition engine operated with waste fried oil methyl ester (WFOME) and blends with diesel. The results revealed that with increase in compression ratio the BTE increased with minimum BSFC and emission of CO and smoke decreased. The EGT was increased with increase in compression ratio.

Ramalingam et al. (2015) explored the influence of IT and compression ratio (CR) on emissions and performance of a CI engine operated with annona methyl ester. From the experimental results they notified a decrease in CO and HC emission, with improved BTE, when there is an advancement in IT from 27° BTDC to 30° BTDC and CR was increased from 17.5:1 to 19.5:1.

Yadav et al. (2017) investigated the effect of compression ratio on performance of a diesel engine run with kaner seed biodiesel. They found that when compression ratio was increased from 16:1 to 18:1 the performance of the engine increased. Brake specific fuel consumption for the biodiesel blend was more than diesel fuel even though the brake thermal efficiency was more than diesel fuel operation.

## 2.7 OPTIMISATION OF PARAMETERS

Performance and emissions of engine depends on parameters like injection pressure, compression ratio and injection pressure. Using statistical tools the optimum levels of parameters were obtained to operate the engine optimum condition.

Lee et al. (2013) investigated the emissions of a common rail direct injection compression ignition engine. Using the taguchi method of optimisation the parameters were optimised and the residual of efficiency were plotted. They reported that the scattering of the plot without any particular pattern concludes the randomness of error and suitability of the model with the data.

Kaliamoorthy and Paramasivam (2013) studied the performance of a diesel engine operated with karanja biodiesel. The operating parameters for high brake power and lower emission were obtained using taguchi method. The parameters having high signal to noise ratio were selected as the optimum level. From the analysis they resolved the optimum levels as 230 bar injection pressure, 27°BTDC injection timing, 17:1 compression ratio at 70% load for 20% biodiesel blend.

Wen Wu and Yi Wu (2013) have analysed the performance of a diesel engine and the operating parameters for lower emission were obtained using taguchi method. The parameters with high signal to noise ratio were chosen as optimal level. They resolved that high BTE and lower BSFC were recorded for 30% hydrogen, 40% EGR for 20% blend.

Yi Wu et al. (2014) have evaluated the emissions of a diesel engine operated with biodiesel, diesel and LPG as fuel and optimised the operating condition using taguchi method. The highest signal to noise ratio was obtained for the combination of 40% LPG, 10% biodiesel and 20% EGR, for which minimum emissions were confirmed by the confirmation test.

Balki et al. (2016) applied the taguchi method to optimise the compression ratio, engine speed and injection timing for higher performance and minimum emissions of the engine operated with methanol, ethanol and gasoline blends. They have considered the parameters with highest signal to noise ratio as the optimum. The optimum compression ratio and speed were 9:1 and 2400 rpm and the injection timing was 20°BTDC for alcohol and 26°BTDC for petrol.

From the detailed review of the earlier research in biofuel, it was observed that with the use of straight vegetable oil as diesel engine fuel there was problems of cold flow due to higher viscosity. With the use of biodiesel as an alternate to diesel fuel in compression ignition engine, the performance and emission results were varying depending upon the feed stock of the biodiesel. By blending biodiesel with kerosene the viscosity reduces and there was improvement in performance and emissions. Since Cardanol does not have glycerine esterification is not necessary. As observed in the literature not much work was reported on Cardanol. So a new blend of biofuel was prepared by blending Cardanol with kerosene, which was not reported till now in the literature.

# CHAPTER 3

# EXPERIMENTAL SET-UP AND PROCEDURE

## 3.1 INTRODUCTION

In this chapter, the details about experimental set up used for performance and emission analysis of diesel engine operated by Cardanol kerosene blends are discussed. The necessary instrumentations and measurement systems connected with the engine set up were elaborated. The instruments required for determining the test fuel properties were explained in the following section.

## 3.2 EXPERIMENTAL SET UP

The block diagram of the experimental setup is shown in Figure 3.1 and the corresponding photographic view depicted in Figure 3.2. The experimental investigations are conducted in a four stroke single cylinder naturally aspirated variable compression ratio engine. The technical specification of the engine are provided in Table 3.1. The engine is connected with an eddy current dynamometer to apply load on the engine. A computer loaded with 'ICEngineSoft' software package is interfaced with the engine for analysing the performance results online. The panel box of the setup has an air box, dual fuel tank, fuel measuring unit, load indicator and rotameters for water flow measurement. Major elements present in the experimental setup are explained in the succeeding paragraphs.

### 3.2.1 Fuel Flow Measurement

The flow rate of fuel to engine is measured by measuring the time for known volume of fuel consumption. By operating the 3-way valve on the panel box, the fuel is made to flow through a calibrated burette to the engine. A differential pressure transmitter (Make Yokogawa, model EJA110-EMS-5A- 92NN) is used to measure the flow rate by sensing the hydrostatic head change in the burette due to fuel consumption.

Table 3.1 Technical specifications of the engine

| | |
|---|---|
| Number of cylinders | 1 |
| Bore | 87.5 mm |
| Number of strokes | 4 |
| Connecting rod length | 234mm |
| Power | 3.5 kW |
| Compression ratio | 12 to 18:1 |
| Stroke | 110 mm |
| Rated speed | 1500 RPM |
| Dynamometer arm length | 184 mm |

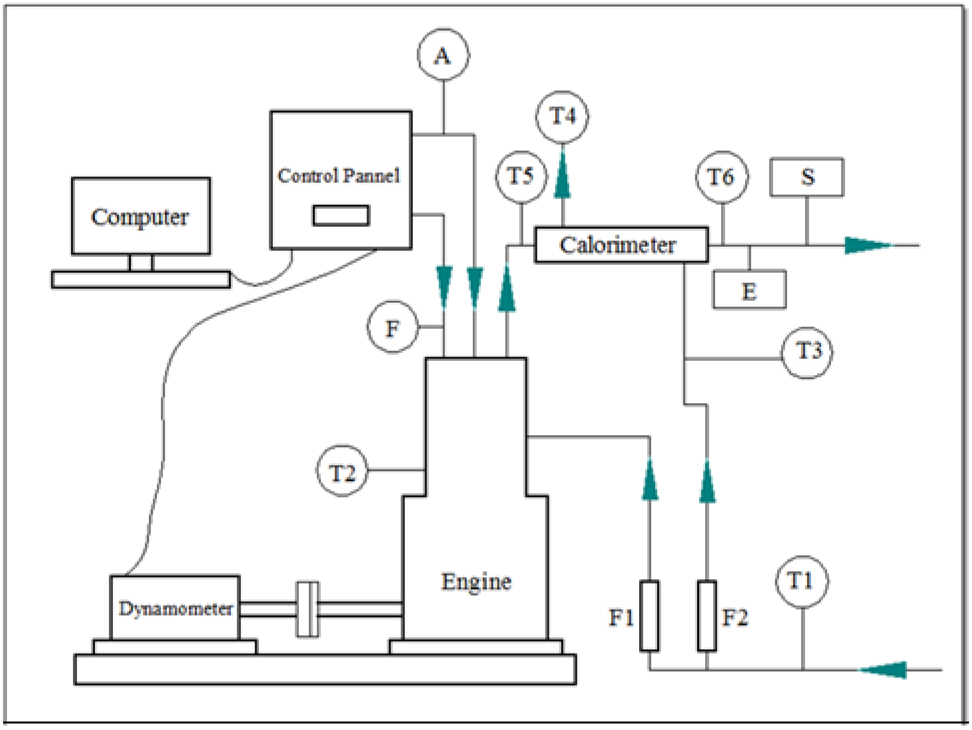


Fig. 3.1 Block diagram of experimental set up

A-Measurement of air flow

F1-Engine cooling water measurement

F-Fuel flow rate measurement

F2- Calorimeter water flow measurement

S- Smoke meter

E-Analyser of exhaust gas

T3&T4-Inlet and outlet calorimeter water temperature

T5&T6-Temperature of exhaust gas at calorimeter inlet and outlet

Fig. 3.2 Photograph of the experimental set up

### 3.2.2 Air Flow Measurement

The air flow to the engine is measured by connecting a large air box with 20mm diameter orifice to the intake manifold. The box is connected to reduce the fluctuations in the flow. A differential air flow transmitter fitted across the orifice measures the pressure difference, which is proportional to the air flow rate.

### 3.2.3 Temperature Measurement

The temperature of engine cooling water is measured with Radix make (Pt 100) resistance thermometer having range of 0- 100°C. The exhaust gas temperature is measured with K-type thermo couple (Wika make) having 0- 1200°C range.

### 3.2.4 Load Measurement

The engine was loaded by an eddy current dynamometer shown in Figure 3.3. The eddy current dynamometer consists of a set of electromagnets in the stator and a rotor disc coupled to the engine. Due to the rotation of rotor an eddy current is induced in the stator which opposes the rotation of rotor and thus the engine was loaded. The eddy current dissipates as heat while loading the engine. Therefore the eddy current dynamometer is supplied with cooling water. A strain gauge type load cell was mounted to measure the applied load. This eddy current dynamometer is capable of producing maximum of 7.5 kW at 3000 rpm.

Fig. 3.3 Eddy current dynamometer

### 3.2.5 Water Flow Measurement

Flow rate of cooling water through engine and calorimeter is measured with rotameter. The engine cooling water inlet hose is connected with a rotameter having range of 40 – 400 lph (Make Eureka, Model PG-6). The calorimeter cooling water inlet is connected with a rotameter having range of 25- 250 lph (Make Eureka, Model PG-5).

### 3.2.6 Exhaust Emission Measurement

The exhaust emissions of carbon monoxide (CO), unburned hydrocarbon (HC) and oxides of nitrogen (NOx) are measured using an exhaust gas analyser (Netel exhaust gas analyser, Model: NPM-MGA-1). The pictorial view of exhaust gas analyser is shown in Figure 3.4. The technical specifications of the analyser are given in Appendix I.


Fig. 3.4 Exhaust gas analyser

### 3.2.7 Smoke Measurement

Smoke is the visible products of combustion in the exhaust gas due to poor combustion. It is originated early in the combustion process as a partial combustion product called as soot. Normally soot is consumed during later part of combustion. However, if the soot does not find sufficient oxygen to burn, it is exhausted and if in sufficient quantity it becomes visible and called as smoke. Continuous exposure to smoke leads to lungs diseases and eye problems in humanbeing. The smoke level is quantified by the level of darkening of a filter paper exposed to it or extinction of light passed through the smoke. Emission of smoke is measured using smoke meter (Netel smoke meter model, NPM-SM-111B) which works on light extinction principle. A pictorial view of smoke meter is shown in Figure 3.5 and the specifications in Appendix I.


Fig. 3.5 Smoke meter

## 3.3 DETERMINATION OF FUEL PROPERTIES

The properties like viscosity, flash point, fire point, calorific value and density of all fuels and fuel blends used in the tests are determined at the laboratory. Initially the purity of Cardanol used for the test is checked at a nearby Cardanol industry. The instrumentations used for testing the properties are discussed in the following section.

### 3.3.1 Measurement of Viscosity

The resistance offered by any fluid flow to flow is the measure of its viscosity. Viscosity of biofuel plays an important role in fuel atomisation and fuel injection. As per ASTM Standard D6751 the maximum viscosity of biofuel is limited to 6 cSt. Viscosity is determined in a Cannon Fenske viscometer.


Fig. 3.6 Line diagram of Cannon Fenske viscometer


Fig. 3.7 View of Cannon Fenske viscometer

Figure 3.6 and Figure 3.7 represents the line diagram and the pictorial view of Cannon Fenske viscometer. The set up consists of a standard viscometer tube, water bath to heat the liquid in the tube and a stop watch.

### 3.3.2 Measurement of Flash Point

Flash point of a liquid fuel is the lowest temperature at which it gives sufficient vapours that can burn in contact with a flame. As per ASTM Standard D6751 for biodiesels the minimum flash point is 130°C. The flash point of any fuel must be high in order to reduce the chances of fire hazard. Flash point is determined in Pensky- Marten Closed cup tester shown in Figure 3.8. The apparatus consists of a heating device, test cup, cup cover with shutter and ignition source.

Fig.3.8 Pensky- Marten Closed cup tester

### 3.3.3 Measurement of Fire Point

Fire point of a fuel is the lowest temperature at which the fuel starts burning continuously even after the ignition source is removed. Fire point is found using a Cleveland open cup tester. Figure 3.9 depicts a Cleveland open cup tester. It consists of a standard cup, heater, thermometer and a heating source.

Fig. 3.9 Clevelands open cup tester

### 3.3.4 Measurement of Calorific Value

Calorific value of a fuel is the energy released by complete combustion of unit quantity of fuel in presence of oxygen. The calorific value of solid and liquid fuels are determined in a bomb calorimeter. Figure 3.10 and Figure 3.11 shows the images of bomb calorimeter. A bomb calorimeter consists of an inner calorimeter vessel, bomb firing unit, stirrer, timer, pellet press and oxygen cylinder with a pressure gauge. Initially the water equivalent of calorimeter is found by using benzoic acid as a fuel. The calorific value of benzoic acid is 6319 Cal/gram. Then unit quantity of the fuel to be tested is filled in the crucible and tested.


Fig. 3.10 Outer box of bomb calorimeter    Fig. 3.11 Inner vessel of bomb calorimeter

### 3.3.5 Measurement of Density

Density of liquids are determined by a hydrometer. Figure 3.12 shows the hydrometer dipped in the standard jar filled with Cardanol and Figure 3.13 shows the line diagram of the hydrometer.


Fig. 3.12 Hydrometer    Fig. 3.13 Line diagram of hydrometer

## 3.4 EXPERIMENTAL METHODOLOGY

The initial settings of the engine are checked and corrected as per the manufacturer specifications. The water flow rate through the engine (250 lph) and calorimeter (100 lph) are set to the catalogue values. The engine dual fuel tank was filled with diesel and a blend. The engine is started with diesel fuel by hand cranking and allowed to reach steady state by running with the same parameters for 15 to 20 minutes. After that load is applied by the dynamometer and the fuel consumption rate was recorded. The exhaust emissions (HC, CO and NOx) are measured using exhaust gas analyser (Netel exhaust gas analyser, Model: NPM-MGA-1) and the smoke opacity is measured by using smoke meter (Netel smoke meter model, NPM-SM-111B) by inserting the sensor probe of the exhaust gas analyser and smoke meter one after the other. Then the engine is operated with the blend filled in the tank by changing the position of the valve fitted in the dual fuel tank and same procedure is repeated. While changing the blend in the tank, the fuel tank was fully drained and rinsed with new blend and filled with the new blend. For every trial, readings are recorded only after engine reaching the steady state. For all the test fuels brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature readings are noted from the computer.

The performance of an engine is evaluated by finding its brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature.

### 3.4.1 Brake Thermal Efficiency

Brake thermal efficiency (BTE) of an engine is the ratio of the brake power to the energy input to the engine by fuel.

BTE= brake power / mass of fuel × calorific value of fuel

Brake power (BP) is the useful power obtainable at the crank shaft of any engine. The brake power is measured with various types of dynamometers connected to the crank shaft. In this experiment eddy current dynamometer is used to measure the brake power.

### 3.4.2 Brake Specific Fuel Consumption

The fuel consumption characteristics of any engine is expressed as specific fuel consumption in kilograms of fuel per kilowatt-hour. Brake specific fuel consumption (BSFC) is the quantity of fuel consumed to produce unit brake power at unit time.

BSFC = fuel consumption per unit time / brake power

### 3.5 EXPERIMENTAL ERROR ANALYSIS

Uncertainty in the experimentations may be due to error in the instrument, environmental conditions, observations and reading. The percentage uncertainty of measured parameters is obtained by repeating measurement of each parameter five times and finding the variations. Table 3.2 shows the uncertainty of all the measured parameters.

Table 3.2 Uncertainties of measured parameters

| S.No. | Parameter | Resolution | Percentage Uncertainty |
|---|---|---|---|
| 1 | HC | 1 ppm | ± 0.2 |
| 2 | CO | 0.01% | ± 0.3 |
| 3 | NOx | 1 ppm | ± 0.2 |
| 4 | Smoke | 0.1% | ± 1 |
| 5 | Load | 0.1N | ± 0.5 |
| 6 | Fuel measurement | 0.1cc | ± 1 |

# CHAPTER 4

# PERFORMANCE AND EMISSION ANALYSIS OF CARDANOL KEROSENE BLENDS IN DIESEL ENGINE

## 4.1 INTRODUCTION

The Cardanol is blended with kerosene in various proportions and the blends were prepared. The performance and emission characteristics of a four stroke single cylinder 3.5kW water cooled diesel engine were analysed with the prepared blends of fuel. The tests were carried out at various loads to investigate performance parameters like brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature and also the emissions of carbon monoxide, unburned hydrocarbon, oxides of nitrogen and smoke. The purity of the Cardanol used in the test were tested before blending with kerosene. Various properties like viscosity, flash point, fire point calorific value and density of all the prepared blends were determined.

## 4.2 TESTING THE PROPERTIES OF FUEL BLEND

As per ASTM (D6751) standards the test blends were tested to determine the required properties. The following sections gives the details about the testing procedures.

### 4.2.1 Testing Purity of Cardanol

Cardanol used for the test was purchased from local cashew industry. The purity of Cardanol was tested in a gas chromatography (G.C). Figure 4.1 shows the graphical representation of the G.C test results and Table 4.1 gives values of the area under various retention time. Cardanol is identified between retention times RT 7.0 to RT7.7. The percentage area under these RT (RT 7.0 to RT7.7.) in the graph will give the percentage of Cardanol present in the sample. The total area between RT7.0 to RT 7.7 was calculated by adding the individual areas between these retention times from Table 4.1 as shown below.

88.82248+0.97342+0.53841+0.06533+0.22329+0.03445+0.04349 =90.6 %

The test results indicates that the Cardanol sample is 90.6% pure.

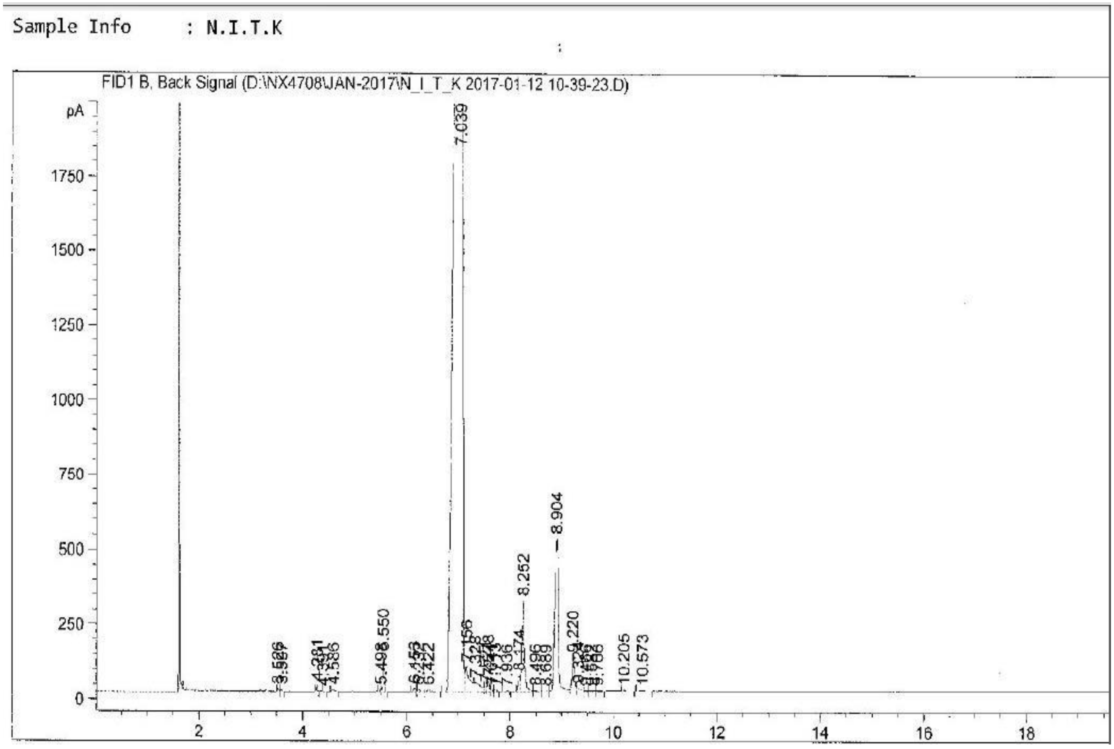


Fig. 4.1 Graphical representation of G.C results

### 4.2.2 Preparation of Test Fuel Blends

The test fuel blends were prepared by mixing Cardanol and kerosene on a volume basis. Diesel and kerosene were purchased from local outlets. Different blends of Cardanol and kerosene, such as BK10 (10% kerosene and 90% Cardanol), BK20% (20% kerosene and 80% Cardanol), BK30 (30% kerosene and 70% Cardanol), BK40% (40% kerosene and 60% Cardanol) were prepared. The blends were checked for miscibility by keeping it for two days. After two day no separation or no layers formation were noticed. This indicates that the two liquids are miscible. Properties like kinematic viscosity, density, flash point, fire point and calorific value of diesel, kerosene, Cardanol and kerosene cardanol blends were determined as per ASTM Standards.

Table 4.1 Area under various retention times.

| Peak # | RetTime [min] | Type | Width [min] | Area [pA*s] | Height [pA] | Area % |
|---|---|---|---|---|---|---|
| 5 | 4.586 | VV | 0.0460 | 22.06513 | 6.86016 | 0.04614 |
| 6 | 5.498 | VV | 0.0242 | 12.36455 | 7.69691 | 0.02586 |
| 7 | 5.550 | VV | 0.0336 | 261.42709 | 121.34418 | 0.54672 |
| 8 | 6.153 | VV | 0.0497 | 37.05583 | 11.31520 | 0.07749 |
| 9 | 6.232 | VB | 0.0494 | 25.44327 | 7.54332 | 0.05321 |
| 10 | 6.422 | BB | 0.0596 | 31.08635 | 7.38400 | 0.06501 |
| 11 | 7.039 | BV | 0.1203 | 4.24729e4 | 4472.68799 | 88.82248 |
| 12 | 7.156 | VV | 0.0718 | 465.46622 | 83.17000 | 0.97342 |
| 13 | 7.328 | VV | 0.1037 | 257.45404 | 30.73548 | 0.53841 |
| 14 | 7.521 | VV | 0.0294 | 31.23810 | 15.22530 | 0.06533 |
| 15 | 7.578 | VV | 0.0465 | 106.77207 | 33.62197 | 0.22329 |
| 16 | 7.641 | VV | 0.0383 | 16.47398 | 6.14915 | 0.03445 |
| 17 | 7.713 | VV | 0.0497 | 20.79698 | 5.82247 | 0.04349 |
| 18 | 7.936 | VV | 0.0490 | 8.23624 | 2.37396 | 0.01722 |
| 19 | 8.174 | VV | 0.0412 | 138.71779 | 52.71337 | 0.29010 |
| 20 | 8.252 | VV | 0.0479 | 1019.47949 | 305.79800 | 2.13201 |
| 21 | 8.496 | VV | 0.0552 | 12.30939 | 3.11077 | 0.02574 |
| 22 | 8.689 | VV | 0.0552 | 5.72913 | 1.40002 | 0.01198 |
| 23 | 8.904 | VV | 0.0607 | 2302.56982 | 510.05081 | 4.81531 |
| 24 | 9.220 | VV | 0.0512 | 400.89325 | 113.62529 | 0.83838 |
| 25 | 9.324 | VV | 0.0547 | 39.54211 | 9.87583 | 0.08269 |
| 26 | 9.466 | VV | 0.0560 | 10.60743 | 2.33351 | 0.02218 |
| 27 | 9.582 | VV | 0.0690 | 11.66838 | 2.09582 | 0.02440 |
| 28 | 9.706 | VV | 0.0664 | 5.62829 | 1.14434 | 0.01177 |
| 29 | 10.205 | VB | 0.0555 | 15.78759 | 4.18662 | 0.03302 |
| 30 | 10.573 | BB | 0.0845 | 16.99665 | 2.69771 | 0.03554 |
| Totals : | | | | 4.78177e4 | 5860.18491 | |

### 4.2.3 Determination of Viscosity

Kinematic viscosity was determined using a Cannon-Fenske Viscometer as per ASTM Standard D445. The viscometer tube (Tube No.100/16) was filled with the biofuel blend up to the top mark with help of a rubber bulb and was fixed inside the viscometer-water bath apparatus. The water in the water bath was heated up to 40$^{\circ}$C and maintained for 20 to 30 minutes, so that the biofuel inside the tube gets heated up to that temperature. After 30 minutes the viscometer tube was opened and simultaneously the stop watch was started .Once the flow of biofuel reached the bottom mark in the viscometer tube the stop watch was stopped and time required for flow was noted in seconds. By multiplying the time (in seconds) and the viscometer calibration constant (0.0212 for tube No. 100/16) the viscosity of the biofuel was obtained. Table 4.2 gives the kinematic viscosities of test fuels.

Table 4.2 Kinematic viscosity of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Kinematic viscosity @40˚C (in cSt) | D445 | 3.17 | 1.18 | 19.4 | 14.2 | 9.3 | 5.9 | 3.8 |

#### 4.2.4 Determination of Flash Point

The flash point was determined using a Pensky-Martens closed cup apparatus as per ASTM Standard D93. The test fuel was filled in the standard brass cup up to the marking. The cup was closed by the cover and then it was placed inside the apparatus. A thermometer was inserted in to the holder in the apparatus such that the bulb of the thermometer was dipped in the test fuel. The heater was switched on and the test flame was lighted. When the temperature of the test fuel was increasing, the mechanism on the cover was operated, so that the lid opens and the test flame was brought down. The procedure was repeated for every degree rise in temperature until a flash in the cup was observed. The minimum temperature at which a flash takes place was recorded, that gives the flash point of the test fuel. The obtained flash points of test fuels were given in Table 4.3.

Table 4.3 Flash point of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Flash point (in ˚C) | D93 | 51 | 42 | 218 | 112 | 82 | 65 | 59 |

#### 4.2.5 Determination of Fire Point

The fire point was determined using Cleveland Open Cup Apparatus as per ASTM Standard D92. The test fuel was filled in the test cup up to the specified marking. A thermometer was inserted in to the support such that the bulb of the thermometer was dipped in the test fuel. The temperature of the test fuel was increased by heating in the heater plate. The test flame was passed across the cup at regular intervals and the temperature was noted. This process was repeated until the test fuel ignites and sustain burning for minimum of five seconds. The temperature at this point was noted, which gives the fire point of the test fuel. The fire points of the test fuels as obtained is given in Table 4.4.

Table 4.4 Fire point of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Fire point (in °C) | D93 | 57 | 47 | 232 | 136 | 98 | 76 | 68 |

### 4.2.6 Determination of Calorific Value

The calorific value of fuel blend was determined using a bomb calorimeter as per ASTM Standard D240. Known weight of fuel was taken in the crucible. A nichrome wire of known length (7.5cm) was connected between the terminals inside the bomb and known length (8cm) of thread was kept in the crucible. The bomb vessel was closed and filled with oxygen up to specified pressure (25kg/cm$^2$). After that the bomb vessel was placed inside the water jacket and the electrical connections were connected and checked for continuity. The water jacket was filled with two litres of water. The stirrer was started and the initial temperature of jacket water was noted. The bomb was fired by pressing the fire button. The temperature of jacket water started increasing and the final temperature of jacket water was noted after steady state. By multiplying rise in temperature to water equivalent of calorimeter (2936 cal/°C) the calorific value of the fuel blend was determined. The calorific value of test fuels were shown in Table 4.5.

Table 4.5 Calorific value of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Calorific value (in kJ/kg) | D240 | 43580 | 44230 | 40246 | 40598 | 40960 | 41331 | 41712 |

### 4.2.7 Determination of Density

The density of biofuel blends was determined using a hydrometer as per ASTM Standard D1298. The fuel blend was filled in the standard measuring flask up to 500ml. The hydrometer was slowly lowered in to the flask until it floats freely. The point which touches the surface of the hydrometer stem was noted, which is the density of the blend. The density of all the test fuels were tabulated in Table 4.6.

Table 4.6 Density of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Density @ 15°C (in $kg/m^3$) | D1298 | 821 | 780 | 903 | 846 | 834 | 825 | 811 |

Various properties of diesel fuel used in the test, kerosene, Cardanol and the biofuel blends were determined in the laboratory as mentioned above. The results of the tests were tabulated in table 4.7.

## 4.3 METHODOLOGY OF PERFORMANCE AND EMISSION TEST

The experimental work was carried out in a single cylinder, water cooled, four stroke variable compression ratio diesel engine. The engine was loaded by a water cooled eddy current dynamometer which is directly coupled to the crankshaft. The dynamometer is capable of producing 7.5kW and rated at a maximum speed of 3000rpm. The engine is connected to a computer for capturing the data. The initial settings of the engine were checked and corrected as per the manufacturer specifications. The water flow rate through the engine and calorimeter were set to the catalogue values. The engine dual fuel tank was filled with diesel and BK10 blend separately. The engine was started and allowed to reach steady state by running with the same parameters for 15 to 20 minutes. After that stepwise load was applied by the dynamometer and the fuel consumption rate was recorded. The exhaust emissions (HC, CO and NOx) was measured using exhaust gas analyser (Netel exhaust gas analyser, Model: NPM-MGA-1) and the smoke opacity was measured by using smoke meter (Netel smoke meter model, NPM-SM-111B). The trail was repeated atv3kg, 6kg, 9kg and 12kg load. The experiments were repeated for BK20, BK30 and BK40 blends. For all the trails brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature readings were noted. Appendix –II shows the experimental results of all the blends.

Table 4.7 Properties of diesel, kerosene, Cardanol and blends

| Properties | ASTM code | Diesel | Kerosene | Cardanol | BK10 | BK20 | BK30 | BK40 |
|---|---|---|---|---|---|---|---|---|
| Kinematic viscosity @40˚C (in cSt) | D445 | 3.17 | 1.18 | 19.4 | 14.2 | 9.3 | 5.9 | 3.8 |
| Density @ 15˚C (in $kg/m^3$) | D1298 | 821 | 780 | 903 | 846 | 834 | 825 | 811 |
| Flash point (in ˚C) | D93 | 51 | 42 | 218 | 112 | 82 | 65 | 59 |
| Fire point (in ˚C) | D93 | 57 | 47 | 232 | 136 | 98 | 76 | 68 |
| Calorific value (in kJ/kg) | D240 | 43580 | 44230 | 40246 | 40598 | 40960 | 41331 | 41712 |

### .4.3.1 Performance Results

The performance of any diesel engine is determined by finding the brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature. The information about the conversion of chemical energy of fuel into heat energy is indicated by the brake thermal efficiency and brake specific fuel consumption. The exhaust gas temperature reveals the energy lost during the combustion process.

#### 4.3.1.1 Brake thermal efficiency

Brake thermal efficiency (BTE) is the ratio of brake power to the heat energy supplied by fuel. Figure 4.2 shows the variations of BTE with load for diesel and different blends of kerosene and Cardanol. It was observed that BTE increases with load for all the fuels. This is due to reduction in heat loss and also due to reduction in friction to brake power ratio at higher loads. The BTE increases with increase in kerosene percentage in the blend up to 30%.This is because of better atomisation and higher volatility of kerosene. Since kerosene is a dry fuel (lower lubricity than diesel) for BK40 blend, the BTE reduces because of increased frictional loss at the pump. At full load BTE for diesel fuel was 28.72% and for BK30 it was 28.3%. Among all the blends, higher BTE was obtained for BK30. However, this is 1.5% less than for diesel which may be due to reduction in calorific value of the BK30 blend compared to diesel.

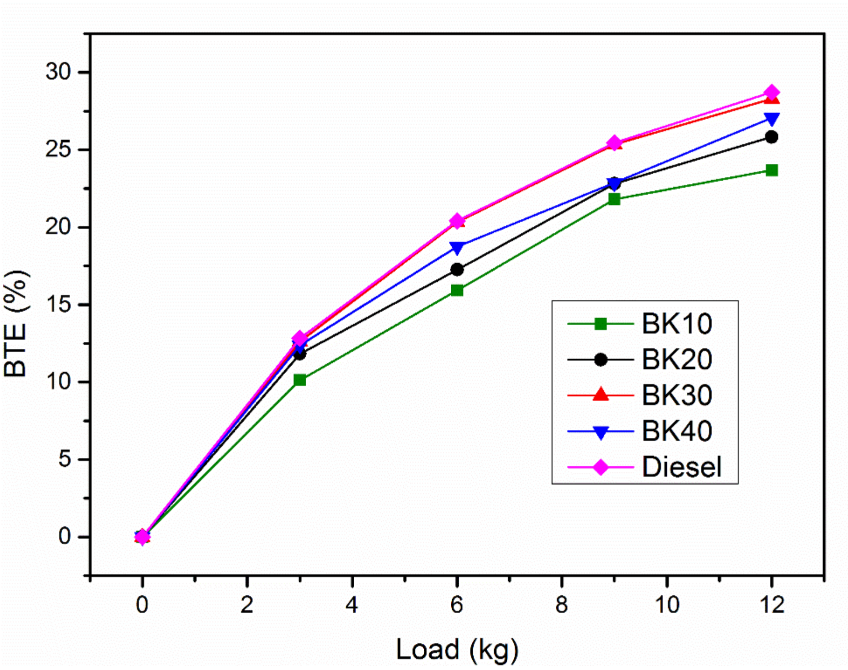


Fig. 4.2 Variation of brake thermal efficiency with load

### 4.3.1.2 Brake specific fuel consumption

Brake specific fuel consumption (BSFC) is the amount of fuel burned in the engine to produce unit power in unit time. Figure 4.3 shows variation of BSFC with load for different blends and diesel. As shown in the figure, biofuel blends reports a higher BSFC than that of diesel. This may be due to higher density and lower calorific value of biofuel blends. There was a reduction in BSFC as the percentage of kerosene increases. Similar findings were reported by Vedharaj et al. (2015) for Cardanol blends with diesel, in which BSFC was higher for the blends compared to diesel.

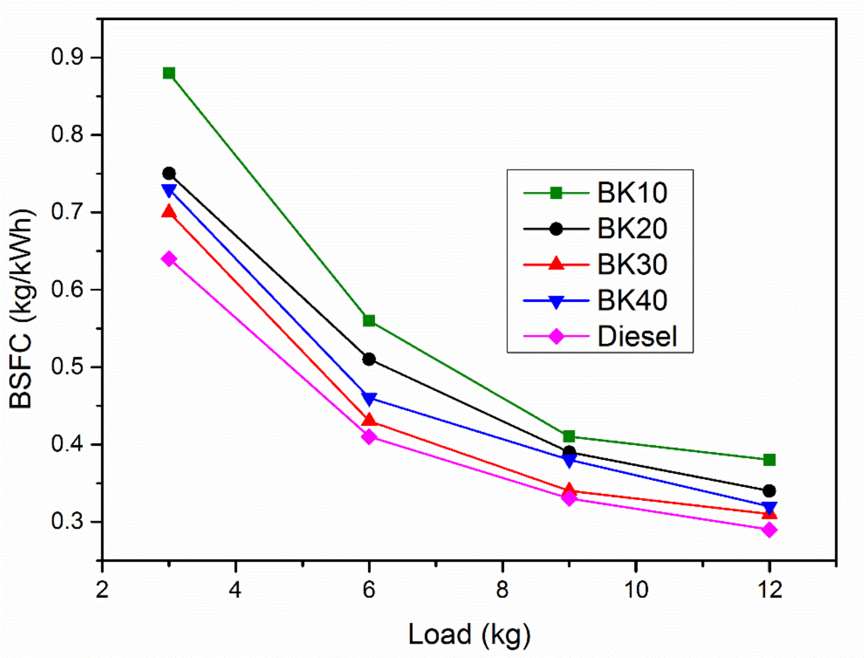


Fig. 4.3 Variation of brake specific fuel consumption with load

#### 4.3.1.3 Exhaust gas temperature

The exhaust gas temperature variations for different test fuels at various loads are shown in Figure 4.4. Very high exhaust gas temperature (i.e. 297.42°C) was observed for BK10 blend. This is because of high viscosity and poor volatility of blend, which leads to incomplete combustion and extension of combustion to exhaust stroke. However, as a result of increased volatility and atomisation of blend for BK20 and BK30 blends, the exhaust gas temperature was reduced(i.e. 281.49°C and 274.96°C respectively). This is mainly due to improvement in combustion. For BK40 blend the exhaust gas temperature was again increased. For diesel fuel the exhaust gas temperature was 269.19 °C, which is lower than the test blends.

### 4.3.2 Emission Results

The products of combustion of any diesel fuel consists of carbon monoxide, unburned hydrocarbon, oxides of nitrogen and smoke. To check quality of combustion of the fuel and to find the extent of pollutants formed during combustion process, the exhaust gas was analysed to quantify various constituents.

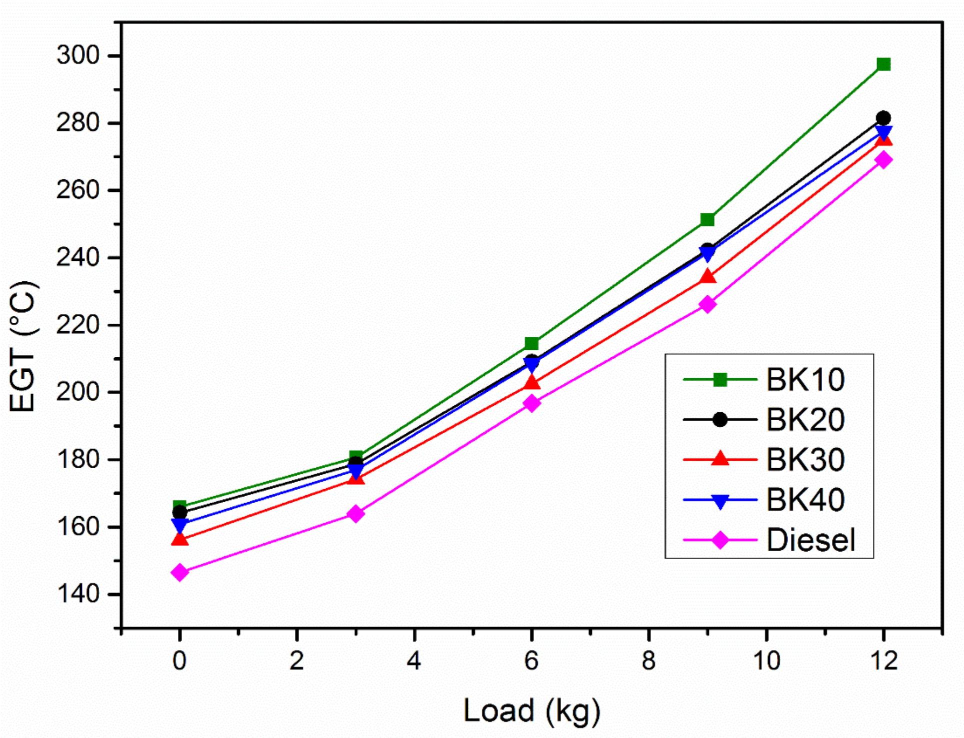


Fig. 4.4 Variation of exhaust gas temperature with load

#### 4.3.2.1 Carbon monoxide

Carbon monoxide is produced due to incomplete combustion owing to deficiency of oxygen in the combustion chamber. Figure 4.5 shows the variation of CO emission with load. For diesel fuel CO emissions was increasing with increase in load. This is because of more fuel injection at higher loads (Natesan, 2013). For kerosene blends CO emission was reduced with increase in load and increase in percentage of kerosene. This is mainly due to complete combustion of blends due to higher volatility of kerosene. At full load for BK10, BK20, BK30 and BK40 blends CO emissions were respectively 37.8%, 41.5%, 42.6% and 45.1%, which is lower than that of diesel fuel.

#### 4.3.2.2 Unburned hydrocarbon

Figure 4.6 shows unburned hydrocarbon emissions at various loads. For all the test fuels HC emissions increased with increase in load. Similarly, though at lower loads the HC emission was low for diesel, as the load increases emission were also increasing. At higher loads HC emissions were reduced with increase in percentage of kerosene in the blends. At lower loads the HC emission for diesel is low compared to the test blends (i.e. BK10, BK20, BK30 and BK40). Due to higher viscosity and incomplete combustion at full load for BK10 blend emission was 5% more than the diesel fuel. For BK20, BK30 and BK40 blends HC emission was respectively1.69%, 5.08% and 11.8% less than diesel. This is because of higher volatility and fast burning of kerosene.

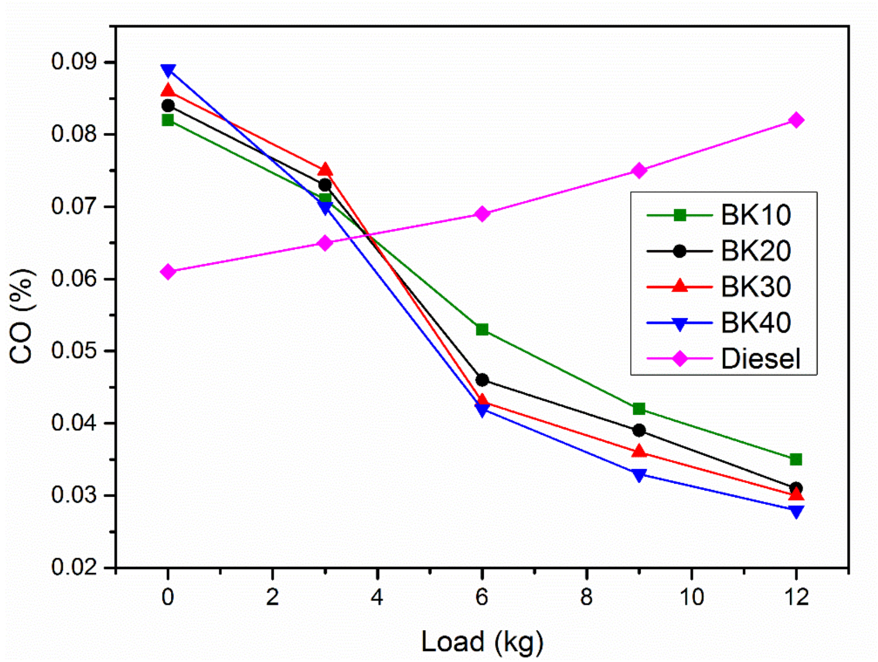


Fig. 4.5 Variation of carbon mono oxide with load

#### 4.3.2.3 Oxides of nitrogen (NOx)

Oxides of nitrogen formed during combustion due to presence of higher concentration of oxygen in combustion chamber and high combustion temperature. Figure 4.7 shows the variation of NOx emissions at different loads for all the test fuels. From the graph it is observed that for all the test fuels NOx emissions are increasing with increase in load. Similar variations were also described by Godigunar et al. (2009). Compared to diesel fuel the emissions of NOx emission increased by 30.8%, 24.4%, 1.9% and 2.6% for blends BK10, BK20, BK30 and BK40, respectively. This increase in NOx emission is mainly due to presence of inbuilt oxygen in the biofuel and also slow combustion of biofuel, which leads to increase in exhaust gas temperature and NOx formation.

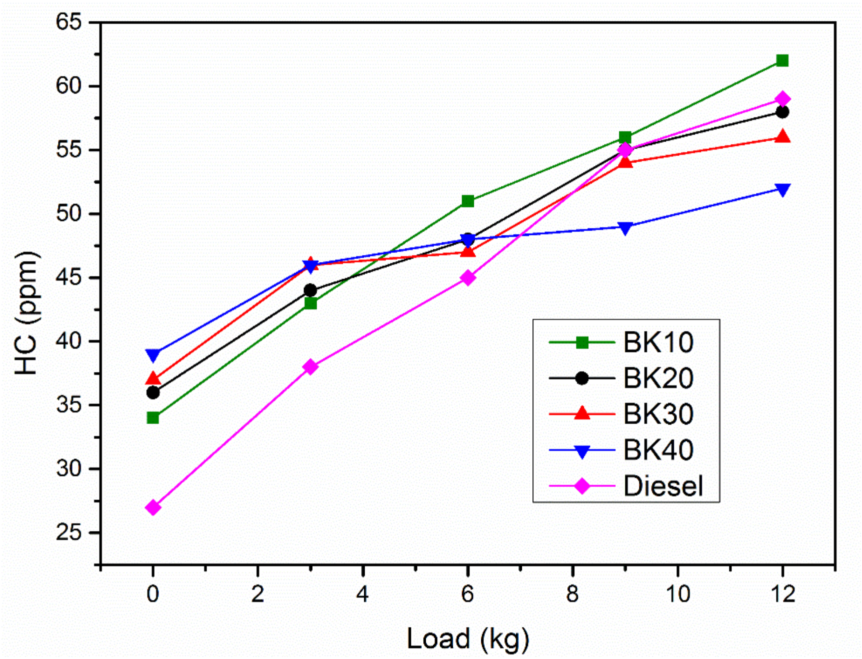


Fig. 4.6 Variation of unburned hydrocarbon emission with load

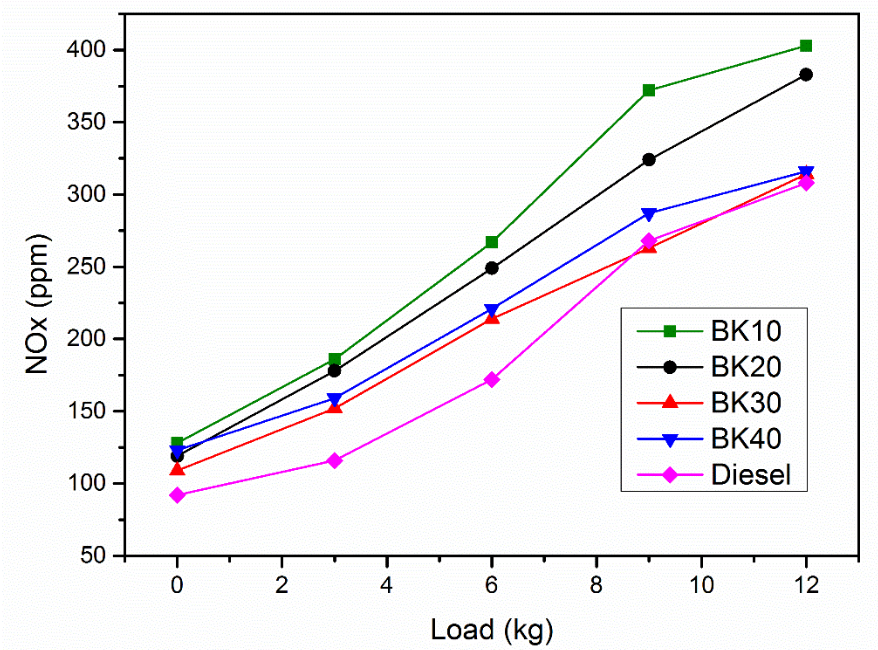


Fig. 4.7 Variation of oxides of nitrogen emission with load

#### 4.3.2.4 Smoke emissions

Figure 4.8 shows the smoke emissions at different loads for the test fuels. As shown in the figure smoke emissions increase with increase in the load. Since more fuel is injected at higher load which leads to decrease in air fuel ratio and some portion of fuel is exhausted in unburned state (Pali et al. 2015). For the biofuel blends used in this study smoke emissions were higher than diesel at all the loads. It was observed from the Figure 4.8 that the smoke emission reduce with increase in percentage of kerosene. For blends BK10 and BK20 smoke emission increased by 39.1% and 17.8% when compared to diesel. This is mainly due to poor combustion, because of higher viscosity of blend. However, in case of blends BK30 and BK40 smoke emission decreased by1.8% and 5.4% compared to diesel. This decrease in smoke emission is due to higher volatility of kerosene, by which combustion will be fast and complete.

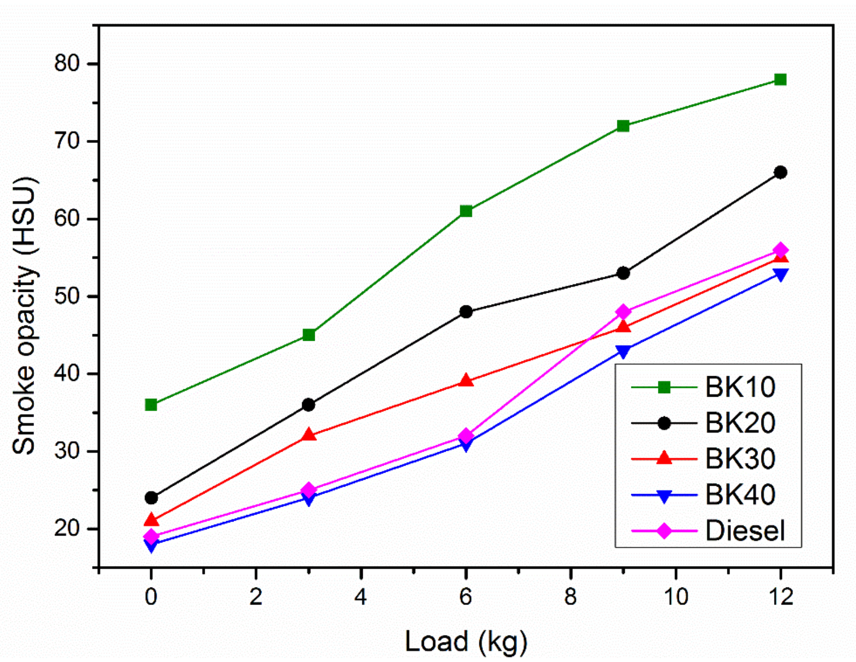


Fig. 4.8 Variation of smoke emission with load

Among all the blends performance of BK30 blend was nearer to the performance of diesel fuel. Further the emissions of BK30 blend was lower than the diesel fuel. Since the viscosity of BK10 blend is very high and performance of this blend is lower, further investigations were not carried out for this blend.

# CHAPTER 5

# EFFECT OF COMPRESSION RATIO ON PERFORMANCE AND EMISSIONS OF CARDANOL KEROSENE BLEND OPERATED DIESEL ENGINE

## 5.1 METHODOLOGY

To find the effect of compression ratio the engine was operated at three different compression ratios within the range (12:1 to 18:1) of the engine (i.e. 16:1, 17:1 & 18:1). The compression ratios at the higher end of the range were selected to check the possibility of knocking. The compression ratio was changed by tilting the cylinder head as per the specifications marked at the indicator.  The bolts on the tilting block are loosened and the adjuster bolt is rotated to set the required compression ratio, which is indicated by the indicator. All the three blends (BK20, BK30 and BK40) were tested at various combinations of compression ratio at 200 bar injection pressure and 23°BTDC injection timing. Diesel and Cardanol kerosene blends were filled in the dual fuel tank separately. The exhaust gas emissions (HC, CO and NOx) were measured with exhaust gas analyser (Netel exhaust gas analyser, Model: NPM-MGA-1) and the smoke opacity was measured by using smoke meter (Netel smoke meter model, NPM-SM-111B). This procedure is repeated for different blends and different compression ratio (16:1, 17:1 & 18:1). The fuel tank was drained and rinsed with new blend before filling the tank with new blend. The trails were repeated for all the test fuels and readings were noted after steady state. The brake specific fuel consumption, brake thermal efficiency and exhaust gas temperature readings were recorded from the computer and exhaust emissions were measured. Appendix III shows the experimental data at various compression ratios.

## 5.2 EFFECT OF COMPRESSION RATIO ON PERFORMANCE

When the compression ratio is varied the pressure and temperature inside the combustion chamber at the time fuel injection also varies, which effects the combustion process. The performance parameters changes with the type of fuel used and the compression ratio at which the engine operates.

### 5.2.1 Brake Thermal Efficiency

The variation of brake thermal efficiency (BTE) of various test blends at different compression ratio with respect to load is shown in Figure 5.1, Figure 5.2 and Figure 5.3. It was observed that, the BTE was increasing with increase in load for all the test fuels due to reduction in power loss with increase in load. When the compression ratio was increased from 16:1 to 18:1, the BTE was increased by 14.3%, 11.3% and 17.8% for the blends BK20, BK30 and BK40, respectively. This increase in BTE is due to higher temperature at the time of fuel injection at higher compression ratios, which favours fast and complete combustion (Roy et al., 2014). It was also observed that at all the compression ratio maximum thermal efficiency was recorded for BK30 blend compared to other blends, which is very close and slightly lower than the diesel fuel performance. Maximum efficiency of 29.87% and 30.36% was observed at full load for BK30 blend and diesel fuel, respectively at CR18.

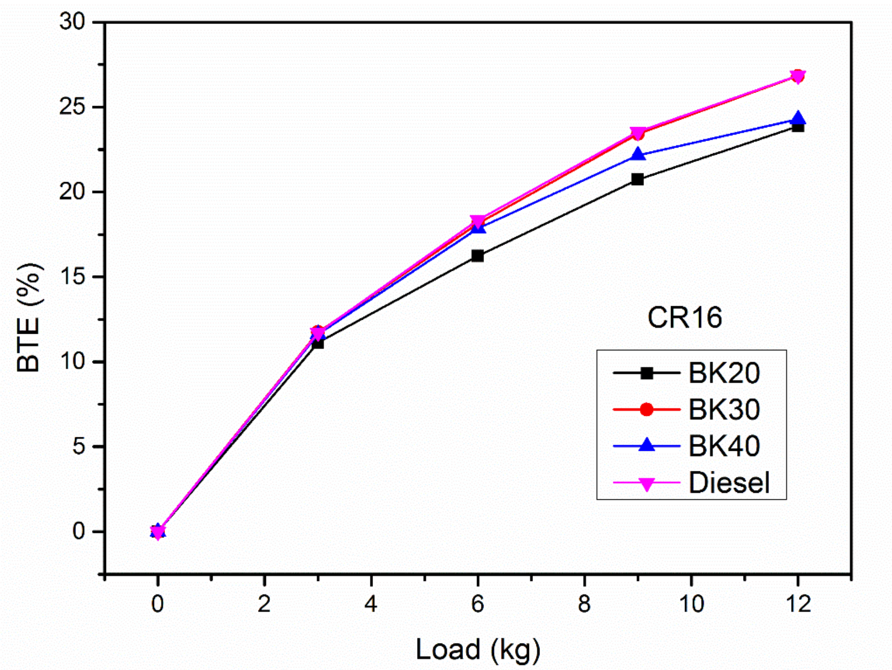


Fig. 5.1 Variation of BTE with load at 16:1 compression ratio

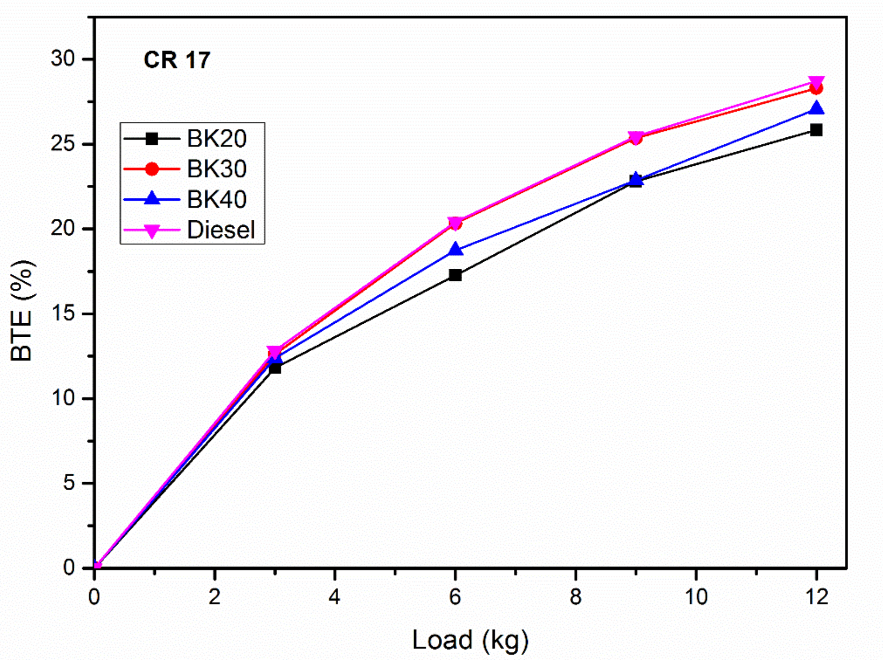


Fig. 5.2 Variation of BTE with load at 17:1 compression ratio

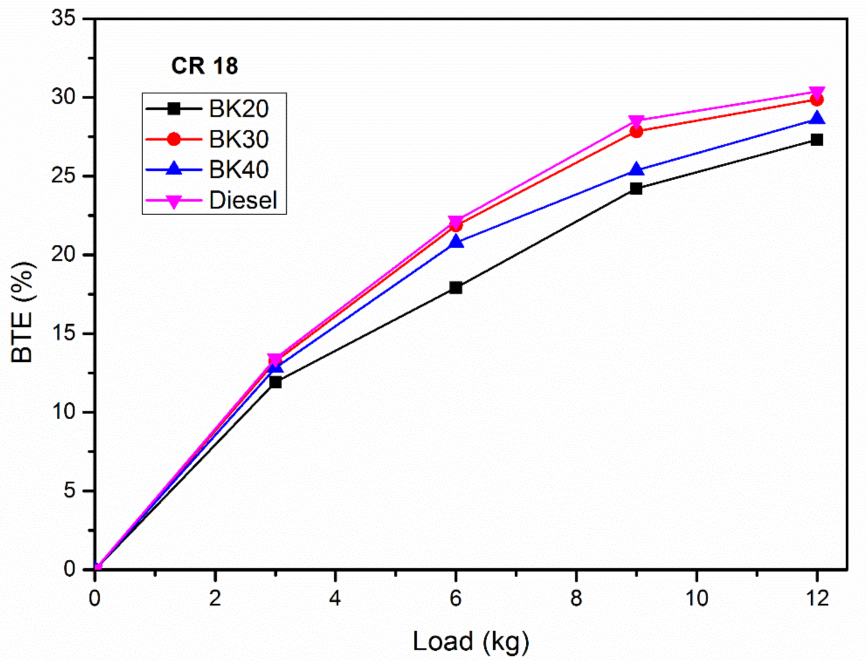


Fig. 5.3 Variation of BTE with load at 18:1 compression ratio

### 5.2.2 Brake Specific Fuel Consumption

Figure 5.4, Figure 5.5 and Figure 5.6 present the variation of brake specific fuel consumption (BSFC) with load for all the test fuels used at different compression ratios. It was observed that the BSFC for biofuel blends were higher than diesel fuel at all the loads, due to the fact that the biofuels have lower calorific value than diesel. Similar observations were also recorded by the other researchers (Harkude and Padalkar, 2014). The BSFC reduced with increase in compression ratio for all the test blends.The BSFC was reduced by 6% when the compression ratio was increased from 16:1 to 17:1 and the reduction was 6.4% when CR was increased from 17:1 to 18:1 for BK30 blend. At higher compression ratio the temperature in combustion chamber was high, so the combustion was complete,which reduced the BSFC.

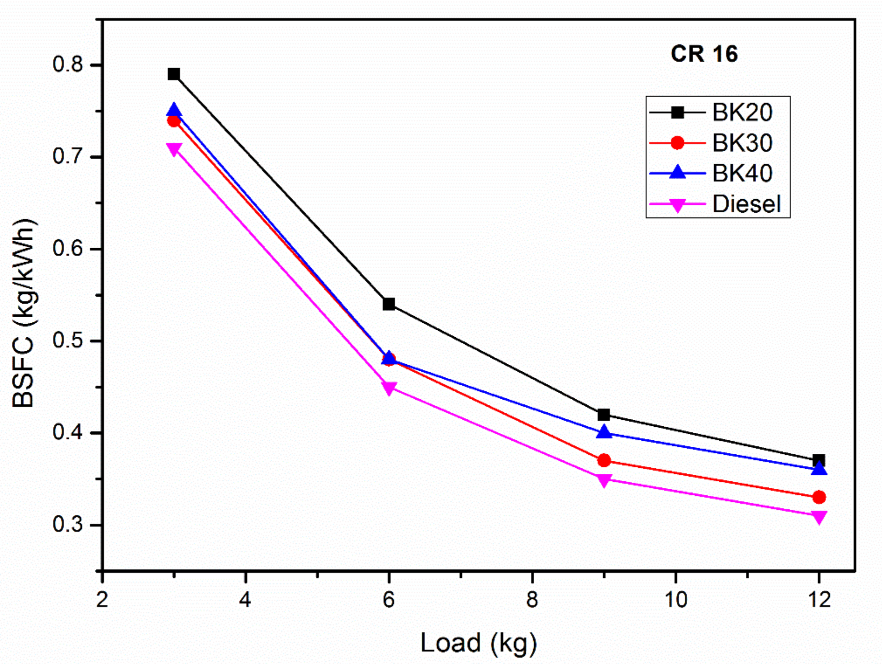


Fig.5.4 Variation of BSFC with load at 16:1 compression ratio

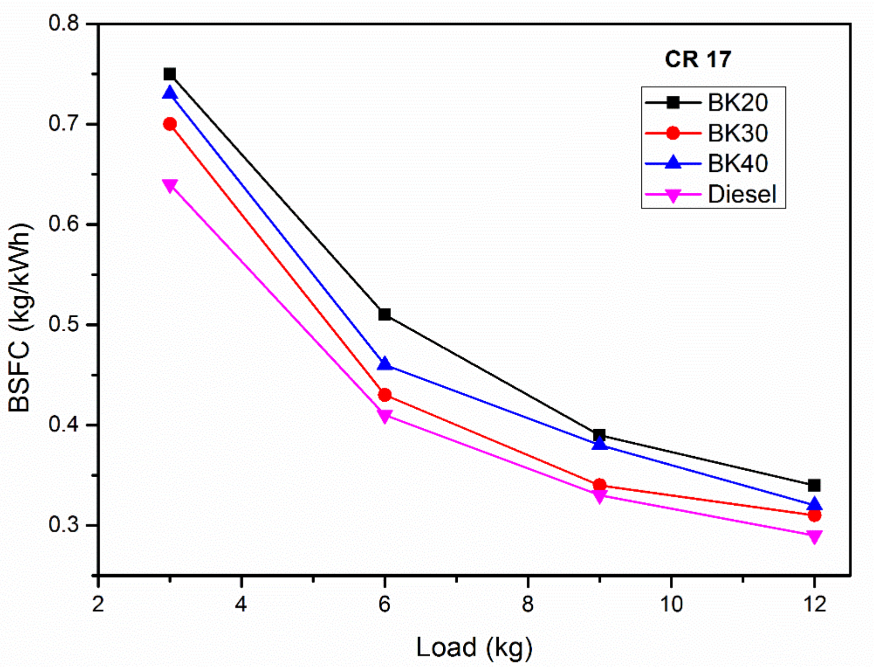


Fig. 5.5 Variation of BSFC with load at 17:1 compression ratio

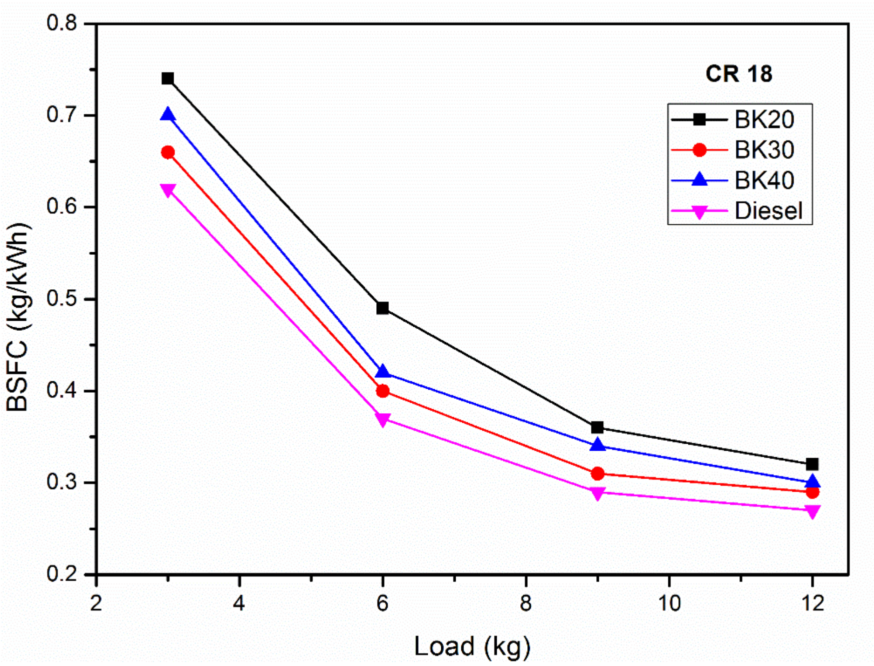


Fig. 5.6 Variation of BSFC with load at 18:1 compression ratio

### 5.2.3 Exhaust Gas Temperature

The exhaust gas temperature variations for different test fuels at various loads at different compression ratio are shown in Figure 5.7, Figure 5.8 and Figure 5.9. It was observed from graph that when compression ratio increases the exhaust gas temperature reduces for all the test fuels. When the compression ratio is increased the air temperature inside the combustion chamber increases, which reduces the ignition lag. Because of this the fuel burns fast and completely (Raheman and Ghadge, 2008). Lower exhaust gas temperature was observed for BK 30 blend compared to other blends at all compression ratios.

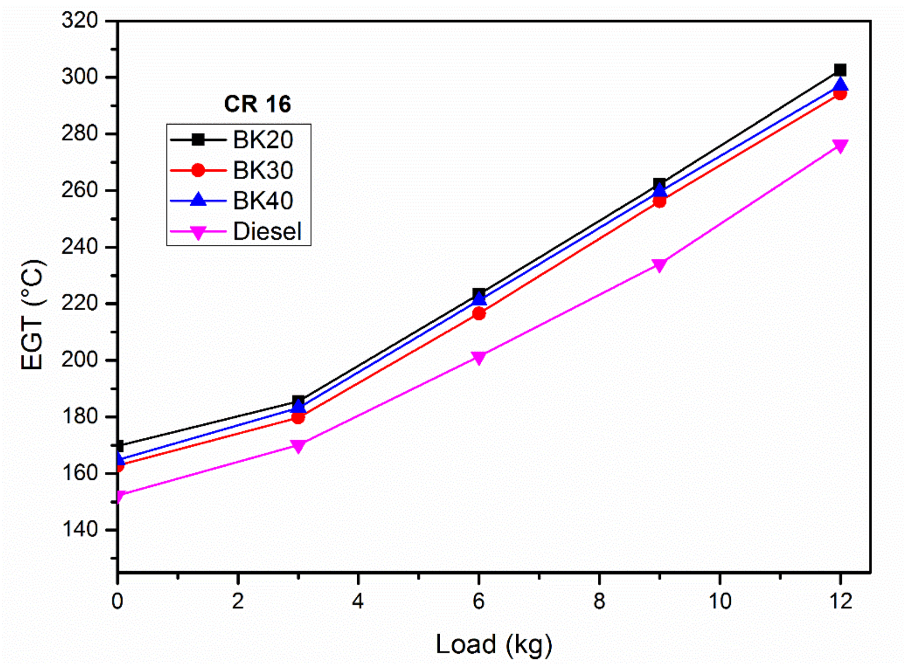


Fig. 5.7 Variation of EGT with load at 16:1 compression ratio

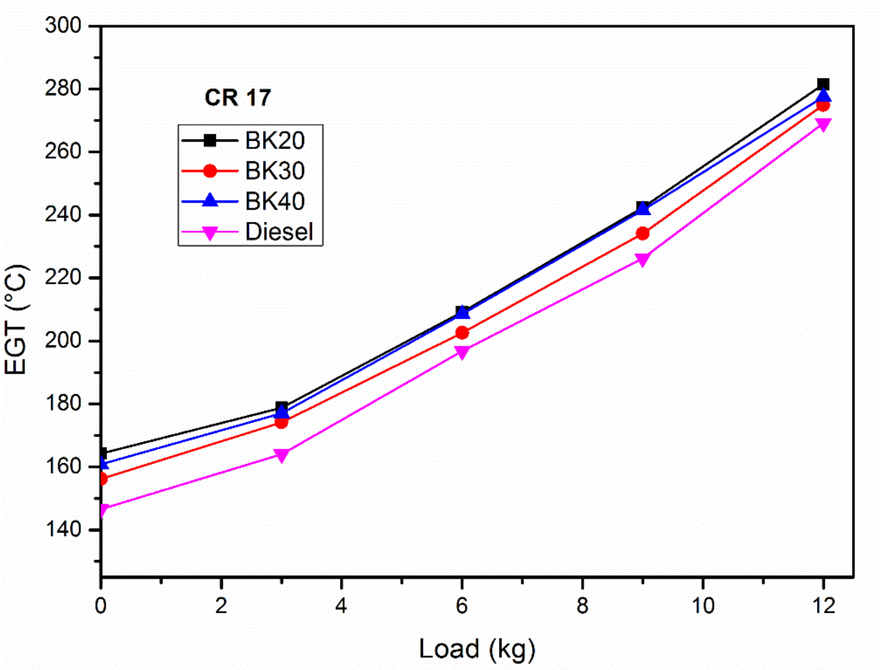


Fig. 5.8 Variation of EGT with load at 17:1 compression ratio

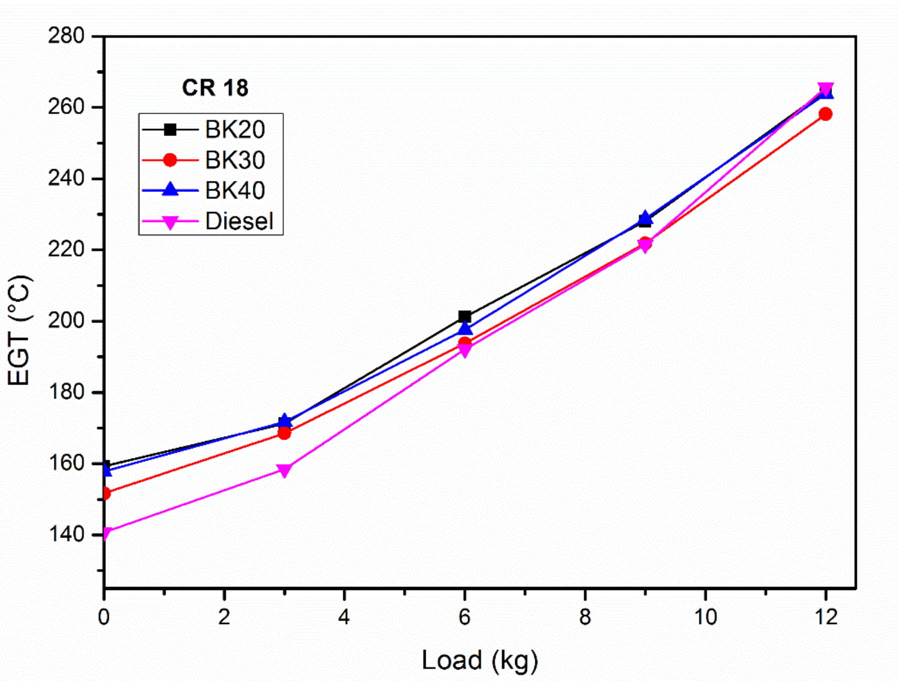


Fig. 5.9 Variation of EGT with load at 18:1 compression ratio

## 5.3 EFFECT OF COMPRESSION RATIO ON EMISSIONS

Formation of pollutants depends on the combustion process and the type of fuel used. With the change in compression ratio the emissions of the engine may increase or decrease, which mainly depends on the type of fuel burnt.

### 5.3.1 Carbon Monoxide Emissions

Carbon monoxide (CO) is produced due to lack of oxygen during combustion in the combustion chamber. Figure 5.10, Figure 5.11 and Figure 5.12 show the variation of CO emission with load for biofuel blends and diesel fuel at various compression ratios. As depicted in the figures the CO emission for the kerosene biofuel blends reduces with increase in load and kerosene percentage in the blend. In case of diesel fuel CO emission increases with the load and this may be due to extra fuel injection at full load (Jindal et al., 2010). It is remarked from the graphs that with increase in compression ratio the CO emission was reduced for all the biofuel blends and diesel fuel. It was mainly because of higher air temperature at higher compression ratio, which leads to complete combustion.

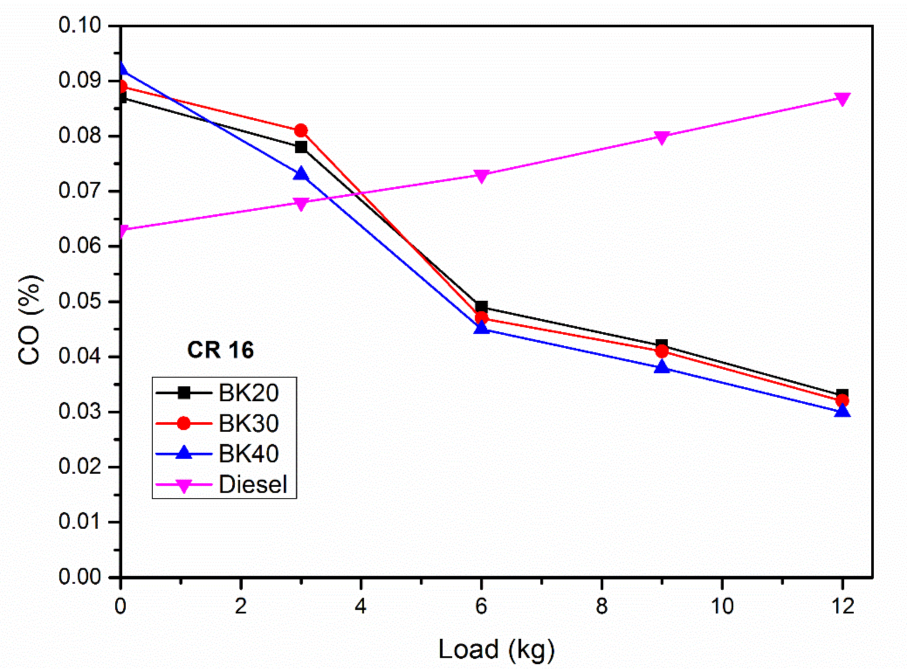


Fig. 5.10 Variation of CO with load at 16:1 compression ratio

When the compression ratio was increased from 16 to 18 the CO emissions were reduced by 21%,13%,10% and 13% for the blend BK20,BK30,BK40 and diesel fuel, respectively at peak load.

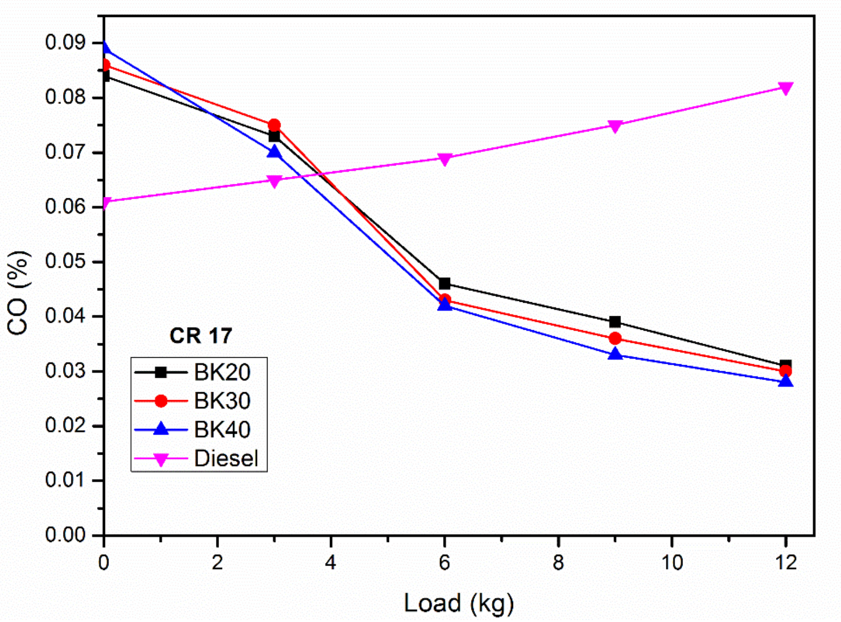


Fig. 5.11 Variation of CO with load at 17:1 compression ratio

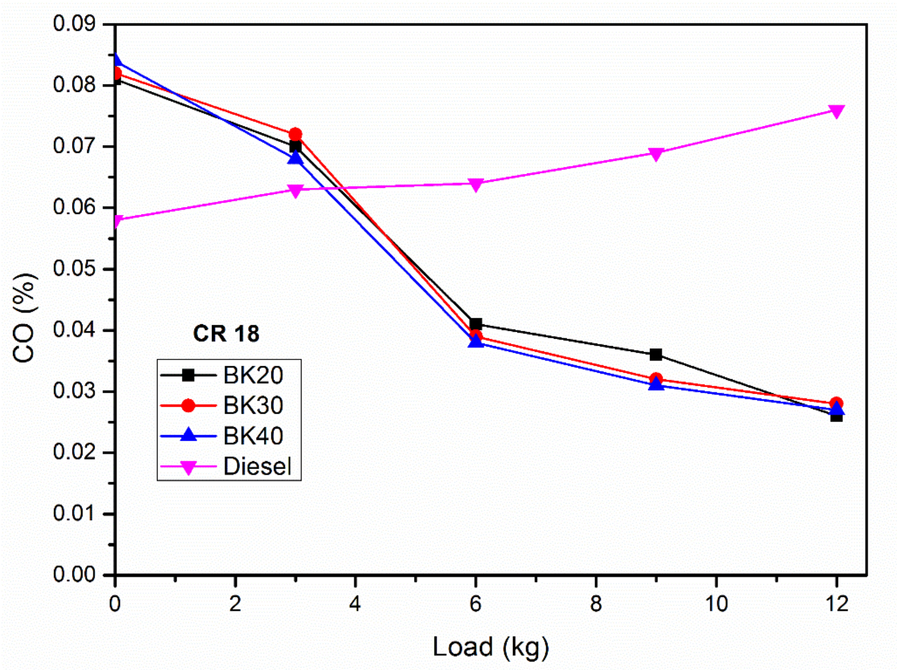


Fig. 5.12 Variation of CO with load at 18:1 compression ratio

### 5.3.2 Unburned Hydrocarbon Emissions

Unburned hydrocarbons (HC) were emitted due to incomplete combustion. Figure 5.13, Figure 5.14 and Figure 5.15 indicates the HC emission at various loads for all the test fuels tested at different compression ratios. As highlighted in the figures the HC emission increases with increase in load for all the biofuel blends tested and the diesel fuel. Similar observations were reported by other researchers also (Natesan, 2013). With increase in compression ratio the HC emission was reduced for all the tested fuels. By increasing compression ratio from 16:1 to 18:1, the HC emission was reduced 14%,15%, 22% and 20% for BK20, BK30, BK40 and diesel fuel respectively. With the increase in percentage of kerosene in the blend, rate of combustion increases due to the higher volatility of kerosene. When the compression ratio was increased the delay period reduces and the combustion was complete,which leads to reduction in HC emission.

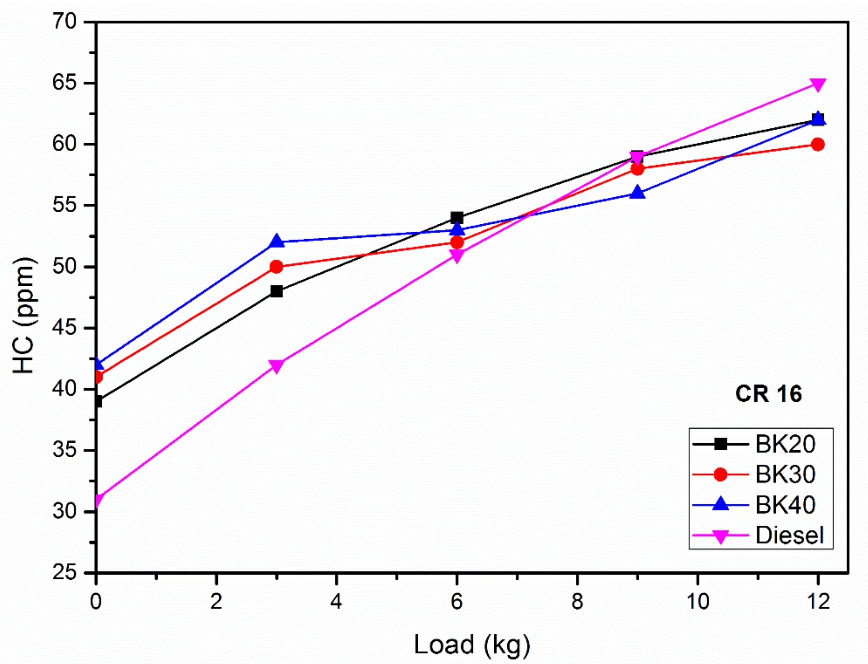


Fig. 5.13 Variation of HC with load at 16:1 compression ratio

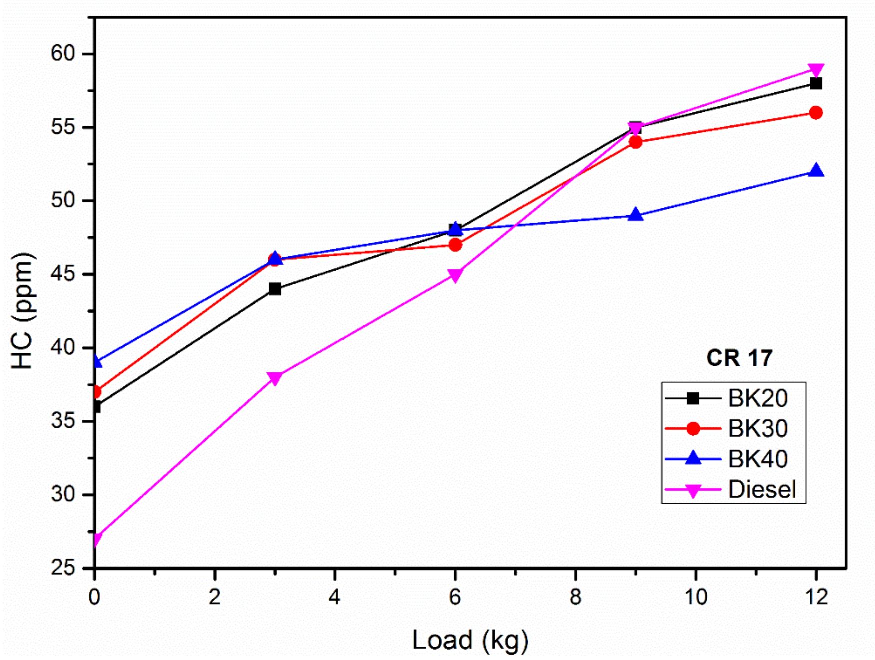


Fig. 5.14 Variation of HC with load at 17:1 compression ratio

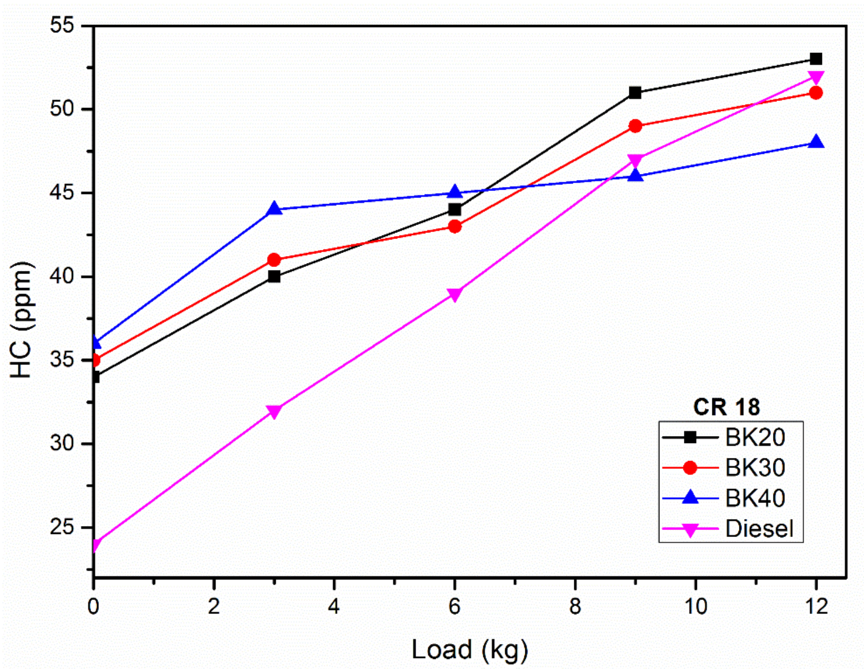


Fig. 5.15 Variation of HC with load at 18:1 compression ratio

### 5.3.3 Nitrogen Oxides (NOx) Emission

Oxides of nitrogen were formed inside the combustion chamber due to high temperature and availability of excess oxygen inside the combustion chamber. Variation of NOx emission with load for different blends is represented in Figure 5.16, Figure 5.17 and Figure 5.18. It is observed from the figures that the NOx emission increases with increase in compression ratio. It is mainly due to higher temperature at higher compression ratio (Sharma and Murugan, 2013). NOx emission for biofuel blends were higher than that of diesel fuel at all the load. BK30 blend emits less NOx than the other tested blends at peak load. But this emission is slightly higher than that of diesel fuel. When the compression ratio was increased from 16 to 18 the NOx emission raised by 20%, 23%, 24% and24% for BK20, BK30, BK40 and diesel fuel, respectively.

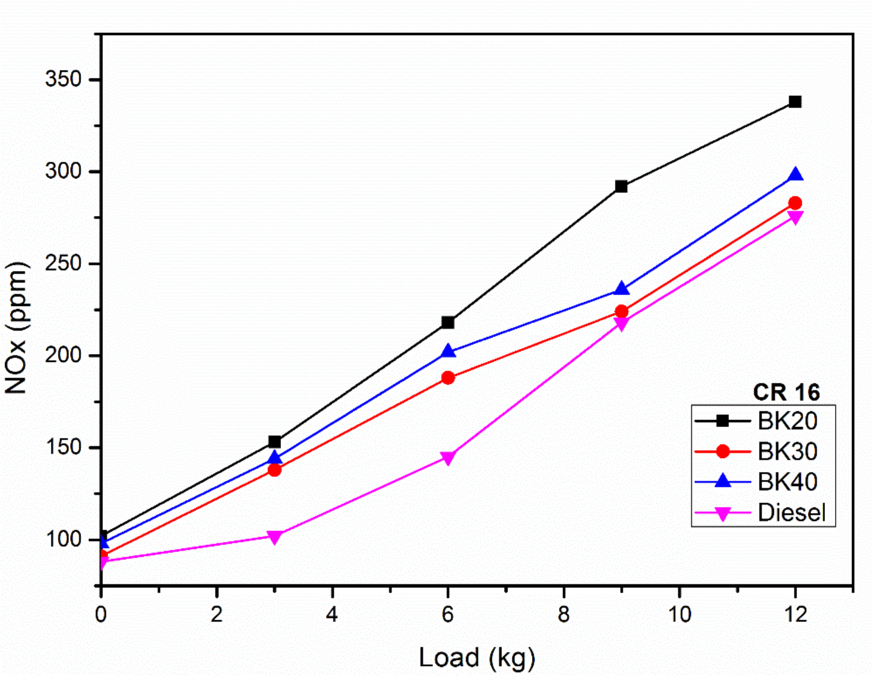


Fig.5.16 Variation of NOx with load at 16:1 compression ratio

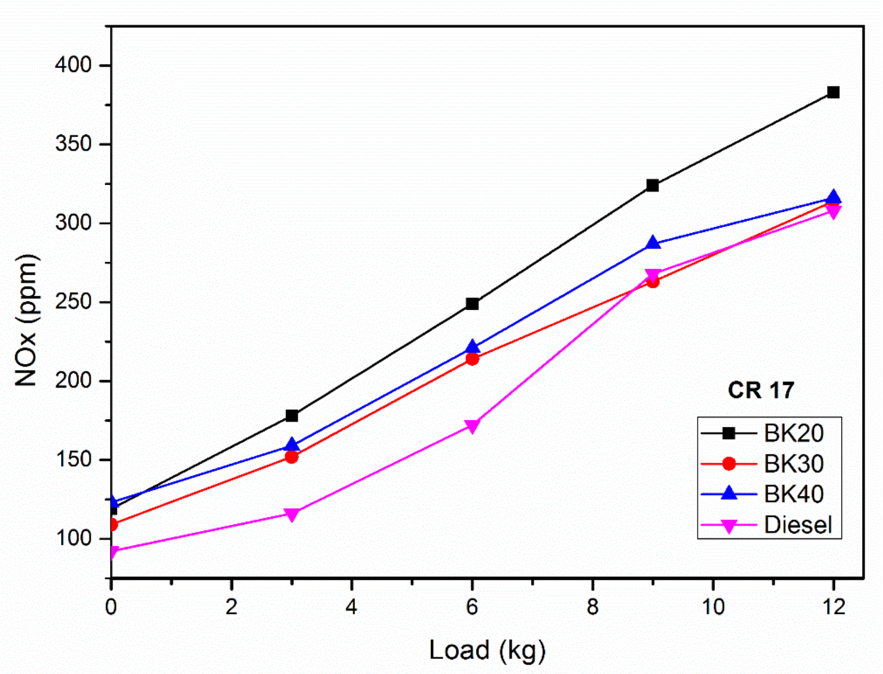


Fig. 5.17 Variation of NOx with load at 17:1 compression ratio

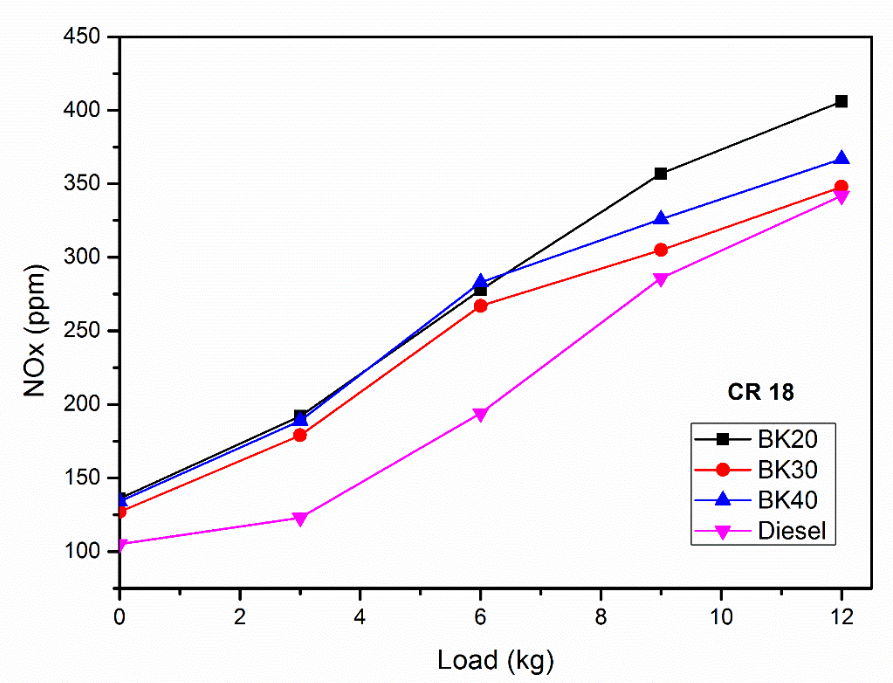


Fig. 5.18 Variation of NOx with load at 18:1 compression ratio

### 5.3.4 Smoke Emission

Figure 5.19, Figure 5.20 and Figure 5.21 represents the smoke emissions at various loads for all the test fuels. From the graphs shown in these figures it is observed that the smoke emission increases with increase in load for all the test fuels. When the percentage of kerosene in the blend increased, the smoke emission reduces because of increased rate of combustion at higher percentage of kerosene. It was also observed that with increase in compression ratio there was reduction in the smoke. When the compression ratio was increased from 16:1 to 18:1 the smoke emission reduced by 5.8%, 20%, 13% and18% for BK20, BK30, BK40 and diesel fuel, respectively. With the increase in compression ratio, the pressure and temperature of combustion chamber increases which promotes the combustion process and reduces the smoke. At peak load BK30 and BK40 blend emits less smoke than diesel fuel at all the tested compression ratios.

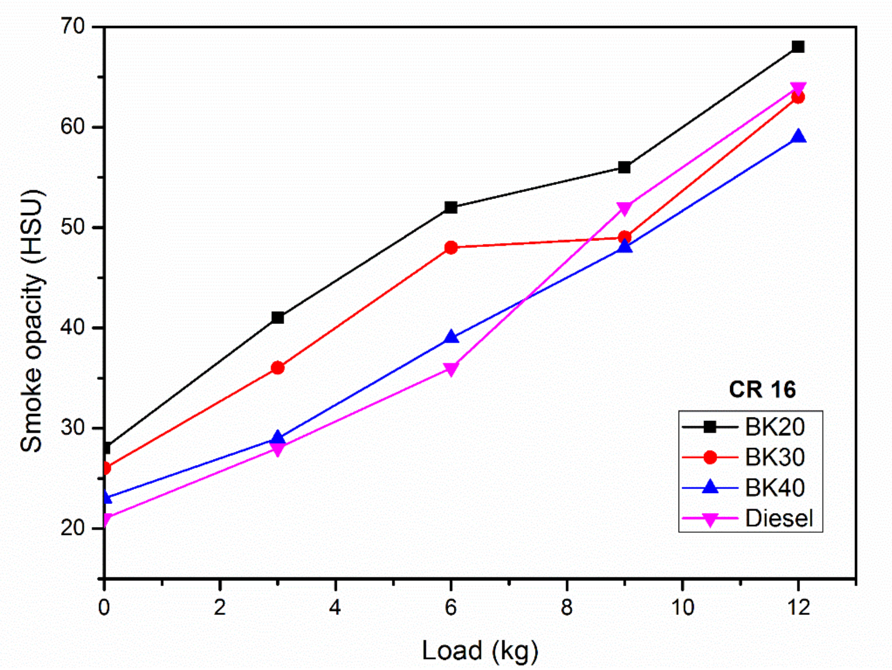


Fig. 5.19 Variation of smoke with load at 16:1 compression ratio

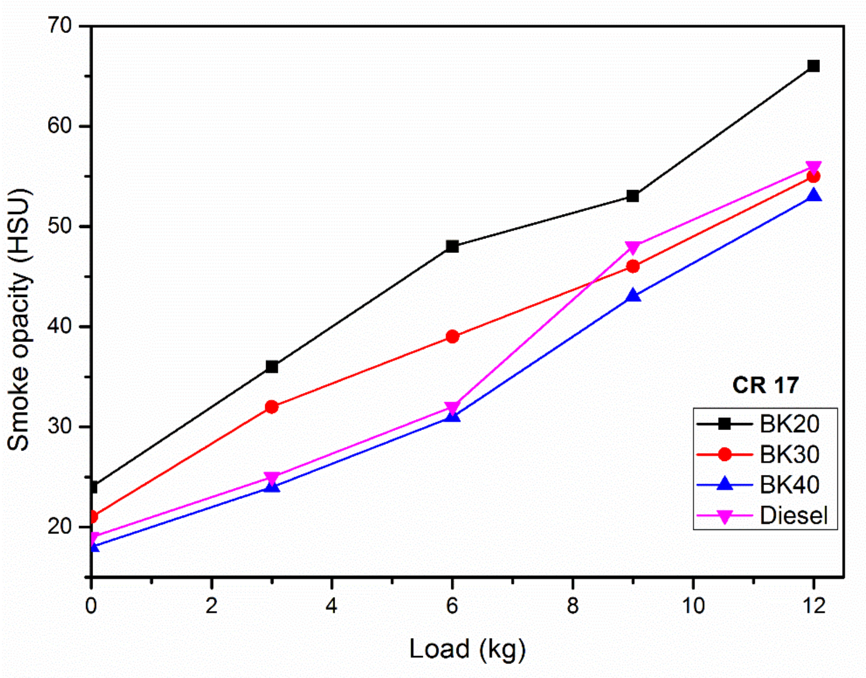


Fig. 5.20 Variation of smoke with load at 17:1 compression ratio

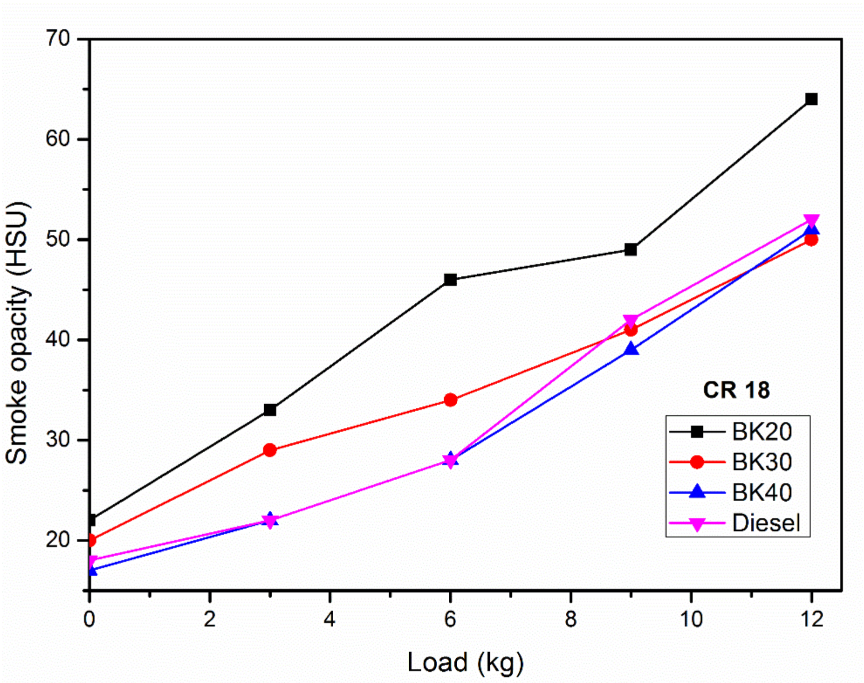


Fig. 5.21 Variation of smoke with load at 18:1 compression ratio

# CHAPTER 6

# EFFECT OF INJECTION PRESSURE ON PERFORMANCE AND EMISSIONS OF CARDANOL KEROSENE BLEND OPERATED DIESEL ENGINE

## 6.1 METHODOLOGY

The experimental investigations were carried out in a water cooled single cylinder direct injection four stroke diesel engine operated at 17:1 compression ratio, 23° BTDC injection timing. A computer has been connected to the engine for capturing data. Netel exhaust gas analyser (Model: NPM-MGA-1) and Netel smoke meter (model: (NPM-SM-111B) were used for measurement of exhaust gas emissions and smoke opacity.

All the experiments were conducted at 23° BTDC injection timing and 17:1 compression ratio at various injection pressures (180bar, 200bar and 220bar). Engine was started with diesel fuel by hand cranking and allowed in idling condition for warm up. The engine was loaded by adjusting the dynamometer. After the steady state fuel consumption rate was noted. Brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature were recorded from the computer and the exhaust emissions were measured using exhaust gas analyser and smoke meter. The trail was repeated for the blend filled in the tank by changing the position of the fuel valve fitted in the fuel tank. The fuel blend in the tank was drained and rinsed with the new blend before filling it. The same procedure was repeated for all the blends (i.e. BK20, BK30 and BK40). Then the trails were repeated for other injection pressures (180 bar and 220 bar). The injection pressure was altered by varying the spring tension of the fuel injector by tightening or loosening the set screw at the top of the injector. Required injection pressure was set by checking the pressure with the help of pressure gauge. The readings of all the trails at various injection pressure are tabulated in Appendix-IV.

## 6.2 EFFECT OF INJECTION PRESSURE ON PERFORMANCE

The performance of the engine depends on the combustion of fuel. The combustion process is influenced by atomisation and mixing of fuel inside the combustion chamber, which changes with the injection pressure.

### 6.2.1 Brake Thermal Efficiency

Figure 6.1, Figure 6.2 and Figure 6.3 show the variation of BTE with load for different kerosene and Cardanol blends and diesel at three different injection pressures. As depicted in these figures, for all the test fuels there is an increase in BTE with increase in load. When the load increased the friction to brake power ratio reduces and also heat loss reduces. The BTE for BK30 blend was higher than the other two blends (i.e., BK20 and BK40) at all injection pressure, but a little lower than the diesel fuel. When the injection pressure was reduced from 200 bar to 180 bar the BTE was reduced by 8.7%, 6.1%, 6.06% and 6.5% for BK20, BK30, BK40 and diesel fuel, respectively at peak load condition. When injection pressure was increased from 200 bar to 220 bar at peak load the BTE was increased by 7%, 10.7%, 10.17% and 11.3% for BK20, BK30, BK40 and diesel fuel, respectively. When injection pressure was increased BTE increases due to better atomisation at increased injection pressure (Kannan and Anand, 2012). At peak load condition the maximum BTE of 31.34% was recorded for BK30 at 220 injection pressure and for diesel fuel it was 31.97% at the same operating condition.

### 6.2.2 Brake Specific Fuel Consumption

Brake specific fuel consumption is the quantity of fuel burned in the engine combustion chamber to produce unit power in unit time. Figure 6.4, Figure 6.5 and Figure 6.6 illustrates the variation of BSFC with load for all the test fuels at different injection pressure.  As observed from these figures, when the injection pressure was increased the BSFC reduces for all the test fuels. This is because of better atomisation at higher injection pressures, which increases the combustion speed. The BSFC of biofuel blends were greater than the diesel due to poor heating values of biofuel. Similar observations were also reported by Jindal et al. (2010) for Jatropha methyl ester blends with diesel.

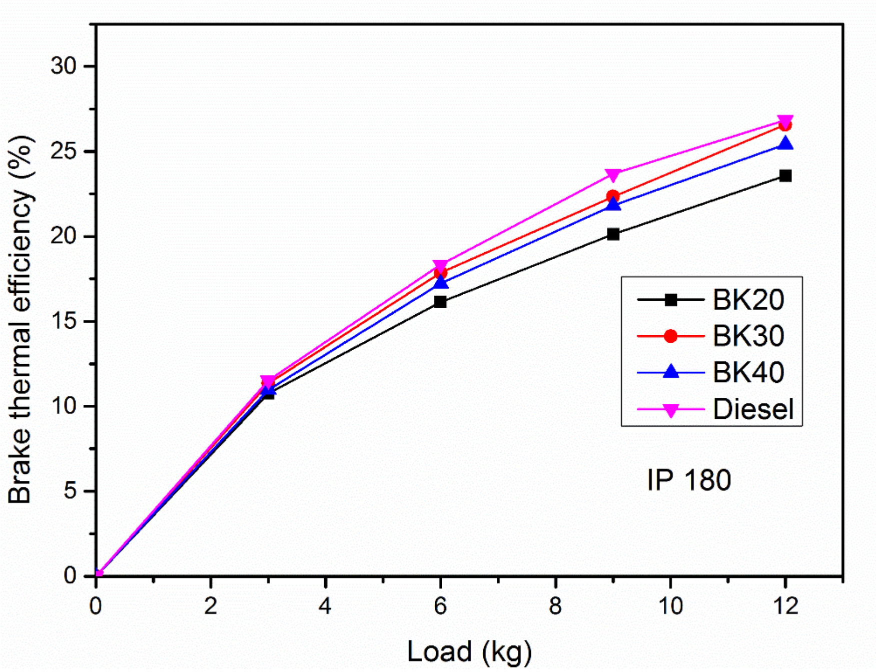


Fig. 6.1 Variation of BTE with load at 180 bar injection pressure

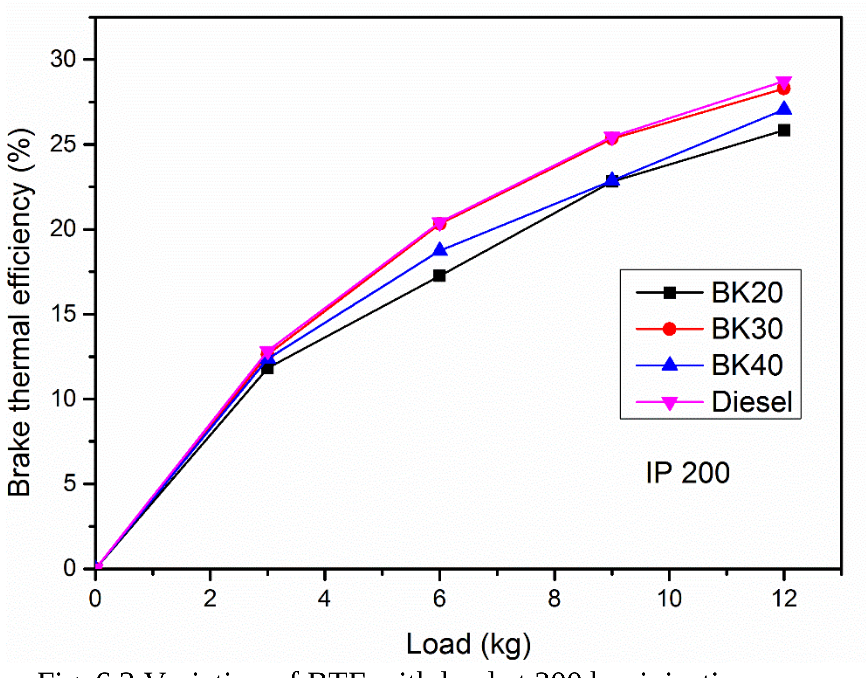


Fig. 6.2 Variation of BTE with load at 200 bar injection pressure

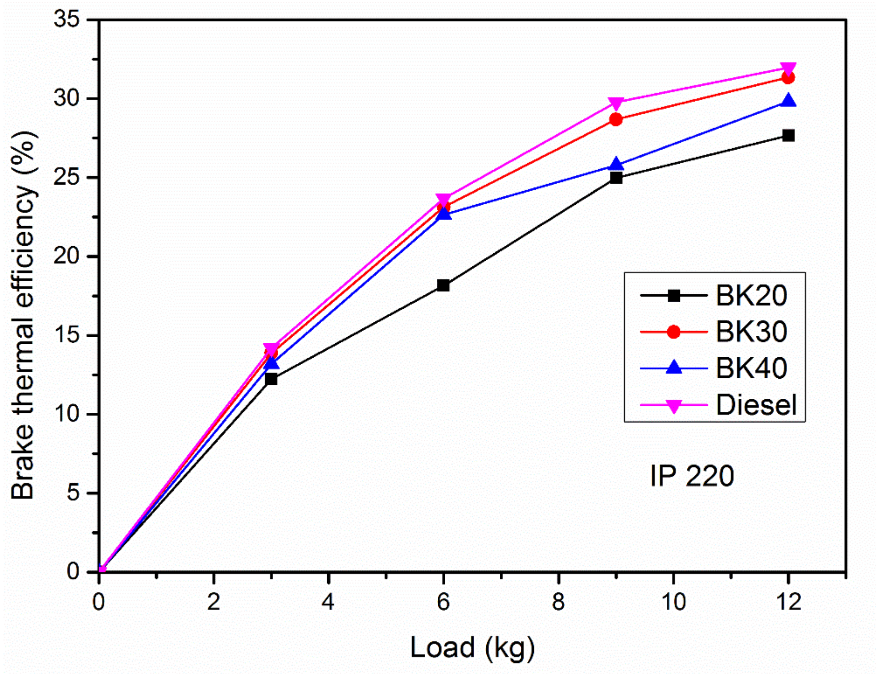


Fig. 6.3 Variation of BTE with load at 220 bar injection pressure

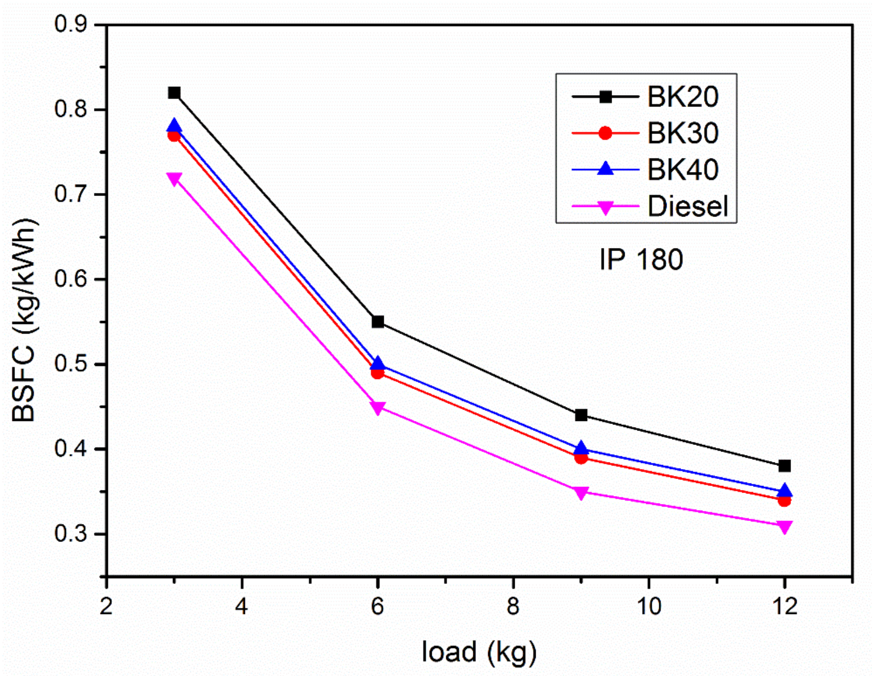


Fig. 6.4 Variation of BSFC with load at 180 bar injection pressure

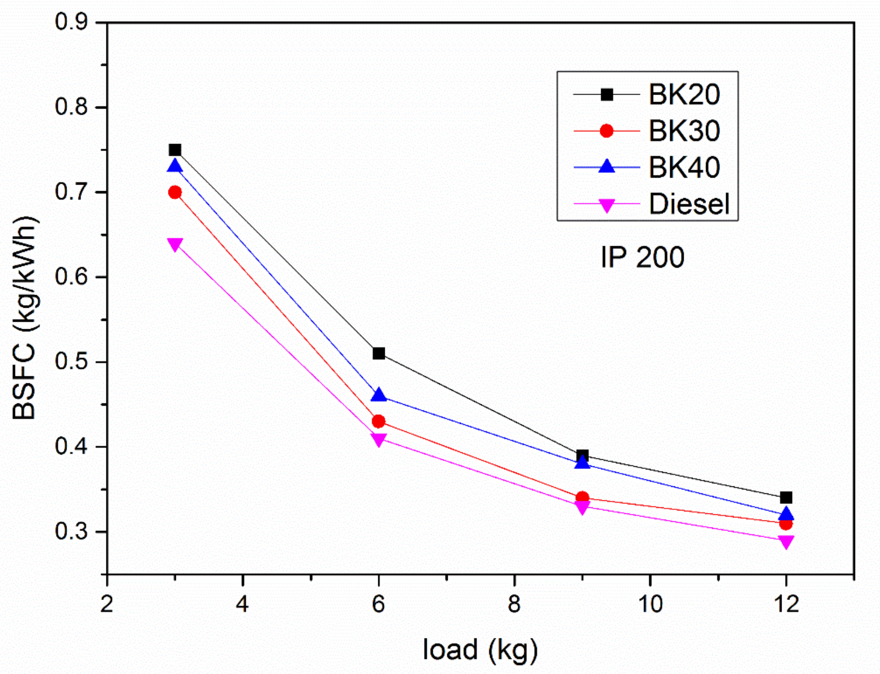


Fig. 6.5 Variation of BSFC with load at 200 bar injection pressure

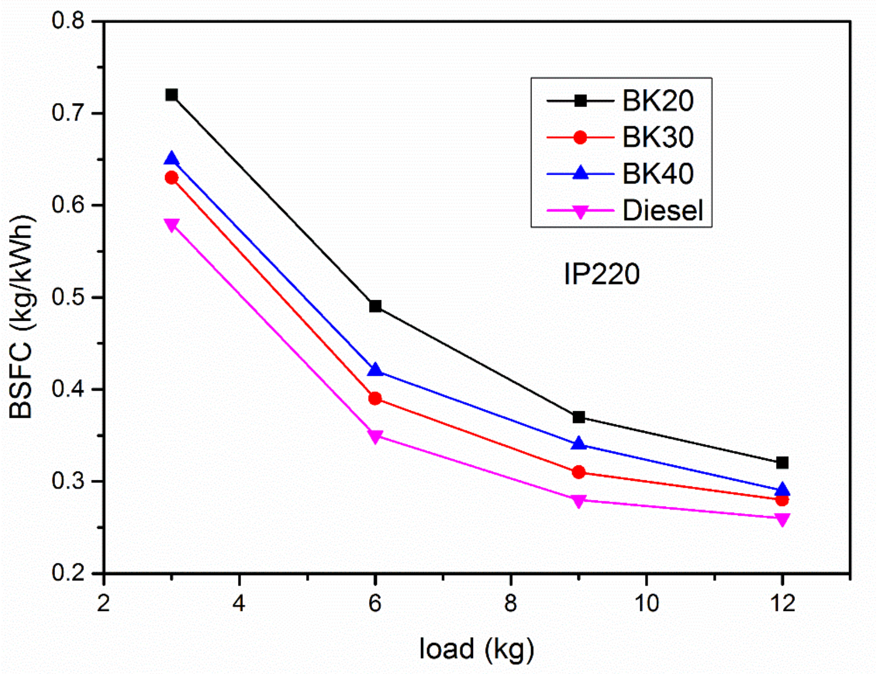


Fig. 6.6 Variation of BSFC with load at 220 bar injection pressure

### 6.2.3 Exhaust Gas Temperature

The exhaust gas temperature variations for different test fuels at various loads at different injection pressure are shown in Figure 6.7, Figure 6.8 and Figure 6.9. It was observed that when injection pressure increases the exhaust gas temperature reduces for all the test fuels. Lower exhaust gas temperature was observed for BK30 blend compared to other blends at all injection pressures. When injection pressure is increased combustion was fast due to proper atomisation which leads to lower exhaust gas temperature.

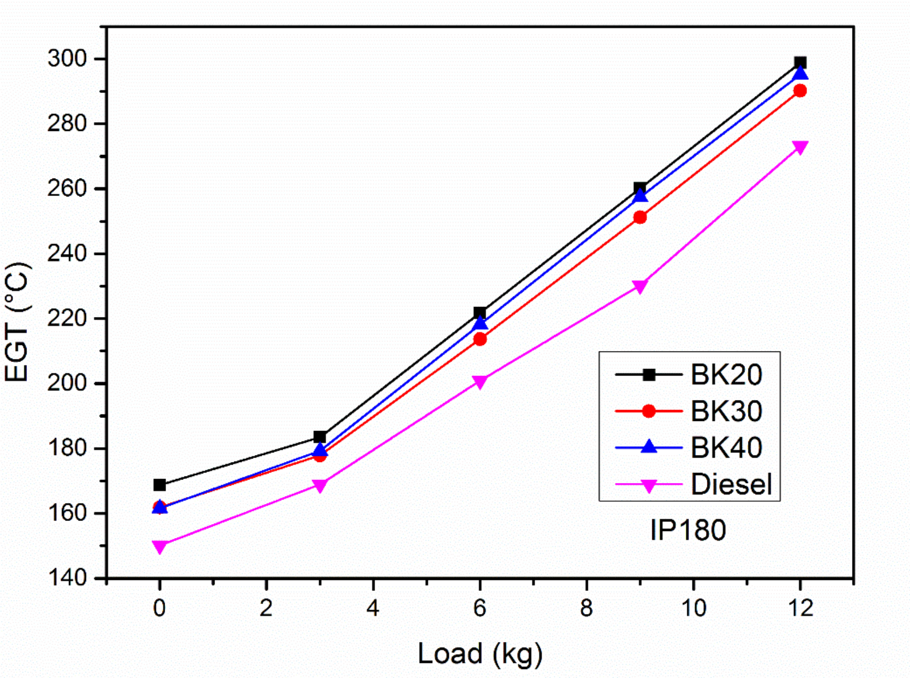


Fig. 6.7 Variation of EGT with load at 180 bar injection pressure

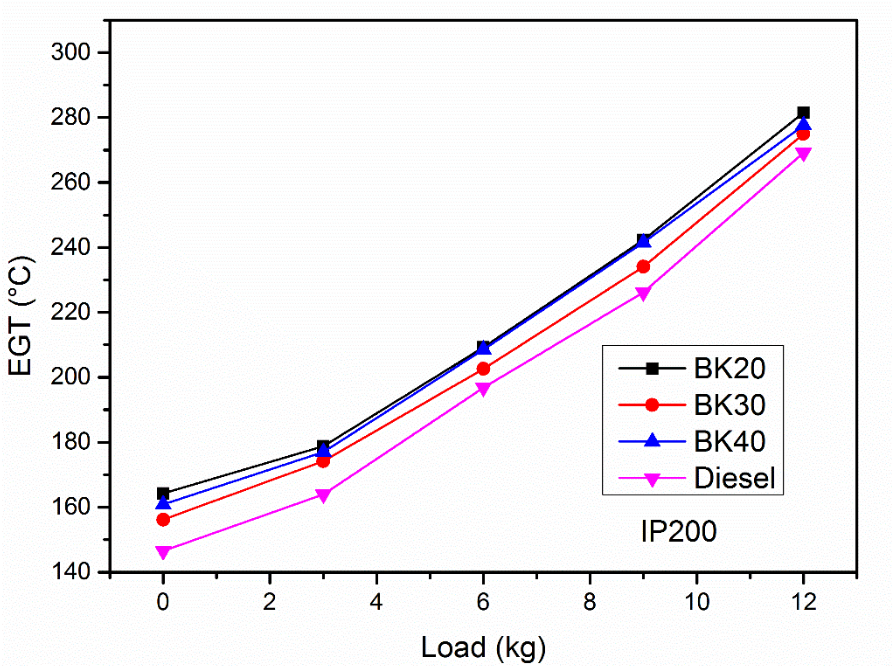


Fig. 6.8 Variation of EGT with load at 200 bar injection pressure

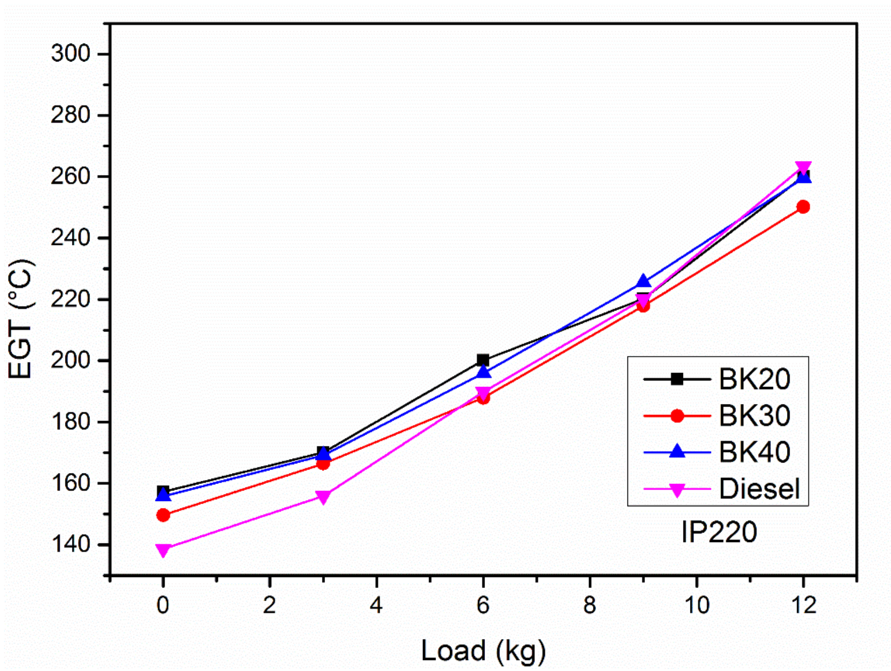


Fig. 6.9 Variation of EGT with load at 220 bar injection pressure

## 6.3 EFFECT OF INJECTION PRESSURE ON EMISSIONS

The emissions of the engine vary with the variations in the injection parameters which effcts the combustion of fuel. When injection pressure is increased the fuel enters into the combustion chamber as fine particle which enhances the combustion.

### 6.3.1 Carbon Monoxide Emission

Figure 6.10, Figure 6.11 and Figure 6.12 depicts emission of CO at various loads for different test fuels. The CO emission is reduced for all the biofuel blends with increase in load, but for diesel fuel a reverse trend was observed. It was because of more diesel injection at elevated loads, which leads to partial combustion of some portion of fuel, hence CO emission increases (Nateshan, 2013). At lower loads, formation of CO was intensified with increase in kerosene percentage in the blend. Because of higher volatility of kerosene, the combustion chamber temperature lowered, which reduces the oxidation of CO into $CO_2$ (Roy et al., 2014). At higher loads CO reduced with increase in kerosene percentage. The combustion was complete due to higher temperature at higher loads and higher volatility of kerosene. When the injection pressure was increased from 200 bar to 220 bar, the CO emission reduced due to good atomization, whereas the CO emission increased when IP was reduced from 200 bar to 180 bar.

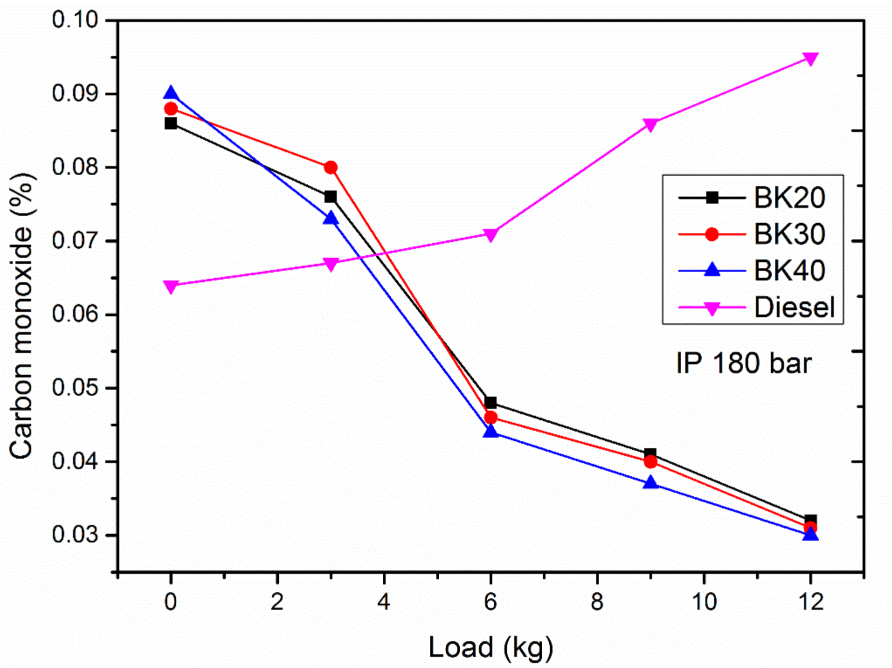


Fig. 6.10 Variation of CO with load at 180 bar injection pressure

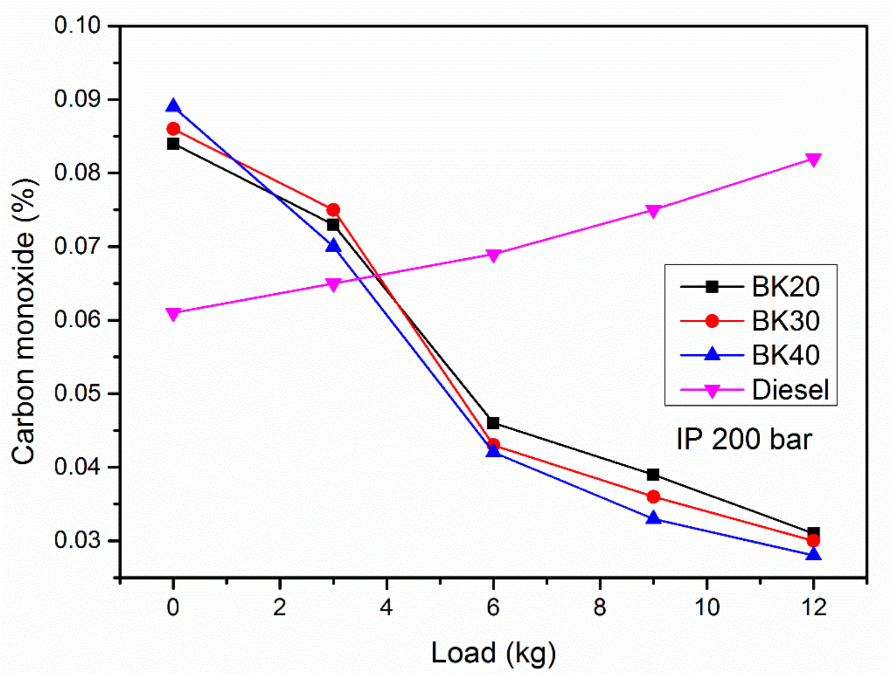


Fig. 6.11 Variation of CO with load at 200 bar injection pressure

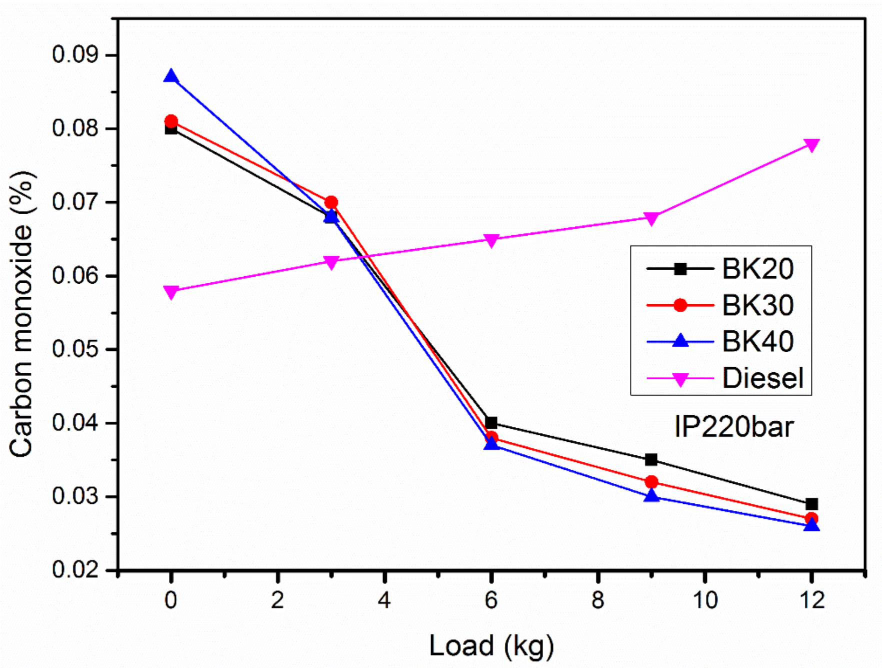


Fig. 6.12 Variation of CO with load at 220 bar injection pressure

### 6.3.2 Unburned Hydrocarbon Emission

The unburned hydrocarbons were emitted due to imperfect combustion. The HC emissions of all the test fuels at various loads for different injection pressure are depicted in Figure 6.13, Figure 6.14 and Figure6.15. As seen in these figures, for all the test fuels HC emission increases with load and this is because of more fuel injection at peak loads. The kerosene blends burn faster than diesel at higher loads due to higher volatility of kerosene. Because of this the HC emission was high for diesel at higher loads. When the injection pressure was increased proper atomization of fuel takes place, which leads to complete combustion and in turn reduction in HC emission. When injection pressure was increased from 200 bar to 220 bar at peak load the HC emission was reduced by 17%, 18.7%, 15.5% and 21.3% for BK20, BK30, BK40 and diesel fuel, respectively.

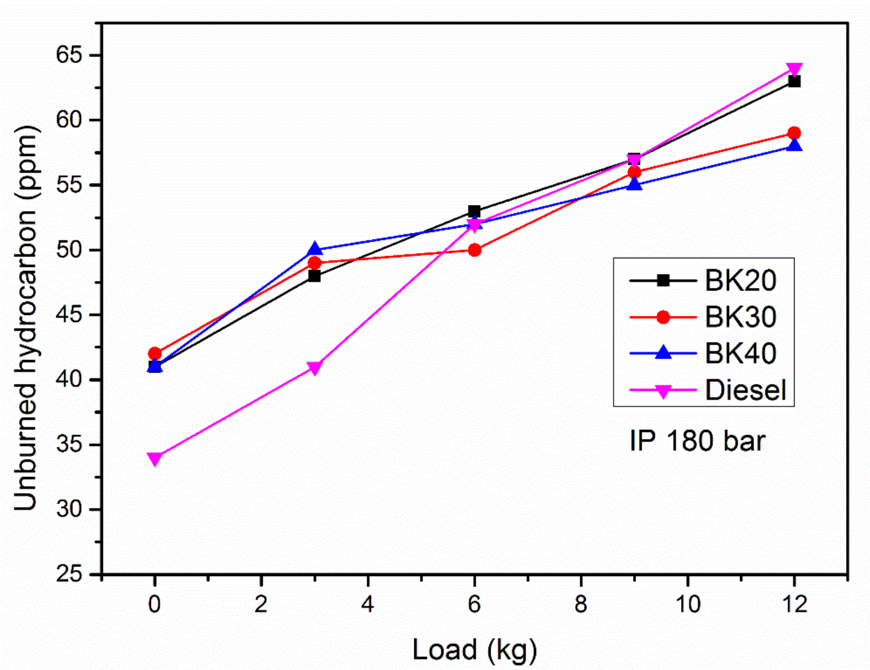


Fig. 6.13 Variation of HC with load at 180 bar injection pressure

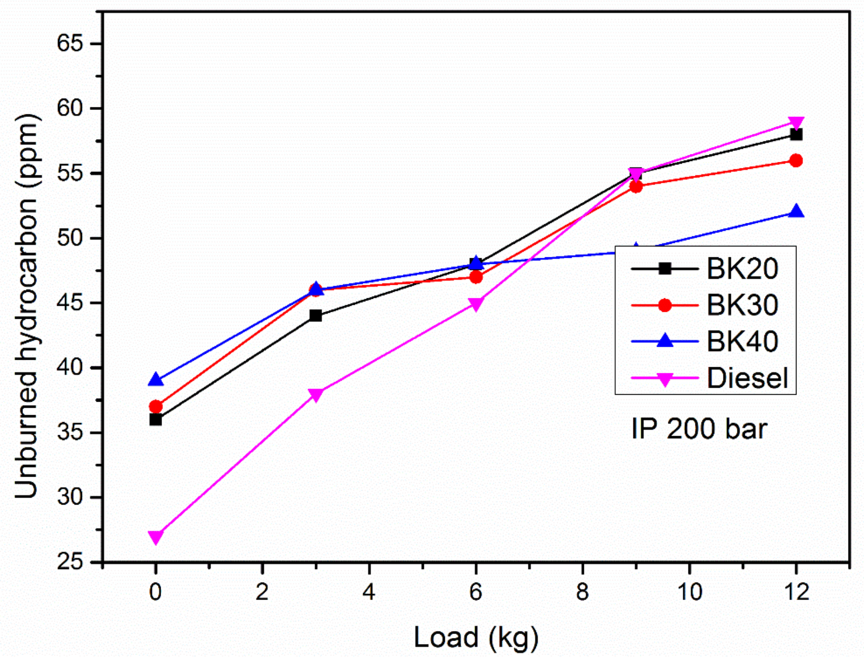


Fig. 6.14 Variation of HC with load at 200 bar injection pressure

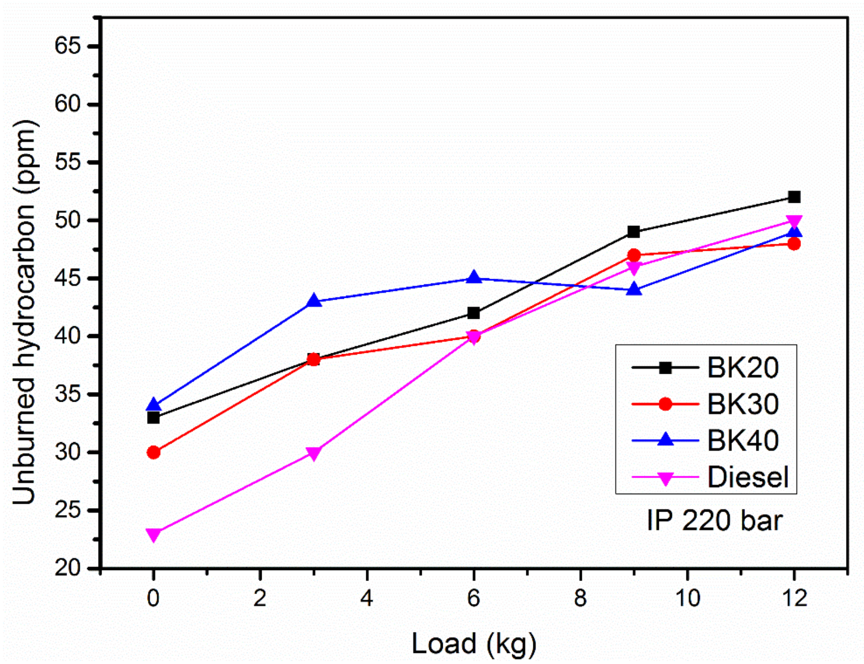


Fig. 6.15 Variation of HC with load at 220 bar injection pressure

### 6.3.3 Oxides of Nitrogen Emission

Oxides of nitrogen were formed due to elevated combustion temperature and higher oxygen concentration inside the combustion chamber. Figure6.16, Figure 6.17 and Figure 6.18 show emissions of NOx at various loads for different injection pressures. When injection pressure was increased the NOx emission also increases. Increase in injection pressure results in faster combustion due to rapid atomisation, which increase the in-cylinder temperature, resulting in more NOx formation. Similar outcomes have been conveyed by Nathagopal et al., (2016) and Aalam et al., (2016) for Calophllum inophyllum methyl ester and mahua methyl ester respectively. Among all blends, BK30 emits lower NOx at all injection pressure, but a little higher than the diesel fuel. When injection pressure was increased from 180 bar to 220 bar, the NOx emission was increased by 16%, 20.7%, 23% and 20% for BK20, BK30, BK40 and diesel fuel, respectively.

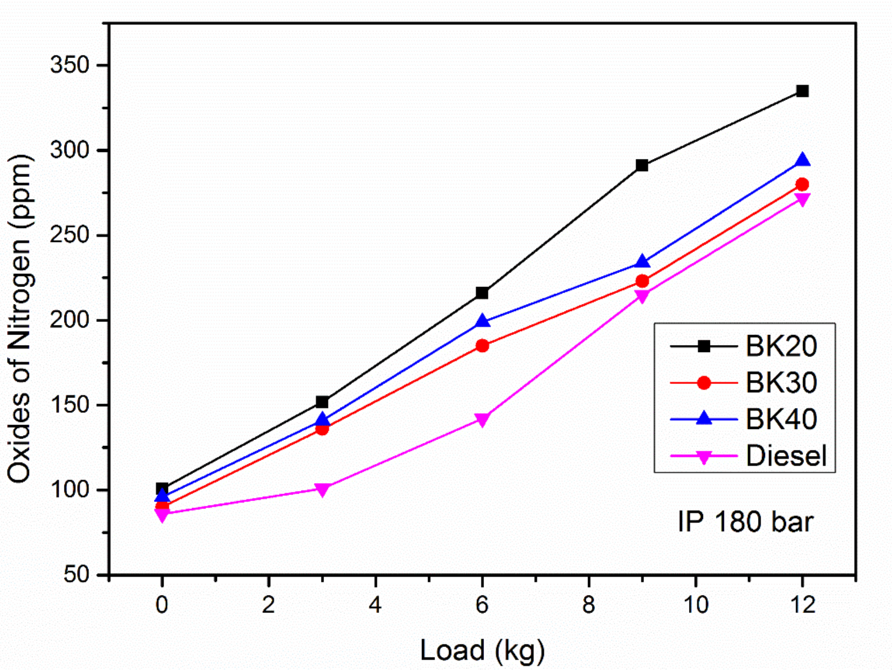


Fig. 6.16 Variation of NOx with load at 180 bar injection pressure

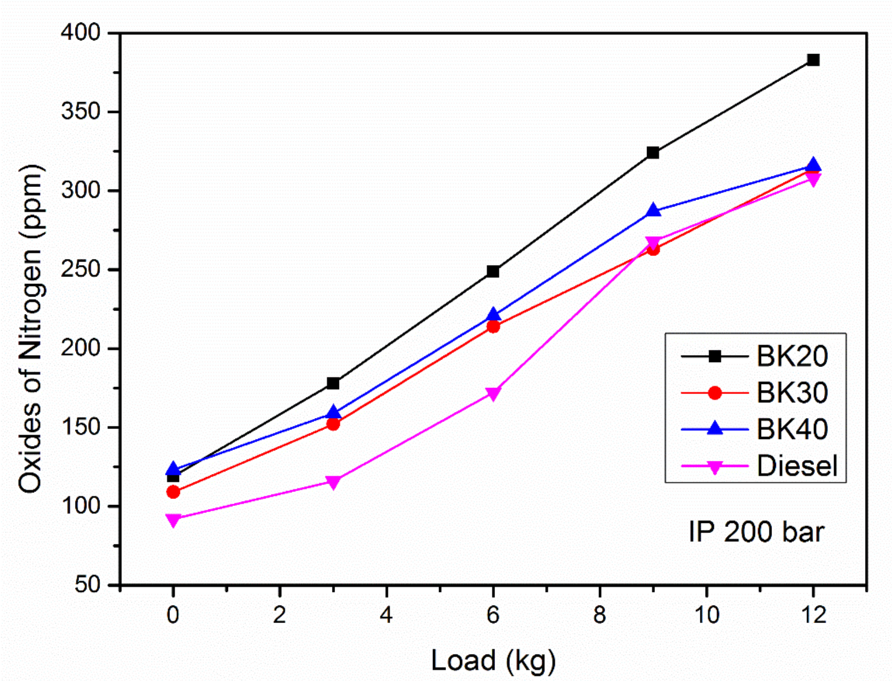


Fig. 6.17 Variation of NOx with load at 200 bar injection pressure

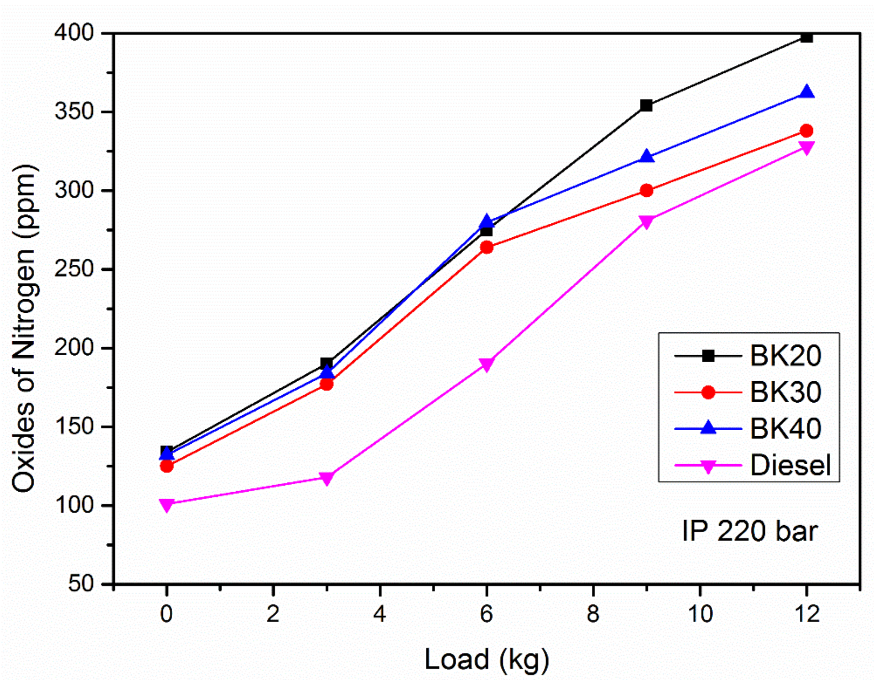


Fig. 6.18 Variation of NOx with load at 220 bar injection pressure

### 6.3.4 Smoke Emission

Variation of smoke emissions with load, for various blends of kerosene and Cardanol and diesel, at three different injection pressures (i.e., 180 bar, 200 bar and 220 bar) are shown in Figure 6.19, Figure 6.20 and Figure 6.21. As seen in these figures, the smoke opacity increases with the load for all the fuels tested. This is because of increase in the quantity of fuel injected with load. As the percentage of kerosene in the blend increased the smoke emission reduced because of fast burning of kerosene. At higher loads for BK30 and BK40 blends smoke emissions were lower than for the diesel fuel. Due to high combustion chamber temperature, at high loads, BK30 and BK40 blends burns faster than the diesel fuel. Also, it was noticed that when the injection pressure was increased the smoke emission reduces. When injection pressure was increased from 200 bar to 220 bar at peak load the smoke emission was reduced by 10%, 23%, 17.5% and 21.% for BK20, BK30, BK40 and diesel fuel, respectively. Increase in injection pressure reduces the injected fuel particles size, which leads to proper mixing with air and good combustion, so the smoke emission reduced. Similar opinions have been reported by Gumus et al. (2012).

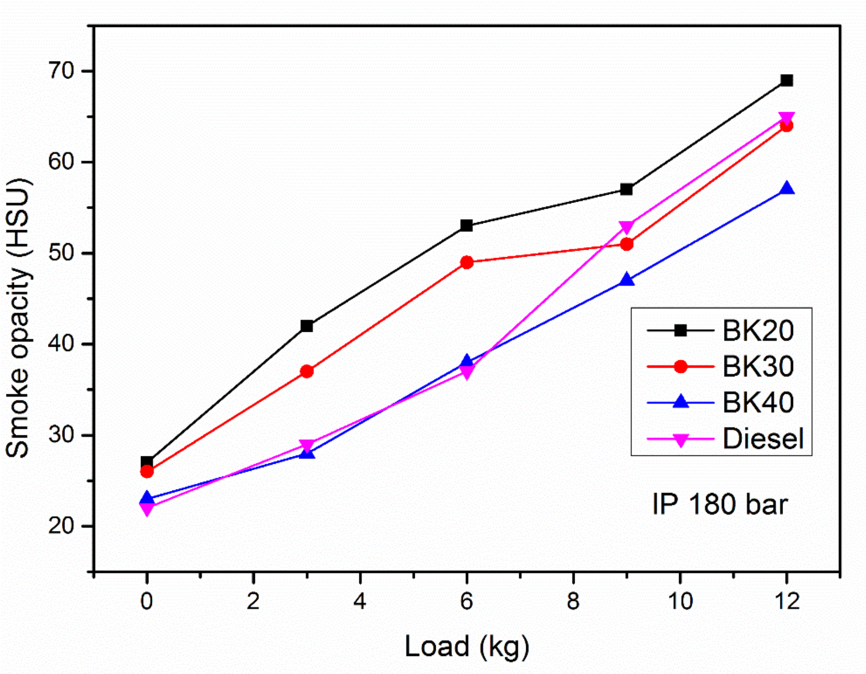


Fig. 6.19 Variation of smoke with load at 180 bar injection pressure

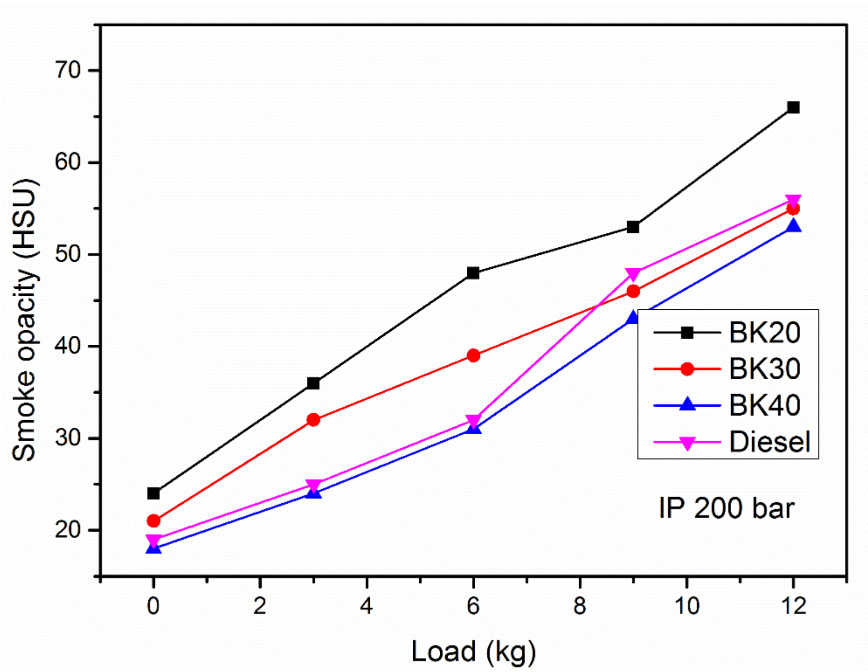


Fig. 6.20 Variation of smoke with load at 200 bar injection pressure

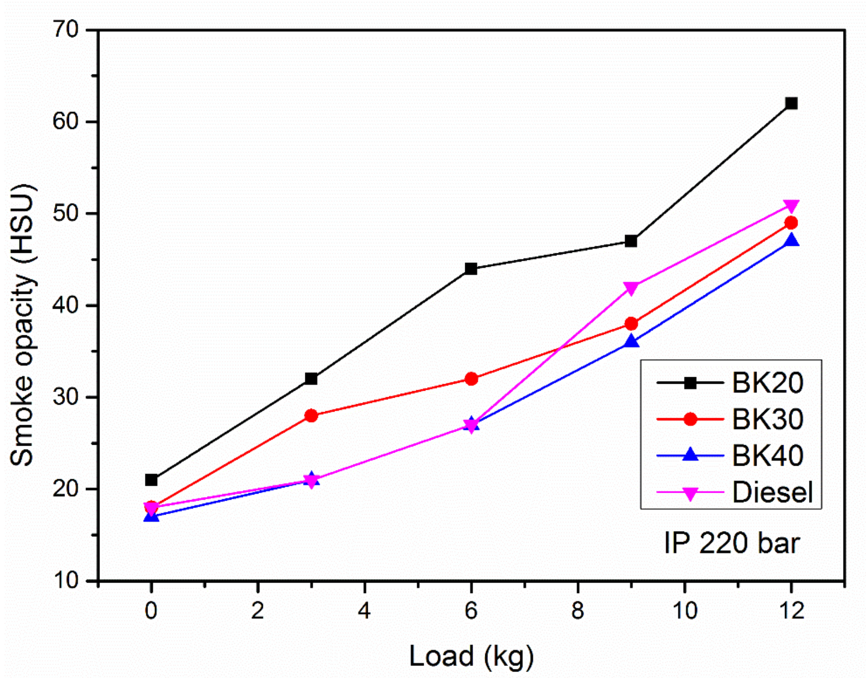


Fig. 6.21 Variation of smoke with load at 220 bar injection pressure

# CHAPTER 7

# EFFECT OF INJECTION TIMING ON PERFORMANCE AND EMISSIONS OF CARDANOL KEROSENE BLEND OPERATED DIESEL ENGINE

## 7.1 METHODOLOGY

The experimental investigations were carried out in a water cooled single cylinder direct injection four stroke diesel engine operated at 17:1 compression ratio, 200 bar injection pressure. A computer was connected to the engine for capturing data. Netel exhaust gas analyser (Model: NPM-MGA-1) and Netel smoke meter (model: (NPM-SM-111B) were used for measurement of exhaust gas emissions and smoke opacity.

The diesel fuel and test blends were filled separately to the dual fuel tank. Initial experiments were conducted at 23° BTDC injection timing and 200 bar injection pressure 17:1 compression ratio. Engine was started with diesel fuel by hand cranking and allowed in idling condition for warm up. The engine was loaded by adjusting the dynamometer. After the steady state fuel consumption rate was noted. Brake thermal efficiency, brake specific fuel consumption and exhaust gas temperature were recorded from the computer and the exhaust emissions were measured using exhaust gas analyser and smoke meter. The trail was repeated for the BK20 blend filled in the tank by changing the position of the fuel valve fitted in the fuel tank and then for BK30 and BK40 blend. The fuel blend in the tank was drained and rinsed with the new blend before filling it.

After that the tests were repeated for all the test fuel by varying the injection timing (21.5° BTDC and 24.5° BTDC). The injection timing was varied by changing the shim thickness in between the fuel pump and engine connection. By removing a shim of 0.25mm thickness the injection timing advanced by 1.5° crank angle and by adding a shim of 0.25mm thickness the timing was retarded by 1.5°crank angle. For each set of shim the injection timing was checked manually by the spill method. Appendix V indicates the experimental data for all the test fuel at various injection timings.

## 7.2 EFFECT OF INJECTION TIMING ON PERFORMANCE

Varying the injection timing affects the temperature and pressure inside the combustion chamber at the time of fuel injection. When the injection timing is advanced the combustion chamber pressure at the time of injection reduces, which alters the performance of the engine.

### 7.2.1 Brake Thermal Efficiency

Figure 7.1, Figure 7.2 and Figure 7.3 show the variation of brake thermal efficiency (BTE) with load for different kerosene and Cardanol blends and diesel at three different injection timings. As depicted in these figures, for all the test fuels there is an increase in BTE with increase in load. When the injection timing (IT) was retarded from 23° BTDC to 21.5° BTDC, at maximum load, the brake thermal efficiency was reduced by 13%, 3.9% and 3.3% for BK20, BK30 and BK40, respectively. Once IT was advanced from 23° BTDC to 24.5° BTDC, at full load, the BTE was increased by 4.4%, 4.1% and 3.8% for BK20, BK30 and BK40, respectively. When IT was advanced, longer time was available for better mixing of fuel and air, which enhances the combustion process and hence BTE increases with advanced IT (Ashok et al., 2017). For diesel fuel, at the rated injection timing (i.e., 23° BTDC) maximum BTE was obtained. Retarding and advancing the IT changes the peak pressure, which reduces the BTE. (Ganapathy et al., 2011).

### 7.2.2 Brake Specific Fuel Consumption

Figure 7.4, Figure 7.5 and Figure 7.6 illustrate the variation of brake specific fuel consumption (BSFC) with load for all the test fuels at different injection timings. As observed from these figures, when the injection timing (IT) was advanced the BSFC reduces for all the test fuels. When IT was retarded from 23° BTDC to 21.5° BTDC the BSFC increases for all the test fuels. Advancing IT from 23° BTDC to 24.5° BTDC the BSFC reduced for the bio fuel blends, and the reason is early start of combustion, due to which the peak pressure occurs very close to top dead centre (Raheman and Ghadge, 2008). But for diesel fuel by advancing IT the BSFC increases.

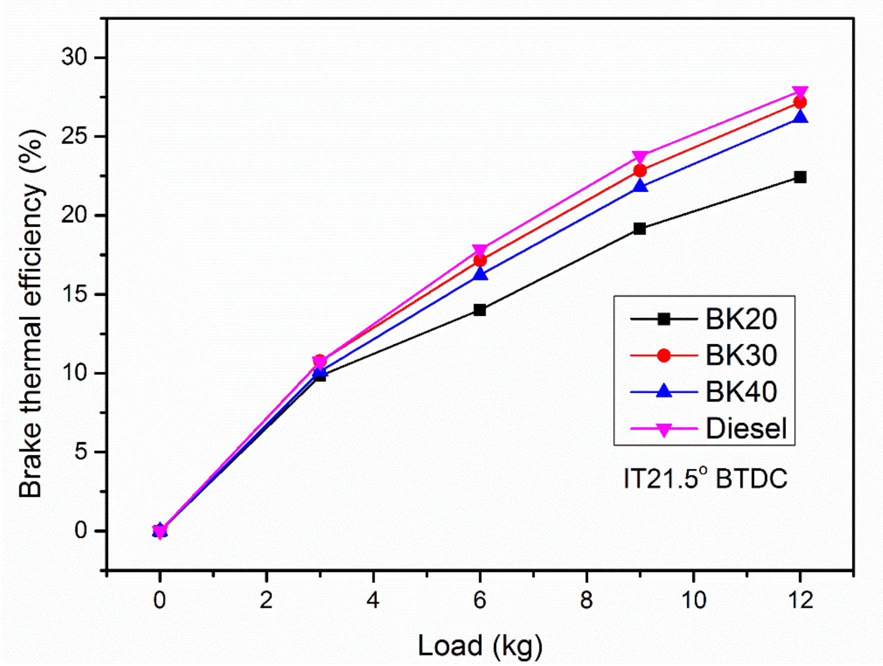


Fig. 7.1 Variation of BTE with load at 21.5° BTDC

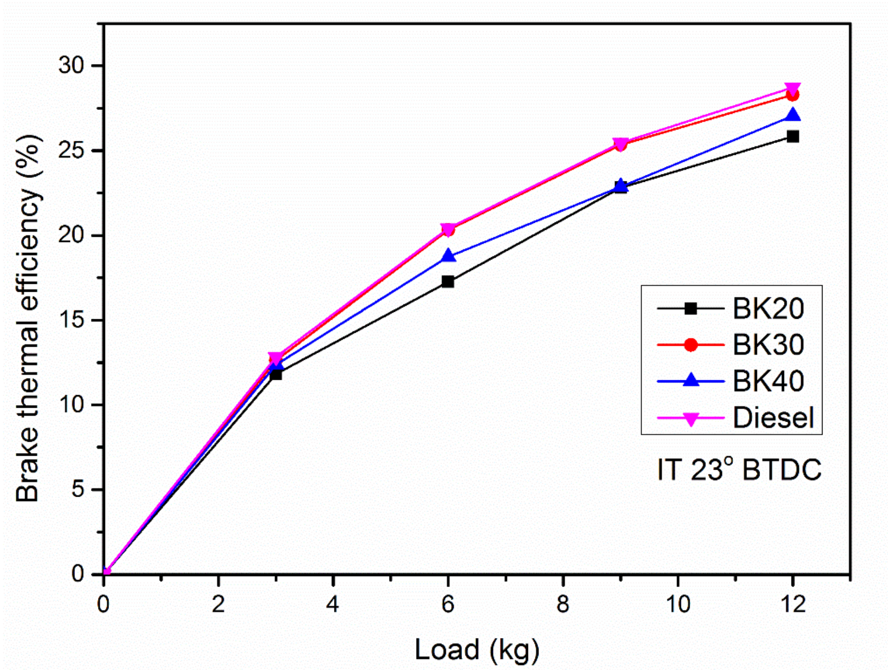


Fig. 7.2 Variation of BTE with load at 23° BTDC

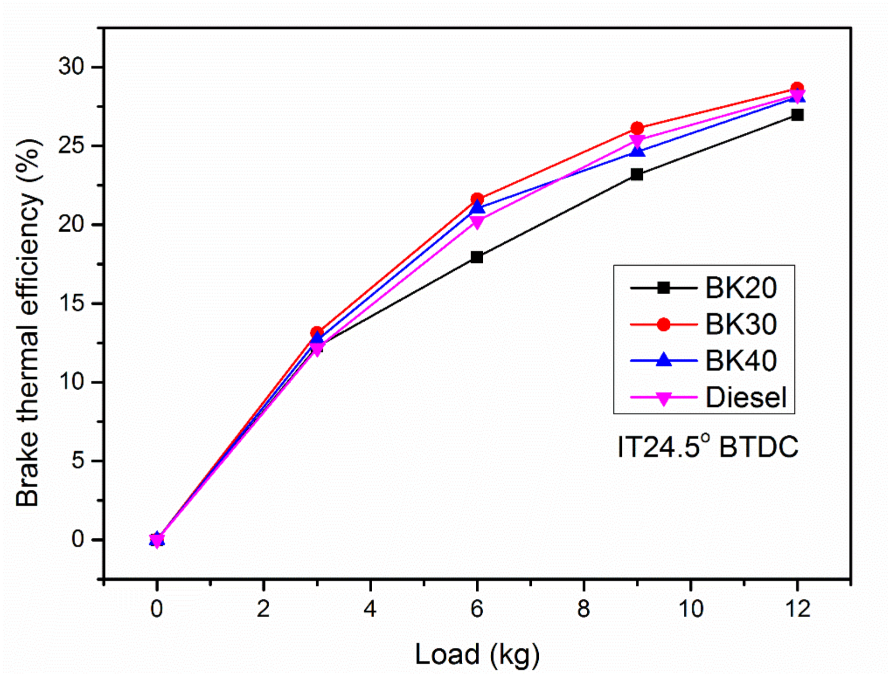


Fig. 7.3 Variation of BTE with load at 24.5° BTDC

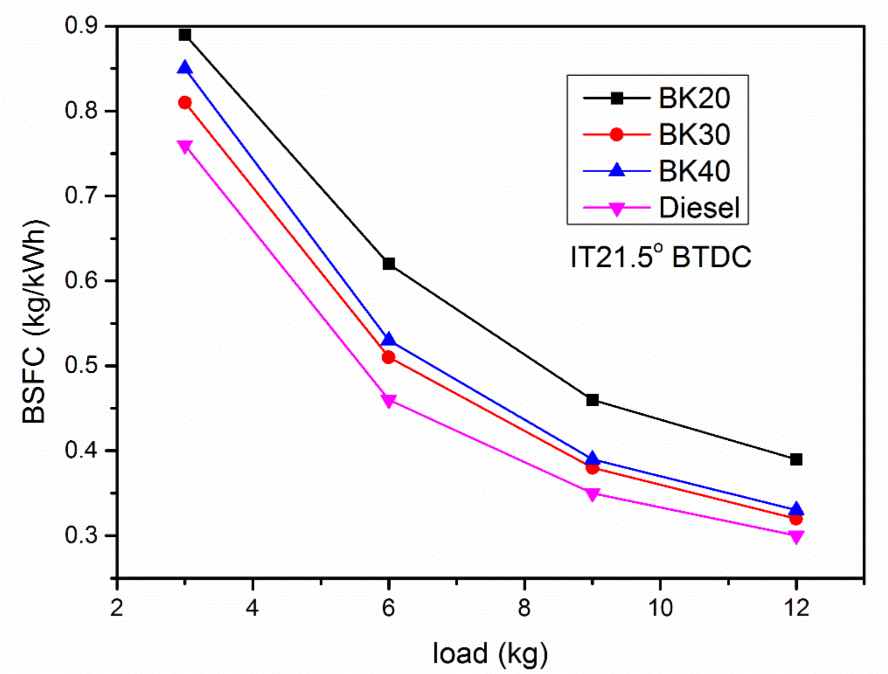


Fig. 7.4 Variation of BSFC with load at 21.5° BTDC

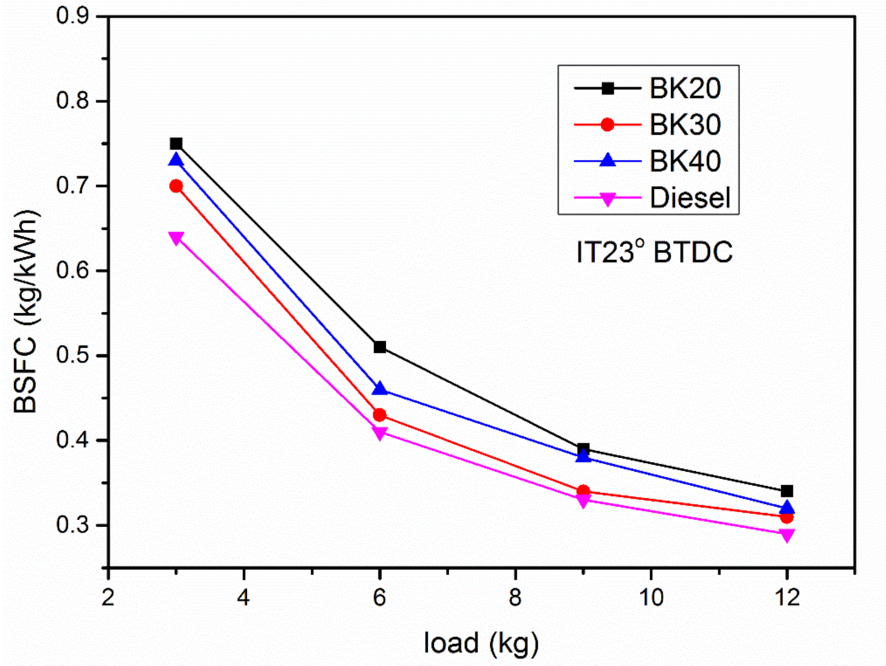


Fig. 7.5 Variation of BSFC with load at 23° BTDC

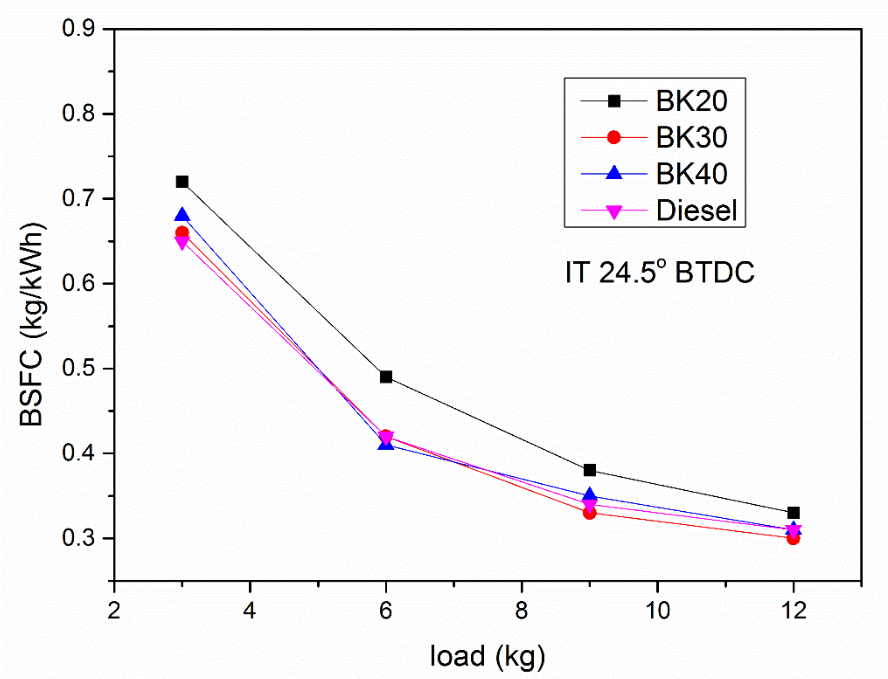


Fig. 7.6 Variation of BSFC with load at 24.5° BTDC

### 7.2.3 Exhaust Gas Temperature

The exhaust gas temperature variations for different test fuels at various loads at different injection timing are shown in Figure 7.7, Figure 7.8 and Figure 7.9. It was observed from graph that when injection timing advances the exhaust gas temperature reduces for all the test fuels. When the injection timing was advanced sufficient time is available for completion of combustion, which reduces the exhaust gas temperature. The exhaust gas also informs about the heat lost in the exhaust gas. Lower exhaust gas temperature was observed for BK30 blend compared to other blends at all injection pressures which was supported by higher BTE for the same blend.

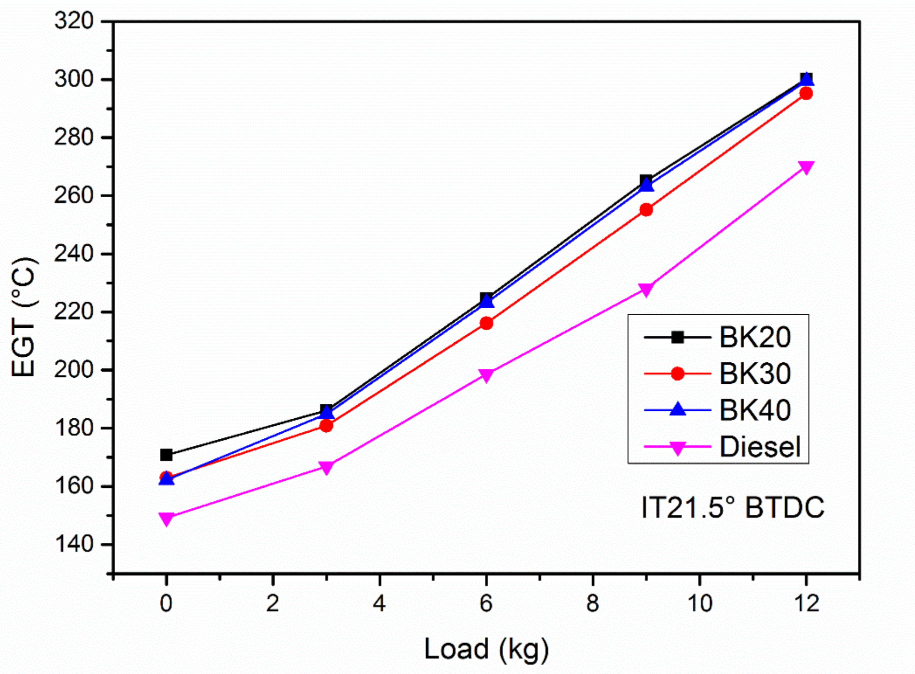


Fig. 7.7 Variation of EGT with load at 21.5° BTDC

## 7.3 EFFECT OF INJECTION TIMING ON EMISSIONS

When the injection timing is altered the time available for combustion and the temperature at the time of fuel injection changes. The emissions of the engine depends on the duration of combustion and the type of fuel used.

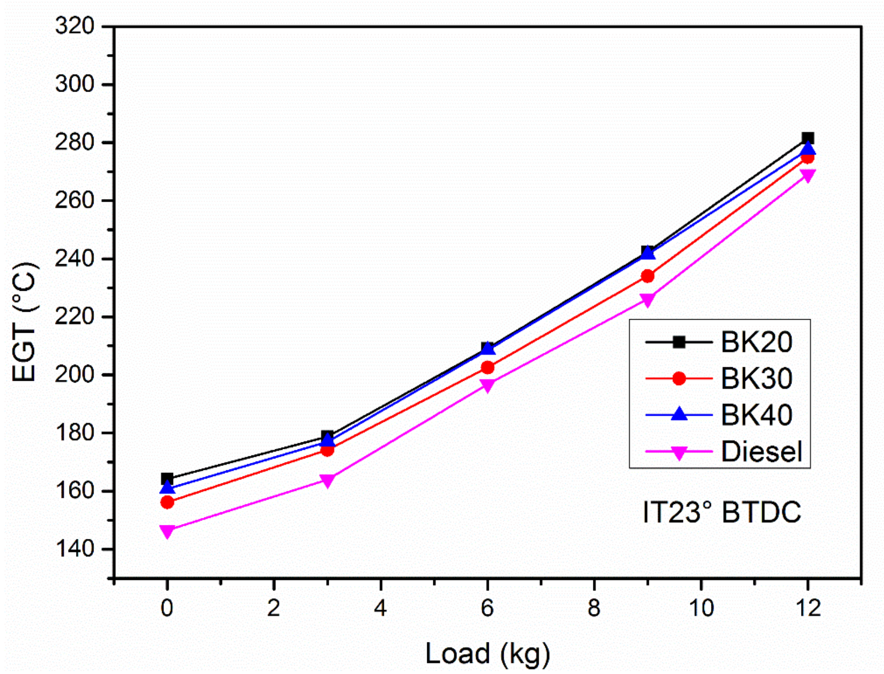


Fig. 7.8 Variation of EGT with load at 23° BTDC

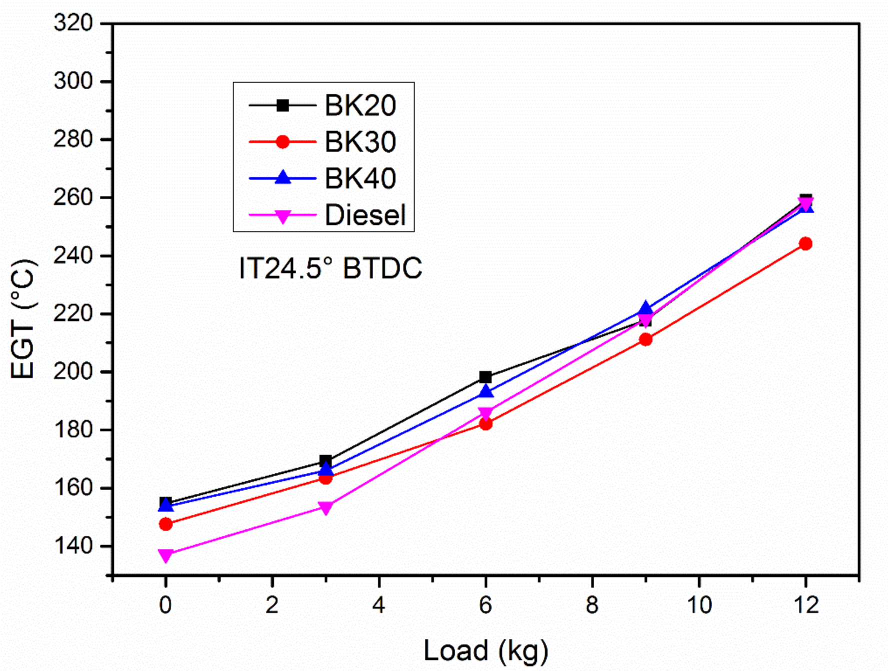


Fig. 7.9 Variation of EGT with load at 24.5° BTDC

### 7.3.1 Carbon Monoxide Emission

Figure 7.10, Figure 7.11 and Figure 7.12 depict emission of carbon monoxide (CO) at various loads for different test fuels at various injection timings. As observed from the figures, the CO emission is reduced for all the biofuel blends with increase in load, but for diesel fuel a reverse trend was observed. It was because of more diesel injection at elevated loads, which leads to partial combustion of some portion of fuel, hence CO emission increases (Nateshan, 2013). At the advanced IT the CO emission reduced because of early start of combustion. When the injection timing (IT) was advanced from 21.5° BTDC to 24.5° BTDC, at maximum load, the CO emission was reduced by 28%, 18% and 19% for BK20, BK30 and BK40, respectively. When the IT was retarded the CO emission increased since the combustion was incomplete due to delayed start.

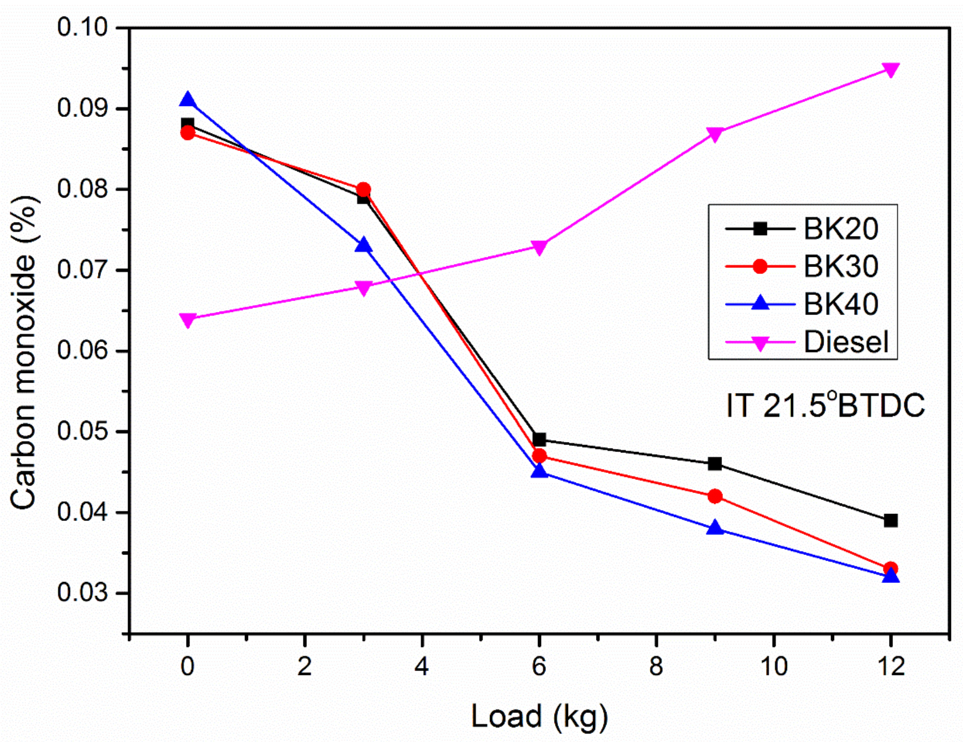


Fig. 7.10 Variation of CO with load at 21.5° BTDC

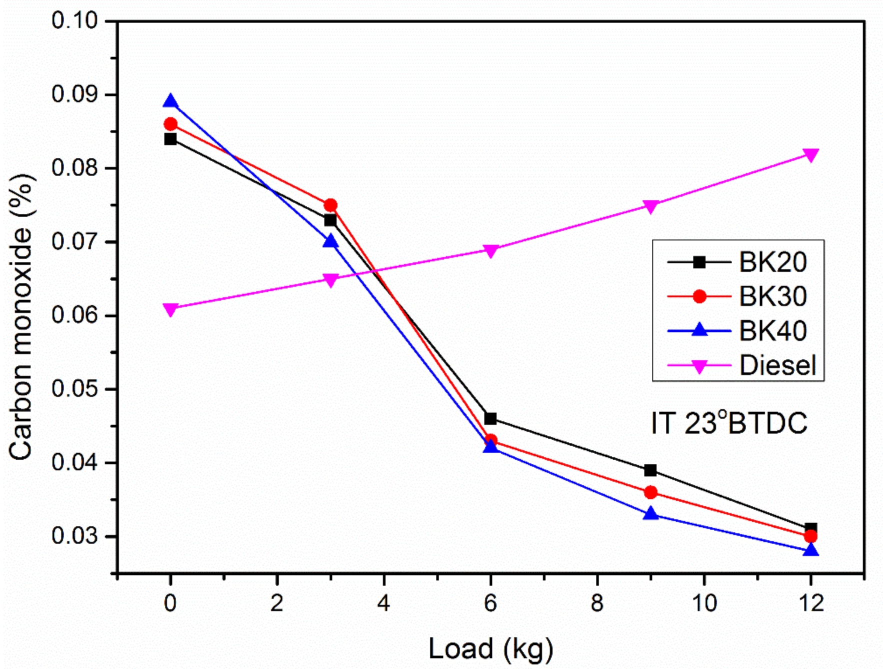


Fig. 7.11 Variation of CO with load at 23° BTDC

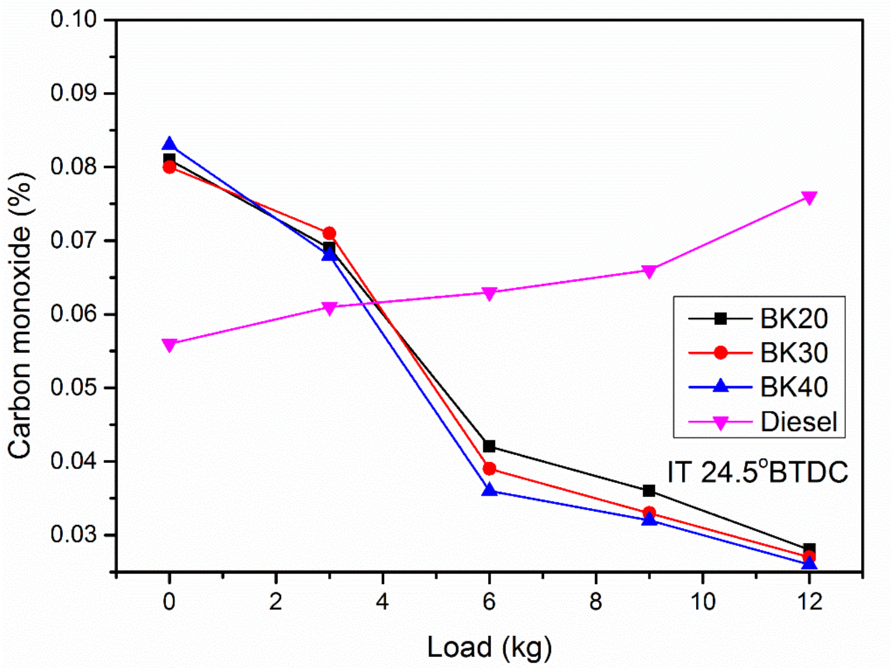


Fig. 7.12 Variation of CO with load at 24.5° BTDC

### 7.3.2 Unburned Hydrocarbon

The unburned hydrocarbons were emitted as a result of incomplete combustion. The HC emissions of all the test fuels at various loads for different injection timings are depicted in Figure 7.13, Figure 7.14 and Figure 7.15. As seen in these figures, for all the test fuels HC emission increases with load and this is because of more fuel injection at peak loads. The kerosene blends burns faster than diesel at higher loads due to higher volatility of kerosene. Because of this the HC emission was high for diesel at higher loads. When IT was advanced from 23° BTDC to 24.5° BTDC the HC emission was reduced. This is because the combustion commences early and is complete. But the HC emission increases with retardation in IT and this is due to late start in combustion process. Ashok et al. (2017) have resolved the similar findings for Calophllum inophyllum methyl ester. When the injection timing (IT) was advanced from 21.5° BTDC to 24.5° BTDC, at peak load the HC emission was reduced by 20%, 17.2%, 17.8% and 19% for BK20, BK30, BK40 and diesel fuel, respectively

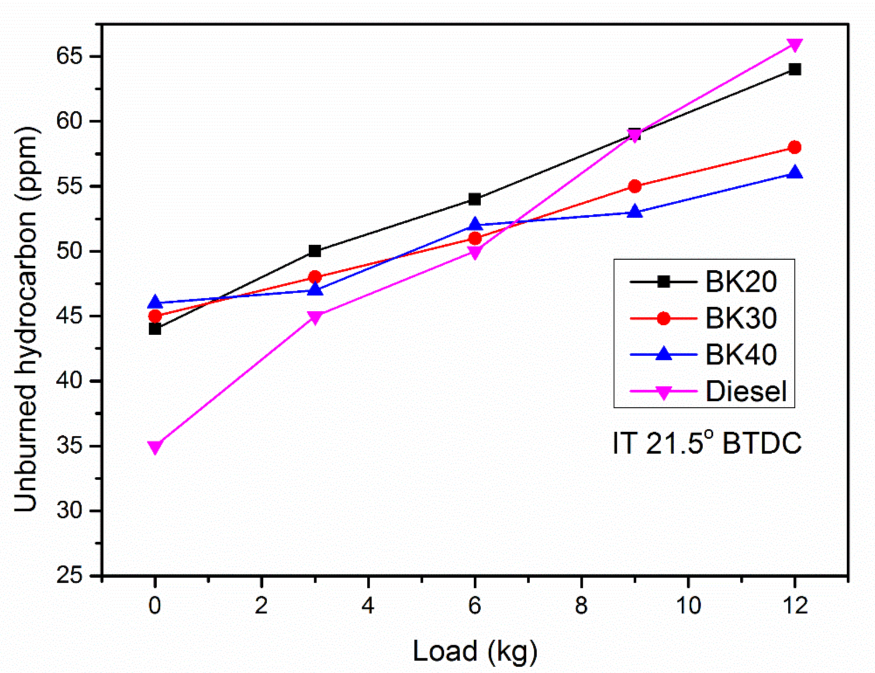


Fig. 7.13 Variation of HC with load at 21.5° BTDC

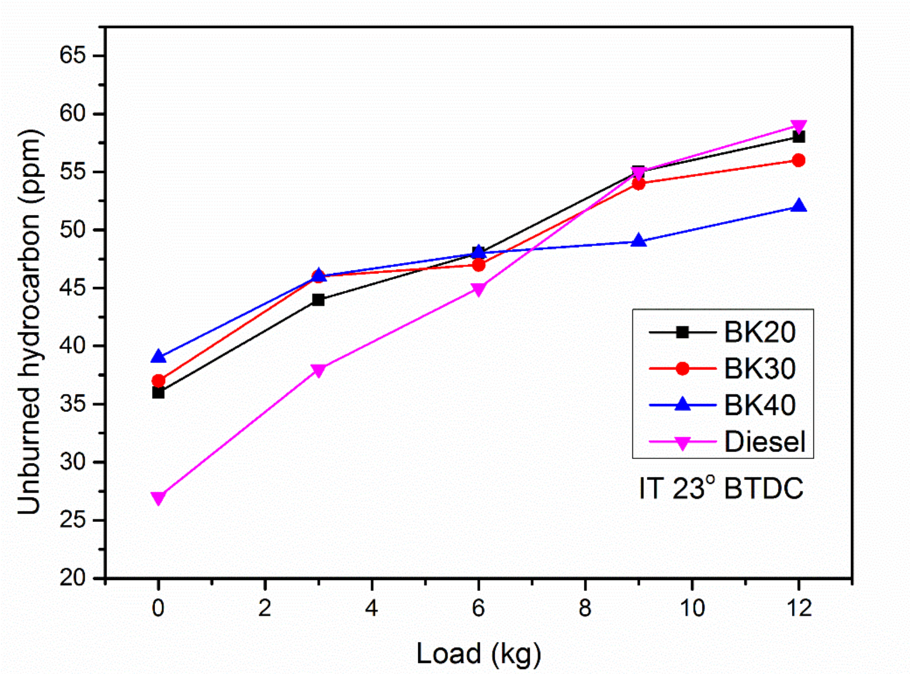


Fig. 7.14 Variation of HC with load at 23° BTDC

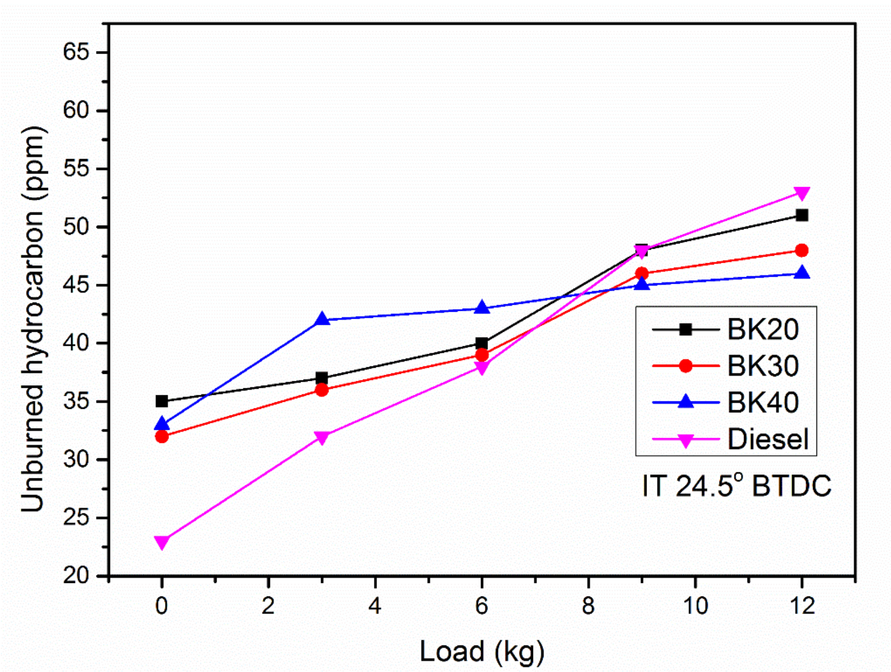


Fig. 7.15 Variation of HC with load at 24.5° BTDC

### 7.3.3 Oxides of Nitrogen Emission

Figure 7.16, Figure 7.17 and Figure 7.18 shows emissions of NOx at various loads for all the test fuels at different injection timings. By advancing injection timing from 21.5° to 24.5° BTDC the NOx emissions were increased for all the test fuels. When the injection timing (IT) was advanced from 21.5° BTDC to 24.5° BTDC, at maximum load, the NOx emission was increased by 20.7%, 21.2%, 24.3% and 21% for BK20, BK30, BK40 and diesel fuel, respectively By advancing the injection time more heat released at the premixed stage of combustion due to accumulation of fuel which leads to increased NOx emissions. At all the injection timing the biofuel blends emitted more NOx than the diesel fuel. Gnanasekaran et al. (2016) found similar observations with fish oil biodiesel blends.

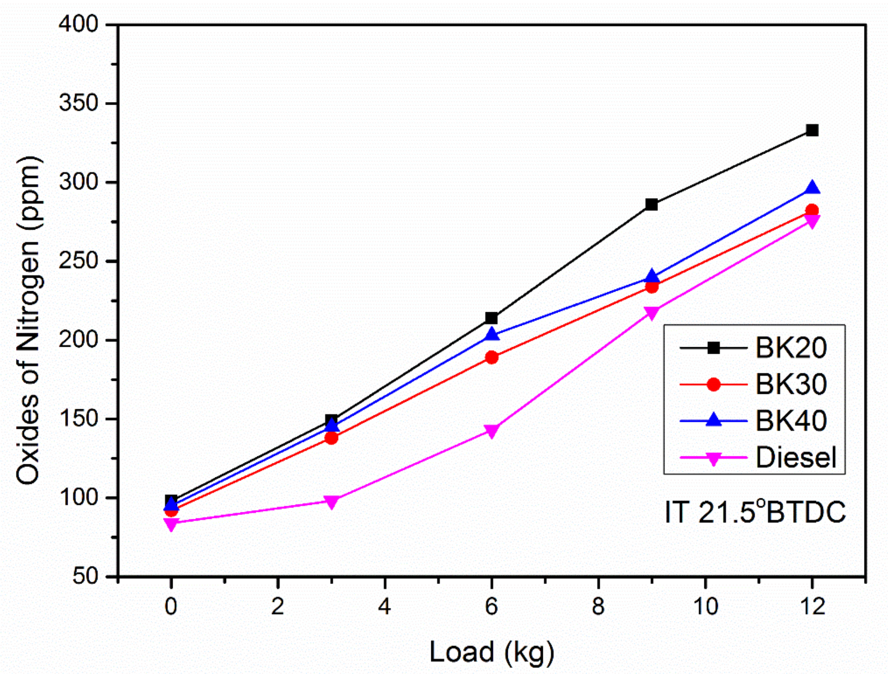


Fig. 7.16 Variation of NOx with load at 21.5° BTDC

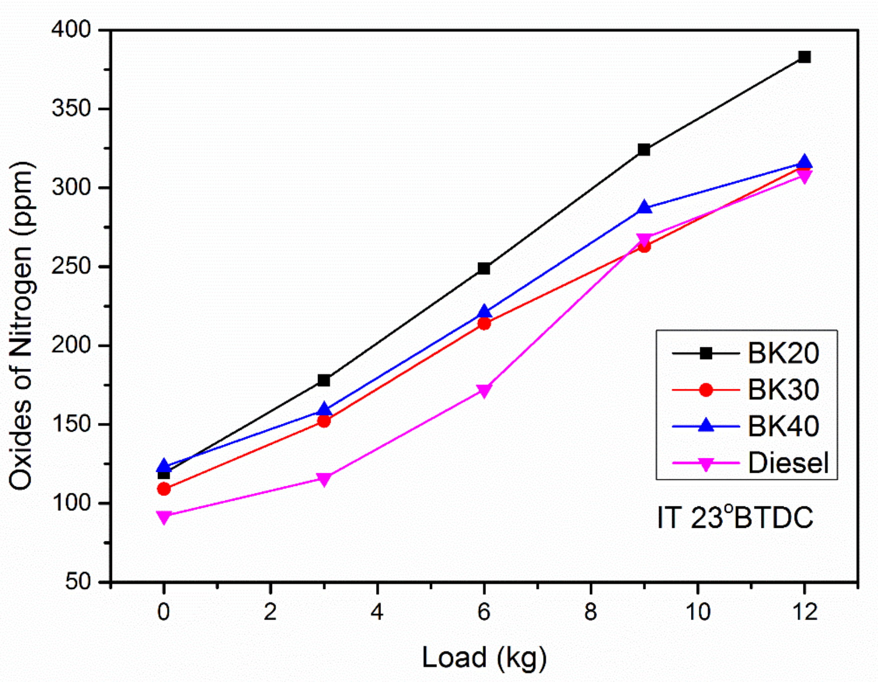


Fig. 7.17 Variation of NOx with load at 23° BTDC

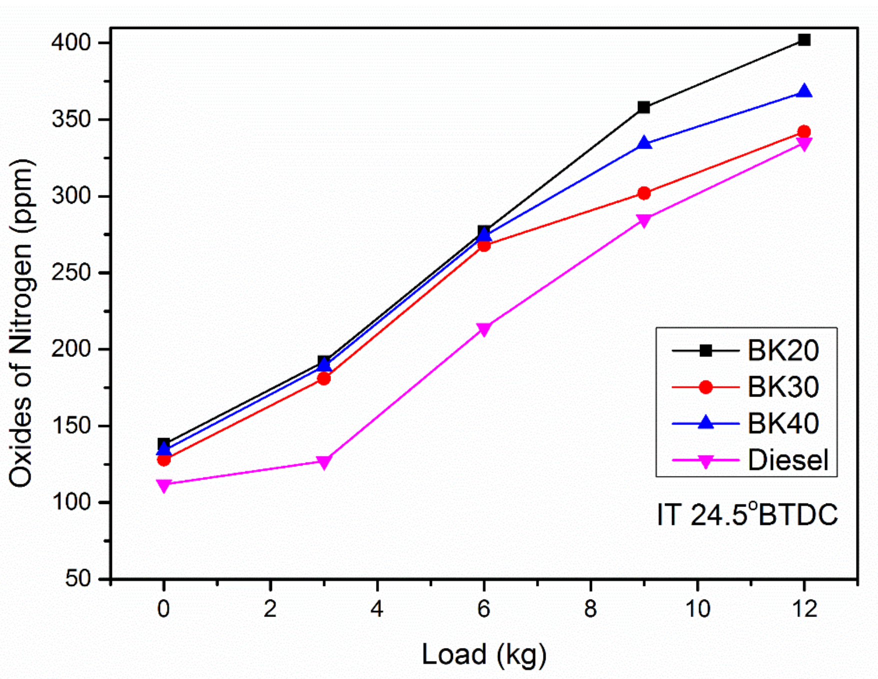


Fig. 7.18 Variation of NOx with load at 24.5° BTDC

### 7.3.4 Smoke Emission

Variations of smoke emissions at various loads for all the test fuels at different injection timing (IT) are shown in Figure 7.19, Figure 7.20 and Figure 7.21. As seen from the figures, by advancing the IT from 23°BTDC to 24.5°BTDC the smoke has been reduced by 10%, 3.6%, 8.1% and 3.5% for BK20, BK30, BK40 and diesel fuel respectively. When the IT was advanced more time will be available for combustion and soot particles were oxidised, so less smoke emissions. Similar results has been reported by Kannan and Anand, (2012). When IT was retarded from 23°BTDC to 21.5°BTDC the smoke was increased by 13%, 10%, 8.6% and 8.9% for BK20, BK30, BK40 and diesel fuel respectively.

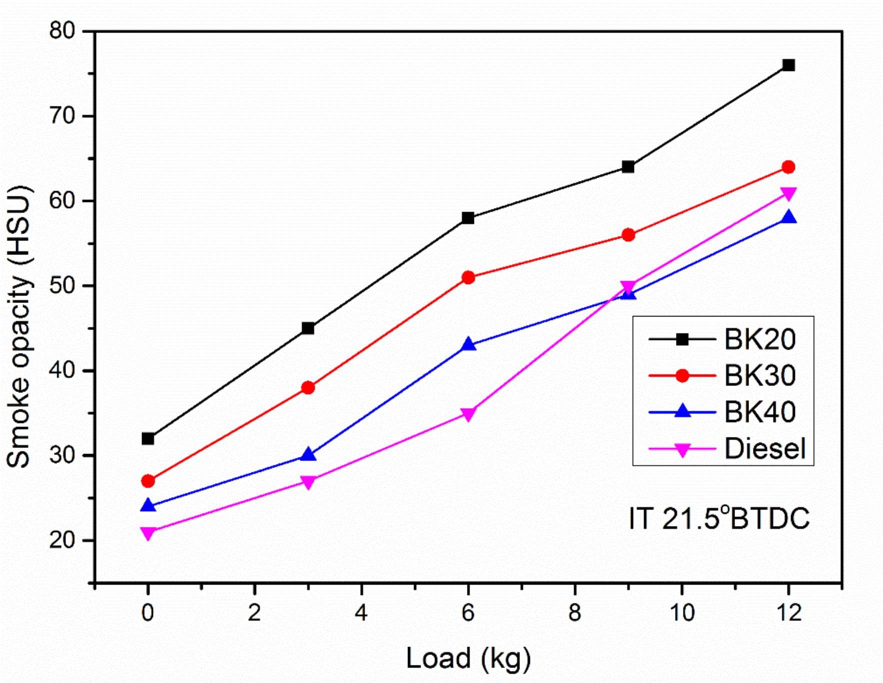


Fig. 7.19 Variation of smoke with load at 21.5° BTDC

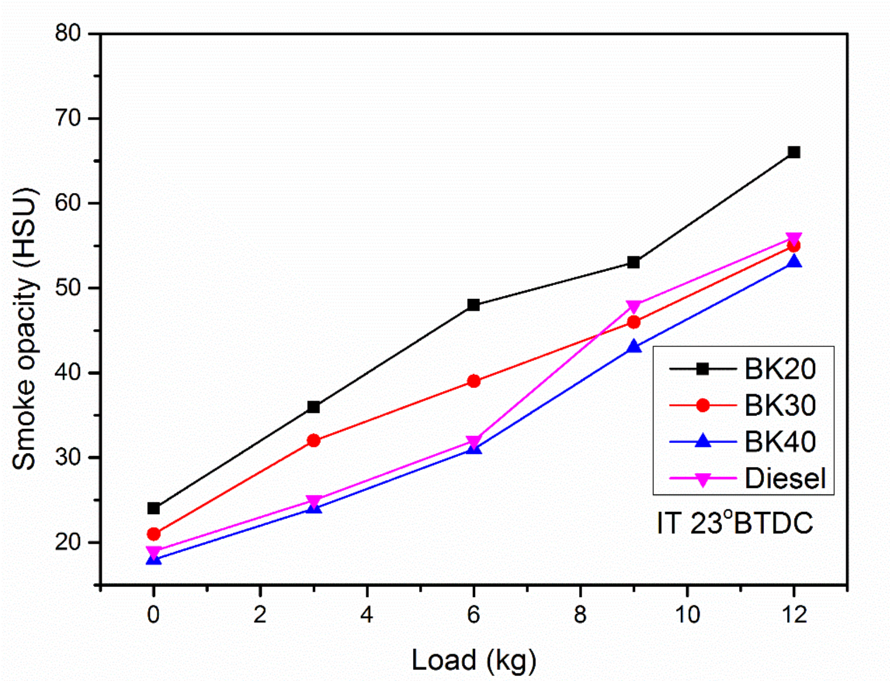


Fig. 7.20 Variation of smoke with load at 23° BTDC

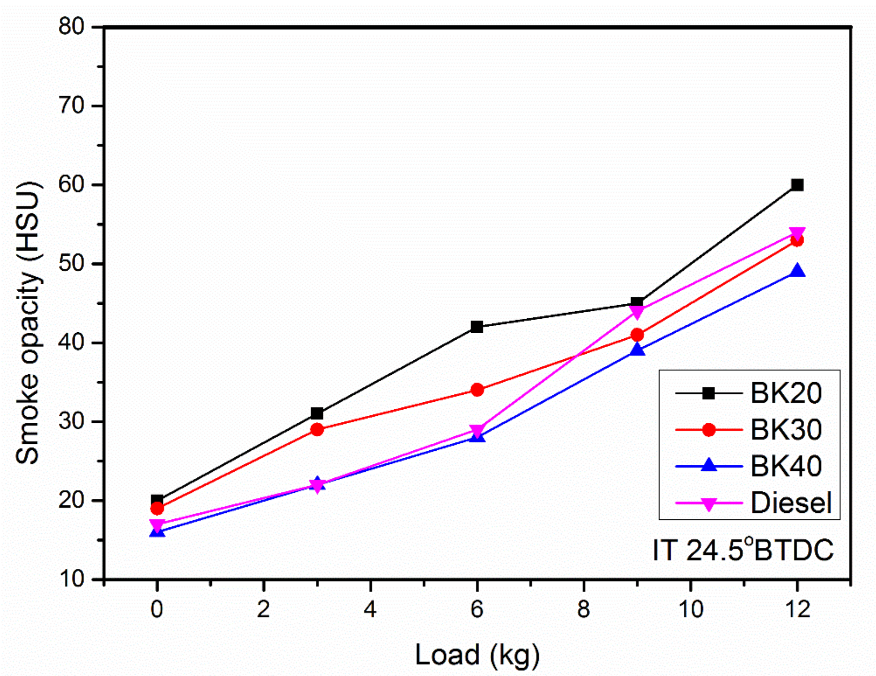


Fig. 7.21 Variation of smoke with load at 21.5° BTDC

# CHAPTER 8

# OPTIMISATION OF DIESEL ENGINE PERFORMANCE AND EMISSION PARAMETERS

## 8.1 INTRODUCTION

The engine parameters are optimised using signal to noise (S/N) ratio analysis and regression analysis. The analysis of variance (ANOVA) is carried out to know the significance of variables on the responses. The various responses are predicted with the equation generated by regression analysis. The experimental values at the optimum operating condition are compared with the predicted values to know the error in the prediction.

## 8.2 Taguchi Method

This is a method of designing the experimental process. The method involves an orthogonal array (Kokkulunk et al. 2014) with the parameters affecting the process and its levels of variation. Selection of the orthogonal array is based on the number of factors and the levels. The factors involved in the analysis and the levels are represented in Table 8.1. Once the factors affecting the experimental process are recognised, based on the number of levels the array is selected. For this design with five factors and three levels, $L_{27}$ array is selected. Various combinations of parameters of the $L_{27}$ design are shown in Table 8.2

Table 8.1 Levels of design parameters

| Factors | Level 1 | Level 2 | Level 3 |
|---|---|---|---|
| Blend % volume (A) | 20 | 30 | 40 |
| Compression Ratio (B) | 16:1 | 17:1 | 18:1 |
| Injection Pressure in bar (C) | 180 | 200 | 220 |
| Injection Timing in °BTDC (D) | 21.5 | 23 | 24.5 |
| Load in kg (E) | 6 | 9 | 12 |

The experimental results are arranged as per the design shown in the Table 8.2. Then analysis is done with Minitab 17 software. (https://www.apponfly.com/en/minitab)

Table 8.2 Array of $L_{27}$ design

| Run Number | Blend (A) | Compression Ratio (B) | Injection Pressure (C) | Injection Timing (D) | Load (E) |
|---|---|---|---|---|---|
| 1 | 20 | 16 | 180 | 21.5 | 6 |
| 2 | 20 | 16 | 180 | 21.5 | 9 |
| 3 | 20 | 16 | 180 | 21.5 | 12 |
| 4 | 20 | 17 | 200 | 23 | 6 |
| 5 | 20 | 17 | 200 | 23 | 9 |
| 6 | 20 | 17 | 200 | 23 | 12 |
| 7 | 20 | 18 | 220 | 24.5 | 6 |
| 8 | 20 | 18 | 220 | 24.5 | 9 |
| 9 | 20 | 18 | 220 | 24.5 | 12 |
| 10 | 30 | 16 | 200 | 24.5 | 6 |
| 11 | 30 | 16 | 200 | 24.5 | 9 |
| 12 | 30 | 16 | 200 | 24.5 | 12 |
| 13 | 30 | 17 | 220 | 21.5 | 6 |
| 14 | 30 | 17 | 220 | 21.5 | 9 |
| 15 | 30 | 17 | 220 | 21.5 | 12 |
| 16 | 30 | 18 | 180 | 23 | 6 |
| 17 | 30 | 18 | 180 | 23 | 9 |
| 18 | 30 | 18 | 180 | 23 | 12 |
| 19 | 40 | 16 | 220 | 23 | 6 |
| 20 | 40 | 16 | 220 | 23 | 9 |
| 21 | 40 | 16 | 220 | 23 | 12 |
| 22 | 40 | 17 | 180 | 24.5 | 6 |
| 23 | 40 | 17 | 180 | 24.5 | 9 |
| 24 | 40 | 17 | 180 | 24.5 | 12 |
| 25 | 40 | 18 | 200 | 21.5 | 6 |
| 26 | 40 | 18 | 200 | 21.5 | 9 |
| 27 | 40 | 18 | 200 | 21.5 | 12 |

### 8.2.1 Signal to Noise Ratio Analysis

One method of optimisation is by using the signal to noise ratio analysis. The S/N ratio of the parameters are optimised based on smaller the better or larger the better responses. The analytical representation of S/N ratio for smaller is the better as below.

$$S/N = -10 \log \left[\frac{1}{n}\sum_{i=1}^{n} {Y_i}^2\right] \quad (8.1)$$

The S/N ratio for larger is better can be represented as below.

$$S/N = -10 \log \left[\frac{1}{n}\sum_{i=1}^{n} 1/{Y_i}^2\right] \quad (8.2)$$

Where Y is the measured value of the response variable and ‘n’ number of observations.

### 8.2.2 Regression Analysis

The experimental data collected from all sets of input parameters from the taguchi design are analysed using Minitab 17 software. Regression analysis is executed to obtain prediction expression for the individual responses corresponding to the input parameters. Prediction expression for performance output parameters like brake thermal efficiency, brake specific fuel consumption, exhaust gas temperature and the output emission parameters like CO emission, HC emission, NOx emission and smoke emission are developed. The effectiveness of regression model in represented by regression coefficient $R^2$.

$$R^2 = \frac{\sum_{i=1}^{n}(y_i - \hat{y}_i)^2}{\sum_{i=1}^{n}(y_i - \bar{y}_i)^2} \quad (8.3)$$

Where $\bar{y}_i$ $y_i$ $\hat{y}_i$ are predicted value, measured value and mean value of response, respectively. The value of the regression coefficient lies between zero and one. Higher value of regression coefficient indicates perfectness of the fit value with the data. The experimental values and the predicted values from the regression model are plotted to check the linear variations of the values. The suitability of regression model is checked by the residual plots. The residual value is the difference between experimental values and the predicted values from the model. Residual plots are the graphs plotted with the residuals in vertical axis and the variable on horizontal axis. The random dispersion of the scatter point about the horizontal axis represents the appropriation of the regression model with the data (Lee et al., 2013).

### 8.2.3 Analysis of Variance (ANOVA)

ANOVA is a statistical method of finding the impact of individual input parameters on the output response. The parameters like sum of squares (SS), degree of freedom (DF), mean square (MS) and probability value (p -value) are evaluated in ANOVA. Based on the p- value the significant parameters in the model are identified. If the p-value is more than 0.05 for any parameter then it is an insignificant parameter in the model.

## 8.3 PREDICTION AND OPTIMISATION OF PARAMETERS

Using the S/N ratio analysis the optimum levels of input parameters for all the responses are obtained. Then individual responses are predicted with the regression models and compared with the experimental value. For the optimised condition experiments are conducted and compared with predicted value to have an estimate of error.

### 8.3.1 Brake Thermal Efficiency

Brake thermal efficiency (BTE) of engine varies with the variations in fuel, compression ratio, injection pressure, injection timing and load. The equation (8.4) predicts the BTE for various inputs.

$$BTE = -383+1.654\times A +37.70\times B +0.0729\times C +1.033\times D +1.2813\times E -0.02587\times (A\times A) -1.068\times (B\times B) \tag{8.4}$$

The Figure 8.1 shows the comparison of the predicted values using the model equation and the experimental values. The residual plots of BTE is depicted in Figure 8.2. The residuals are randomly scattered around the horizontal axis, which infers that the model is suitable for the data. The predicted values of brake thermal efficiency is very close to the experimental values as seen from the Figure 8.1 with a predicted $R^2$ of 0.98.

ANOVA results for BTE is represented in Table 8.3, which shows the significant factors with their level of significance. From the analysis results, it is observed that the BTE is influenced by fuel blend, compression ratio, injection pressure, injection timing and load. The $R^2$ value for the analysis is 0.9693, with a randomly distributed residual plot shown in Figure 8.2, which indicates the highest fitness of the predicted value.

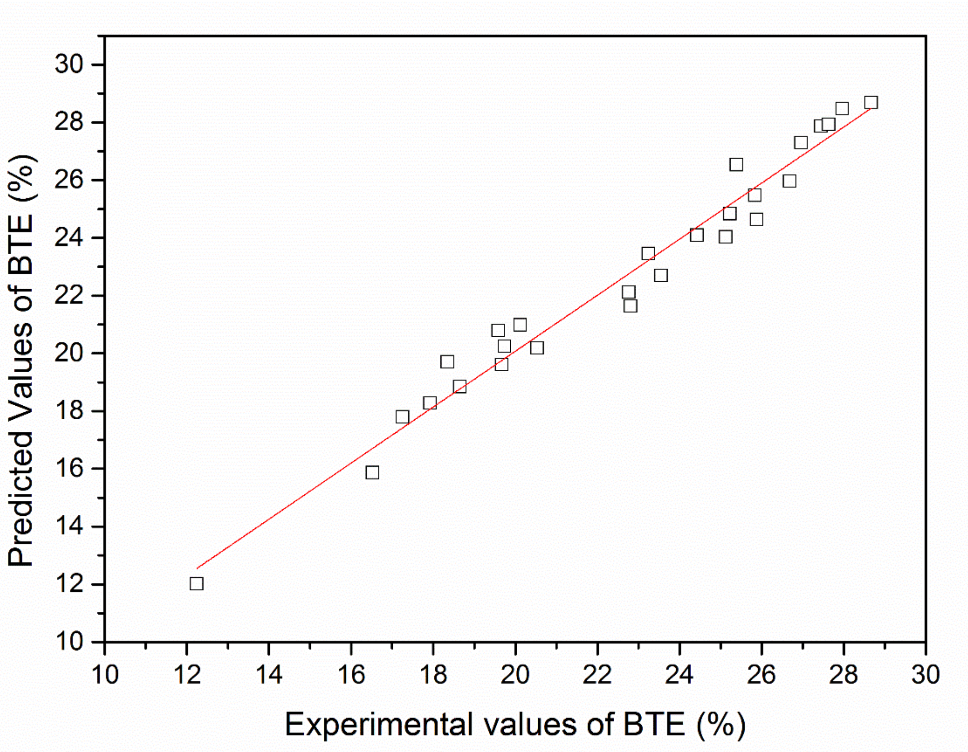


Fig. 8.1 Comparison of predicted values and experimental values of BTE

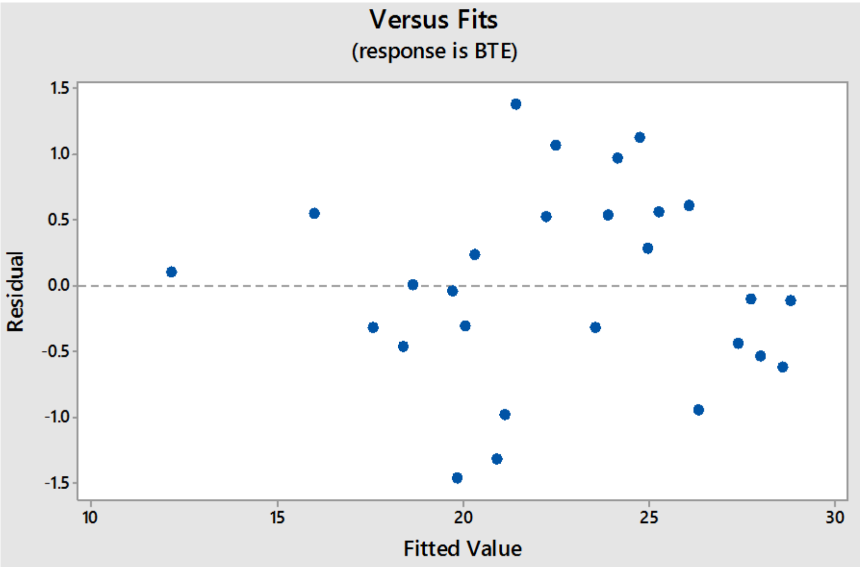


Fig. 8.2 Residual plot for BTE

Table 8.3 ANOVA results for brake thermal efficiency

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 45.183 | 45.1830 | 0.000 |
| Compression ratio (B) | 1 | 7.375 | 7.3750 | 0.000 |
| Injection pressure (C) | 1 | 38.281 | 38.2812 | 0.005 |
| Injection timing (D) | 1 | 43.214 | 43.2140 | 0.000 |
| Load(E) | 1 | 265.959 | 16.4936 | 0.000 |
| A*A | 1 | 40.145 | 40.145 | 0.000 |
| B*B | 1 | 6.848 | 6.8480 | 0.007 |
| Error | 19 | 14.181 | | |
| Total | 26 | 461.537 | | |

Figure 8.3 represents the signal to noise ratio plots of brake thermal efficiency for all the factors considered in the analysis. For optimising the brake thermal efficiency larger is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition (Zhan-Yi Wu et al. 2014; Balki et al. 2016). From the Figure 8.3 the optimum combination of parameter for BTE is 30% blend (BK30), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 12kg load. Experiments at this combination of parameters were conducted and obtained maximum thermal efficiency of 31.64%, which is higher than other blends.

### 8.3.2 Brake Specific Fuel Consumption

The brake specific fuel consumption (BSFC) is predicted using the equation (8.5). The parameters influencing BSFC are compression ratio, injection pressure, injection timing and load.

$$BSFC = 8.39 - 1.676 \times B - 0.06594 \times C + 1.224 \times D - 0.0831 \times E + 0.0261 \times B \times B - 0.02716 \times D \times D + 0.00327 \times E \times E + 0.003778 \times B \times C \tag{8.5}$$

Figure 8.4 depicts the variation of predicted values of BSFC using the model equation and the experimental values and the Figure 8.5 illustrates residual plot of it. The predicted values of BSFC agree with the experimental values with a predicted $R^2$ of

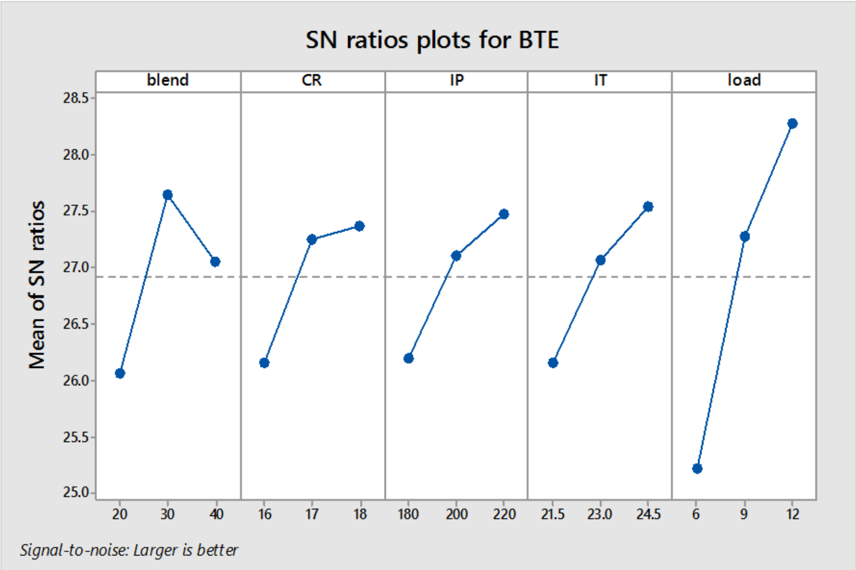


Fig. 8.3 The S/N ratios of BTE for all parameters

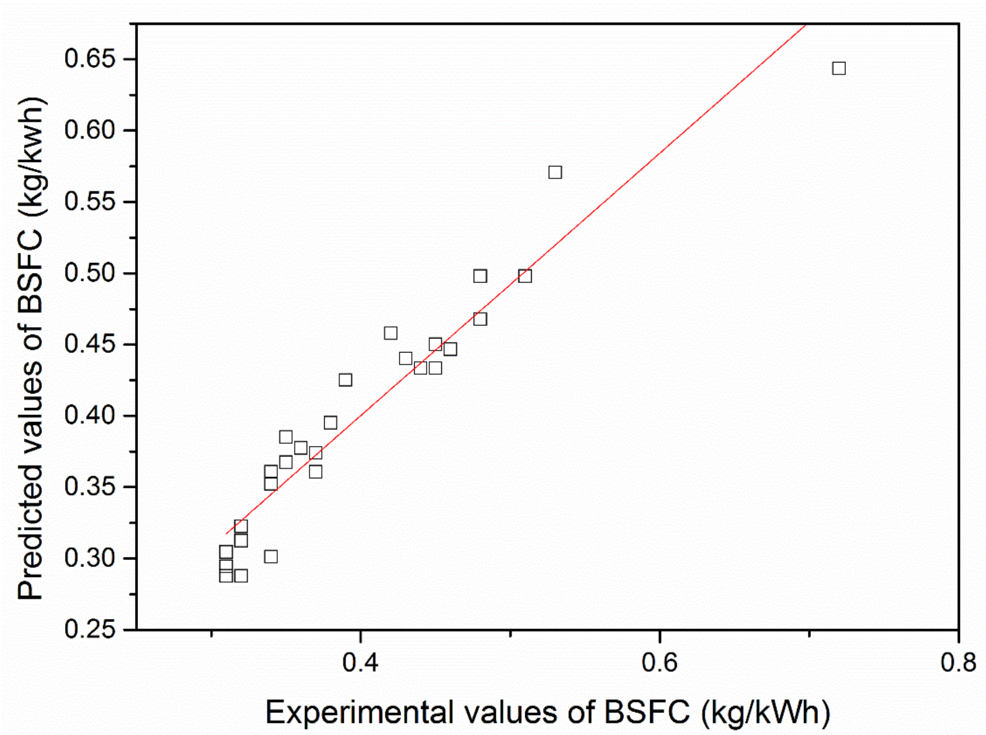


Fig. 8.4 Comparison of predicted values and experimental values of BSFC

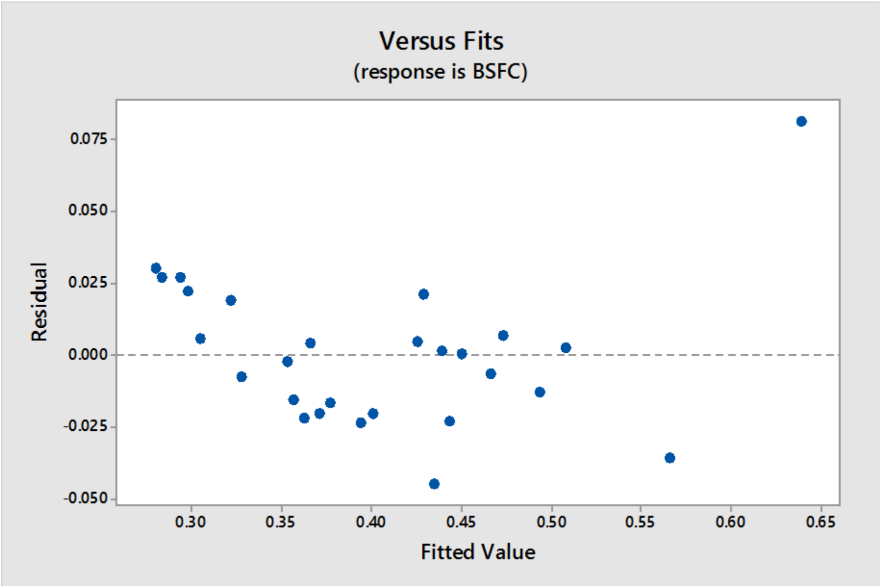


Fig. 8.5 Residual plot for BSFC

0.95. The residuals are also randomly dispersed about the horizontal axis which justifies the fitness of the model with the data.

ANOVA results for BSFC are shown in Table 8.4, which indicates various factors influencing BSFC with their level of significance. The analysis shows that the BSFC is influenced by compression ratio, injection pressure, injection timing and load. The $R^2$ value for the analysis is 0.9438.

Figure 8.6 indicates the signal to noise ratio plots of brake specific fuel consumption of all the factors considered in the analysis. For optimising the brake specific fuel consumption smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition. The optimum combination of parameter for BSFC is 30% blend (BK30), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 12kg load.

Table 8.4 ANOVA results for brake specific fuel consumption

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Compression ratio (B) | 1 | 0.013417 | 0.013417 | 0.000 |
| Injection pressure (C) | 1 | 0.036072 | 0.036072 | 0.000 |
| Injection timing (D) | 1 | 0.010755 | 0.010755 | 0.001 |
| Load(E) | 1 | 0.010275 | 0.010275 | 0.001 |
| B*B | 1 | 0.004091 | 0.004091 | 0.025 |
| D*D | 1 | 0.011204 | 0.011204 | 0.001 |
| E*E | 1 | 0.005202 | 0.005202 | 0.013 |
| B*C | 1 | 0.034252 | 0.034252 | 0.000 |
| Error | 18 | 0.012289 | | |
| Total | 26 | 0.218807 | | |

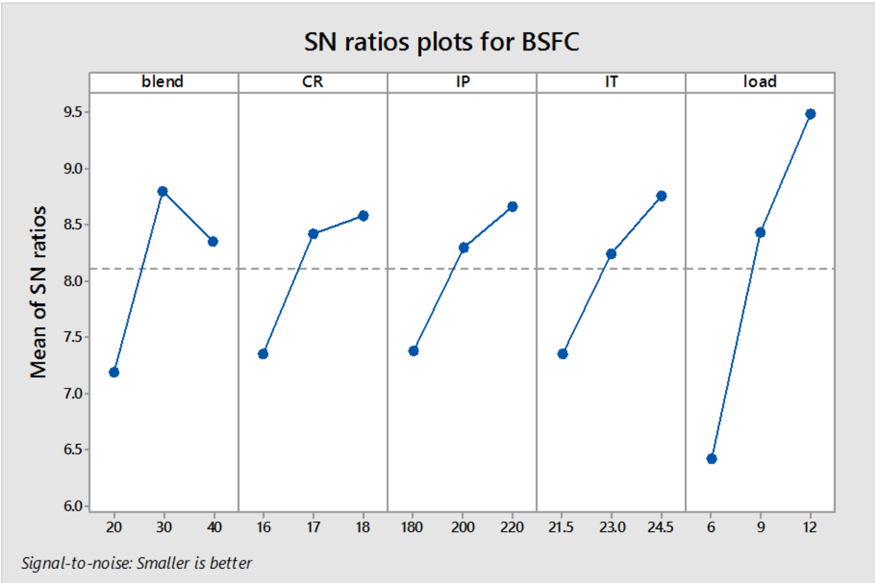


Fig. 8.6 The S/N ratios of BSFC for all parameters

### 8.3.3 Exhaust Gas Temperature

The exhaust gas temperature (EGT) can be predicted by using equation (8.6) obtained from regression modelling in terms of fuel blend, compression ratio, injection pressure, injection timing and load.

$$EGT = 811.4 - 4.90 \times A - 11.75 \times B - 0.7050 \times C - 11.608 \times D + 12.079 \times E + 0.0795 \times A \times A \quad (8.6)$$

The predicted values of EGT using the model equation and the experimental values are shown in Figure 8.7. It is observed that the predicted EGT values are obeying the experimental values with a high predicted $R^2$ value of 0.99. The residual plot of EGT is depicted in Figure 8.8. The residuals are randomly scattered around the horizontal axis, which indicates that the analysed model is suitable for the data.

ANOVA results for EGT is represented in Table 8.5.It is observed that the EGT is influenced by fuel blend, compression ratio, injection pressure, injection timing and load. The $R^2$ value for the analysis is 0.9793.

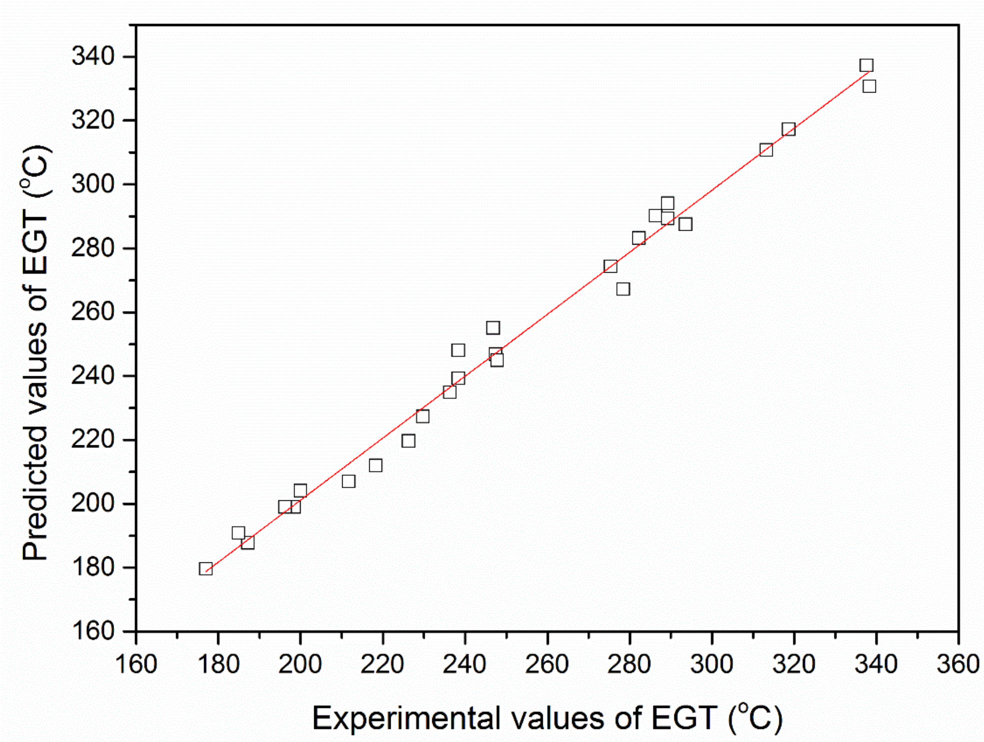


Fig. 8.7 Comparison of predicted values and experimental values of EGT

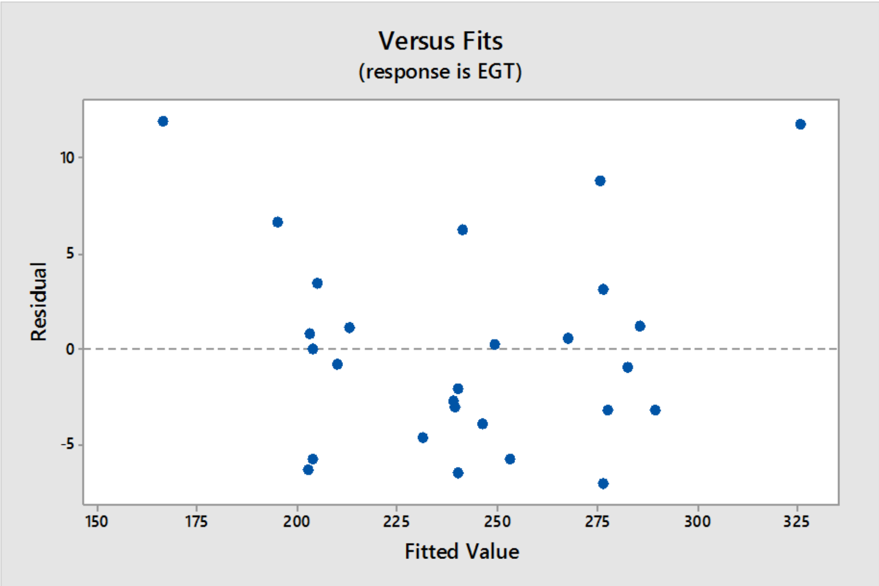


Fig. 8.8 Residual plot for EGT

Table 8.5 ANOVA results for exhaust gas temperature

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 396.2 | 396.2 | 0.004 |
| Compression ratio (B) | 1 | 2485.6 | 2485.6 | 0.000 |
| Injection pressure (C) | 1 | 3578.9 | 3578.9 | 0.000 |
| Injection timing (D) | 1 | 5457.0 | 5457.0 | 0.000 |
| Load(E) | 1 | 23635.0 | 23635.0 | 0.000 |
| A*A | 1 | 379.5 | 379.5 | 0.005 |
| Error | 20 | 752.8 | | |
| Total | 26 | 36317.5 | | |

The signal to noise ratio plots of Exhaust gas temperature is presented in Figure 8.9. S/N ratio for all the input factors are considered for the analysis. For optimising the EGT smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition. The optimum combination of parameter for BSFC is 30% blend (BK30), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 6kg load.

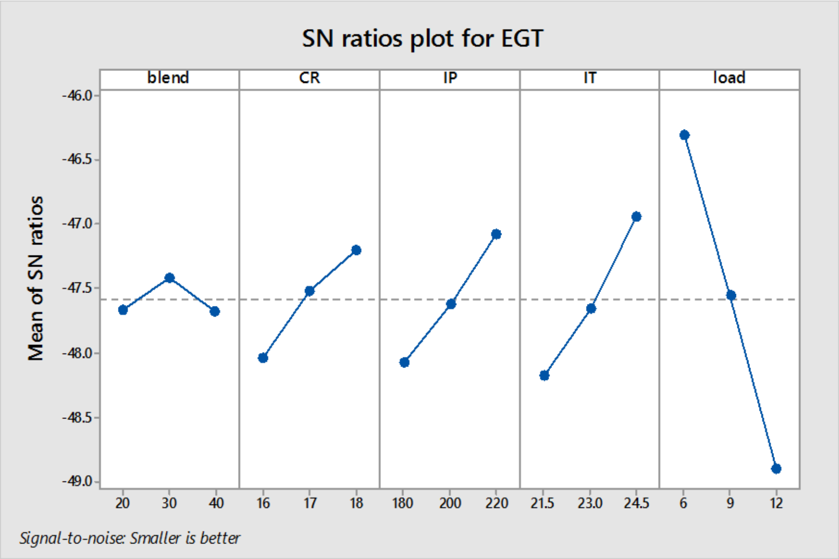


Fig. 8.9 The S/N ratios of EGT for all parameters

### 8.3.4 Carbon Monoxide Emission

Carbon monoxide emissions can be predicted using the modelled equation (8.7) in terms of fuel blend, compression ratio, injection pressure, injection timing and load. The values carbon monoxide emissions predicted using the equation (8.7) is plotted against the investigated values in Figure 8.10.

$$CO = 0.3915 - 0.000167 \times A - 0.01367 \times B - 0.001017 \times C - 0.002778 \times D - 0.002167 \times E + 0.000050 \times B \times C \quad (8.7)$$

The predicted value and the experimental values are very much close together with a predicted $R^2$ of 0.99. The residual plot of carbon monoxide is depicted in Figure 8.11. The residuals are randomly scattered about the horizontal axis, which witness the fitness of model with the data.

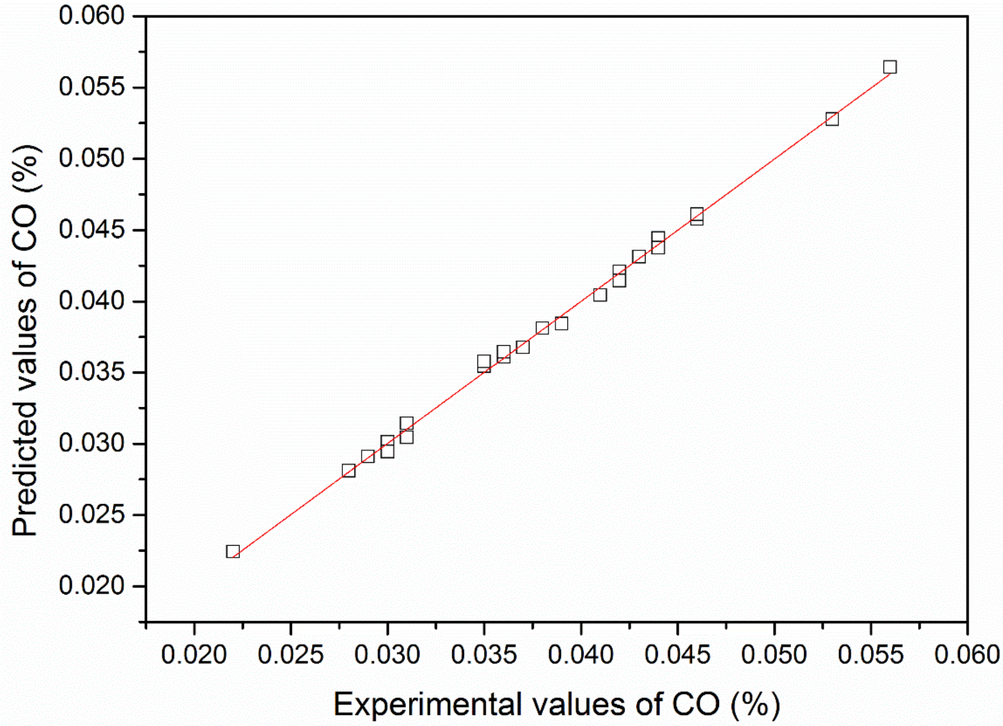


Fig 8.10 Comparison of predicted values and experimental values of carbon monoxide

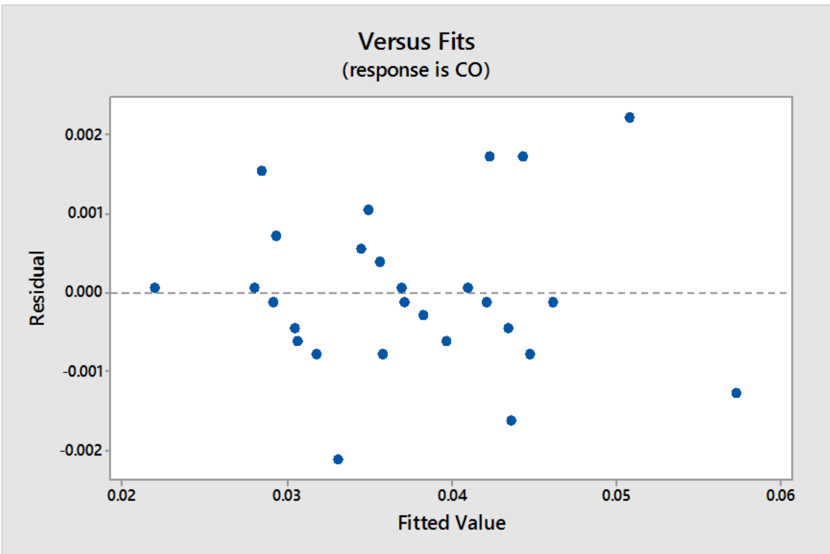


Fig. 8.11 Residual plot for carbon monoxide

ANOVA results for CO emission are shown in Table 8.6, which indicates various factors influencing CO emission with their level of significance. Carbon monoxide emission is influenced by fuel blend, compression ratio, injection pressure, injection timing and load. The $R^2$ value for the analysis is 0.9879.

Figure 8.12 indicates the signal to noise ratio plots of CO emission for all the factors considered in the analysis. For optimising the carbon monoxide emission smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition. The optimum combination of parameters for carbon monoxide emission is 40% blend (BK40), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 12kg load.

Table 8.6 ANOVA results for carbon monoxide

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 0.000031 | 0.000031 | 0.000 |
| Compression ratio (B) | 1 | 0.000014 | 0.000014 | 0.001 |
| Injection pressure (C) | 1 | 0.000011 | 0.000011 | 0.004 |
| Injection timing (D) | 1 | 0.000313 | 0.000313 | 0.000 |
| Load(E) | 1 | 0.000760 | 0.000760 | 0.000 |
| B*C | 1 | 0.000007 | 0.000007 | 0.012 |
| Error | 20 | 0.000020 | | |
| Total | 26 | 0.001627 | | |

### 8.3.5 Unburned Hydrocarbon Emissions

Equation (8.8) predicts the emissions of unburned hydrocarbon (HC) for these biofuel blends in terms of compression ratio, fuel blend, injection pressure, injection timing and load. The predicted values using this equation are plotted against the experimental values in Figure 8.13. The prediction $R^2$ value is 0.97. Residual plot of hydrocarbon emissions are represented in Figure 8.14.

$$HC = 231.6 - 0.100 \times A - 5.333 \times B - 0.1722 \times C - 2.815 \times D + 1.648 \times E \quad (8.8)$$

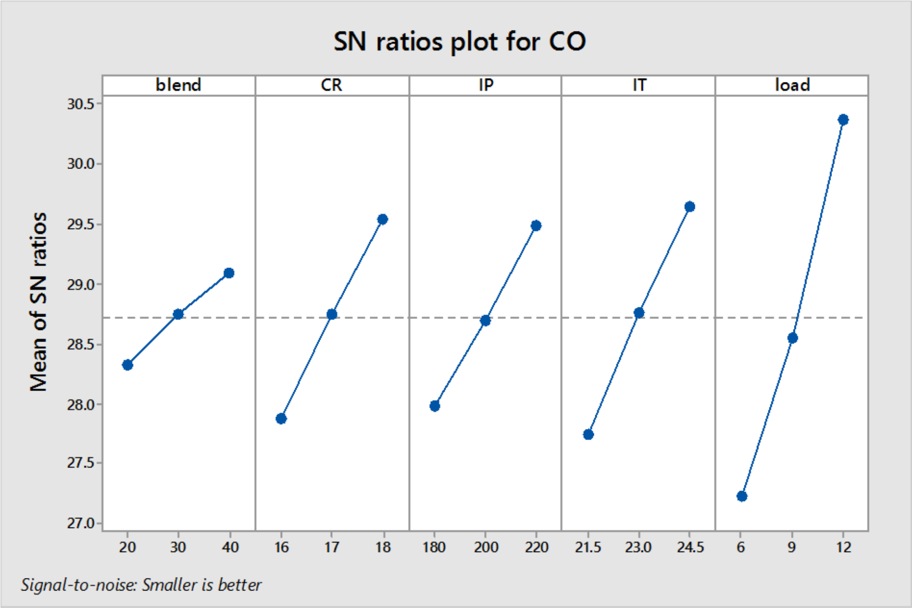


Fig. 8.12 The S/N ratios of carbon monoxide for all parameters

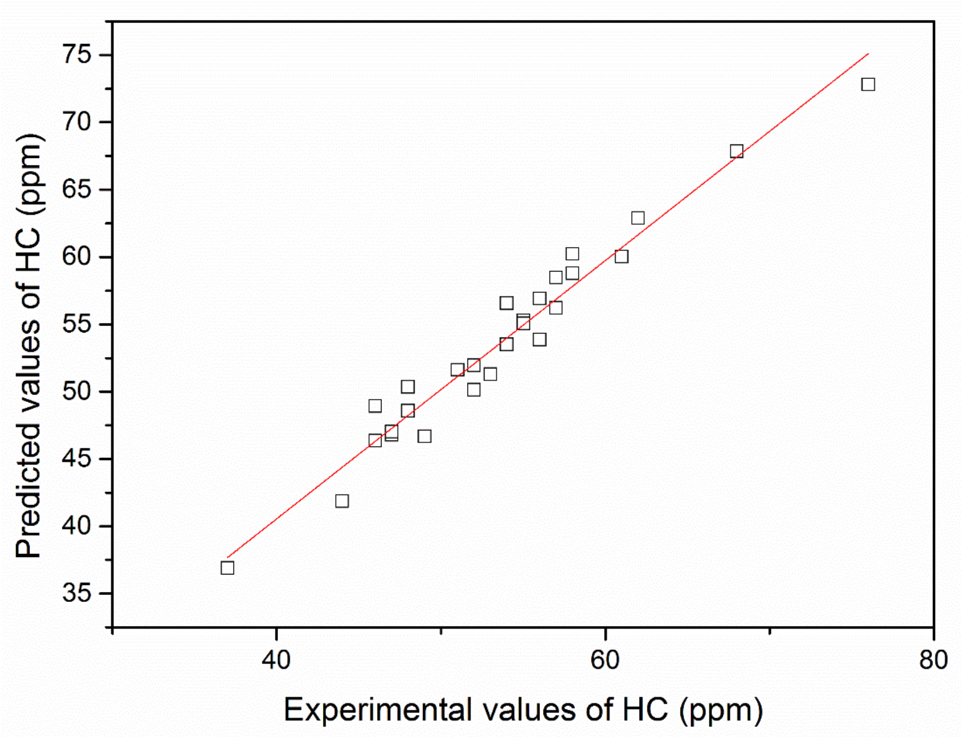


Fig 8.13 Comparison of predicted values and experimental values of unburned hydrocarbon

The residuals are well scattered around the horizontal axis, which supports the fitness of the model with the data.

Table 8.7 represents the results of ANOVA for unburned hydrocarbon emission. The fuel blend, compression ratio, injection pressure, injection timing and load are significant for the emission of unburned hydrocarbon since the p-values are less than 0.05. The $R^2$ value for the analysis is 0.9531. Figure 8.15 indicates the signal to noise ratio plots of HC emission for all the factors considered in the analysis. For optimising the HC emission smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition. From the S/N plots the optimum combination of parameter for HC emission is 30% blend (BK30), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 6 kg load.

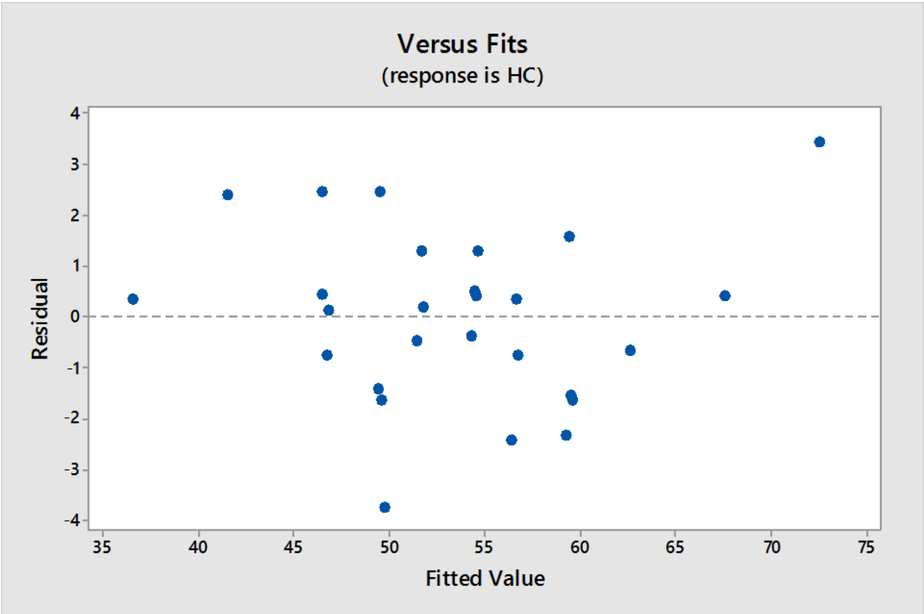


Fig. 8.14 Residual plot for unburned hydrocarbon

Table 8.7 ANOVA results for unburned hydrocarbon

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 18.00 | 18.000 | 0.035 |
| Compression ratio (B) | 1 | 512.00 | 512.000 | 0.000 |
| Injection pressure (C) | 1 | 213.56 | 213.556 | 0.000 |
| Injection timing (D) | 1 | 320.89 | 320.889 | 0.000 |
| Load(E) | 1 | 440.06 | 440.056 | 0.000 |
| Error | 21 | 74.02 | | |
| Total | 26 | 1578.52 | | |

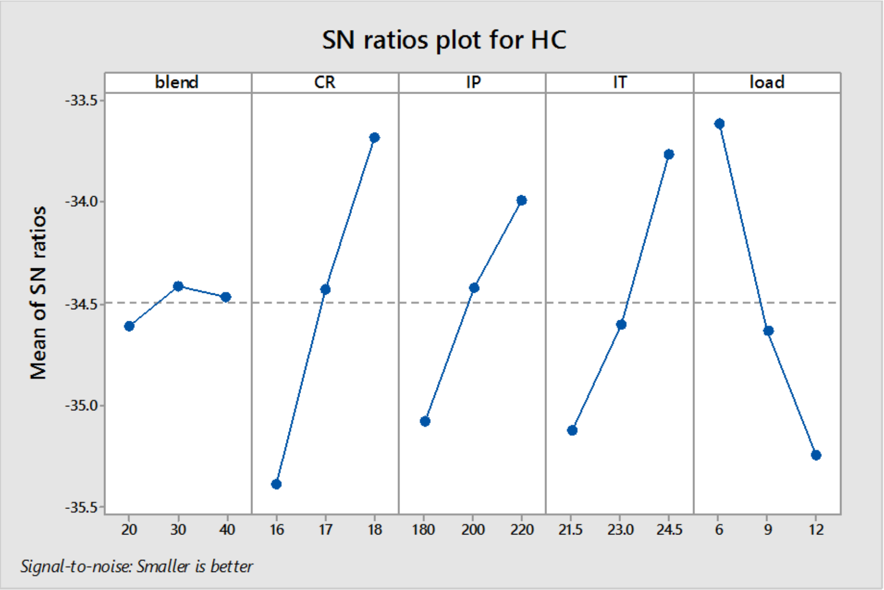


Fig. 8.15 The S/N ratios of unburned hydrocarbon for all parameters

### 8.3.6 Oxides of Nitrogen Emission

Oxides of nitrogen (NOx) emissions can be predicted using the equation (8.9) in terms of fuel blend, compression ratio, injection pressure, injection timing and load. The values NOx predicted using equation (8.9) is plotted against the investigated values in Figure 8.16.

$$NOx = 796 - 20.84 \times A + 20.50 \times B + 11.95 \times C - 189.9 \times D + 18.296 \times E + 0.3172 \times A \times A - 0.0282 \times C \times C + 4.54 \times D \times D \quad (8.9)$$

It is observed that the predicted NOx values are obey the experimental values with a high predicted $R^2$ value of 0.98. The residual plot of NOx is depicted in Figure 8.17. The residuals are randomly scattered around the horizontal axis, which indicates that the analysed model is suitable for the data.

ANOVA results for NOx is represented in Table 8.8. It is observed that the NOx is influenced by fuel blend, compression ratio, injection pressure, injection timing and load. The $R^2$ value for the analysis is 0.9758.

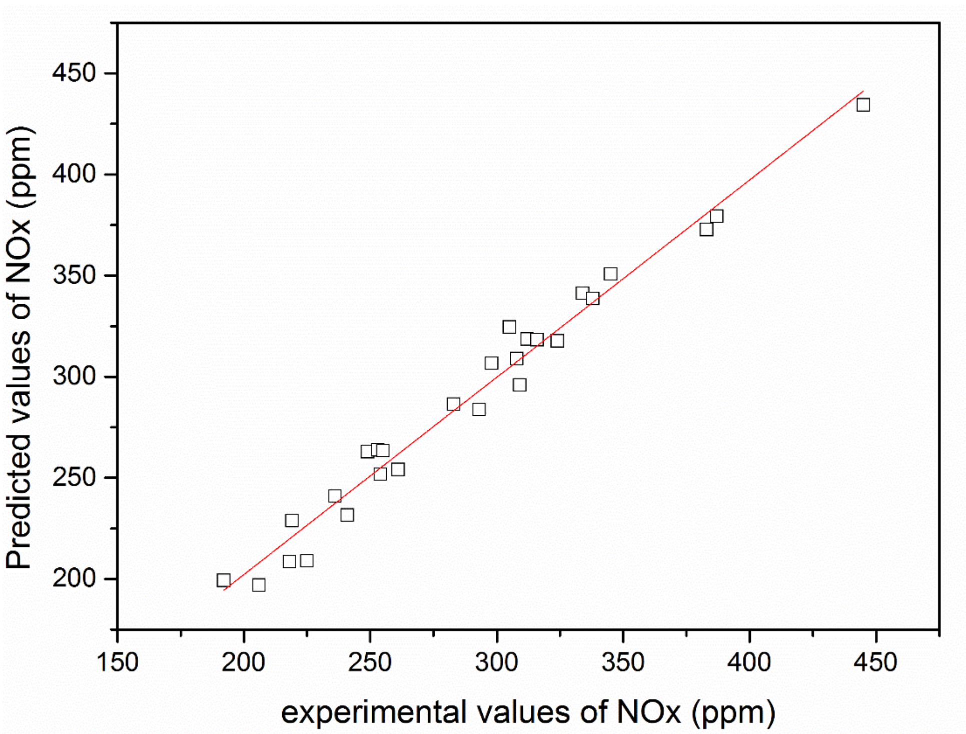


Fig 8.16 Comparison of predicted values and experimental values of NOx

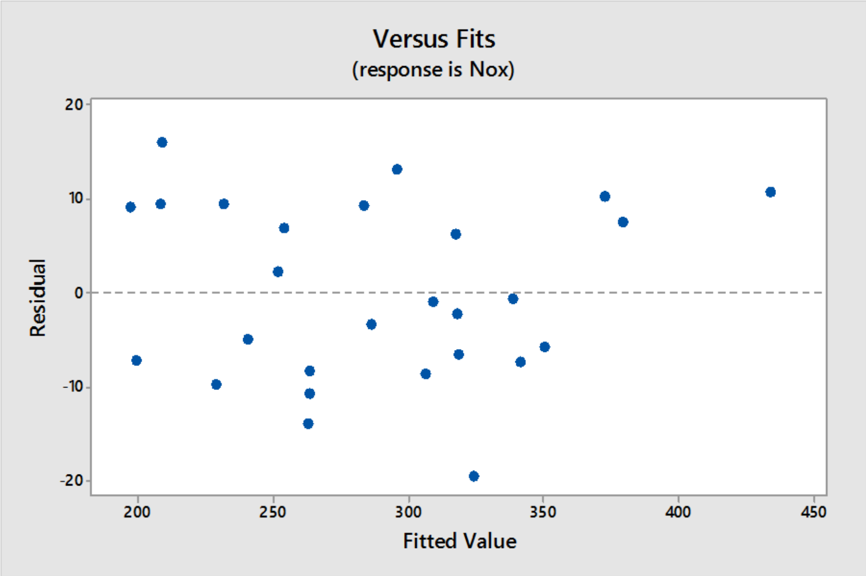


Fig. 8.17 Residual plot for NOx

Table 8.8 ANOVA results for oxides of nitrogen

| | Degrees of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 7171.3 | 7171.3 | 0.000 |
| Compression ratio (B) | 1 | 7564.5 | 7564.5 | 0.000 |
| Injection pressure (C) | 1 | 856.5 | 856.5 | 0.019 |
| Injection timing (D) | 1 | 517.4 | 517.4 | 0.060 |
| Load(E) | 1 | 54230.2 | 54230.2 | 0.000 |
| A*A | 1 | 6037.8 | 6037.8 | 0.000 |
| C*C | 1 | 763.1 | 763.1 | 0.025 |
| D*D | 1 | 627.0 | 627.0 | 0.040 |
| Error | 18 | 2309.6 | | |
| Total | 26 | 95472.7 | | |

Figure 8.18 presents the signal to noise ratio plots of NOx emission for all the factors considered in the analysis. For optimising the NOx emission smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are

selected as optimum condition. From the S/N plots the optimum combination of parameters for NOx emission is 30% blend (BK30), 16:1 compression ratio, 180 bar injection pressure, 21.5°BTDC injection timing and at 6 kg load.


Fig. 8.18 The S/N ratios of NOx for all parameters

### 8.3.7 Smoke Emission

Equation (8.10) predicts the emissions of smoke for these biofuel blends in terms of compression ratio, fuel blend, injection pressure, injection timing and load. The predicted values using this equation are plotted against the experimental values in the Figure 8.19. The prediction $R^2$ value is 0.98. Residual plot of smoke emission is represented in Figure 8.20.

$$SMOKE= 279.2 – 0.6889 \times A -5.500 \times B -0.2306 \times C -4.148 \times D + 2.926 \times E \quad (8.10)$$

The residuals are randomly scattered about the horizontal axis, which witness the fitness of model with the data.

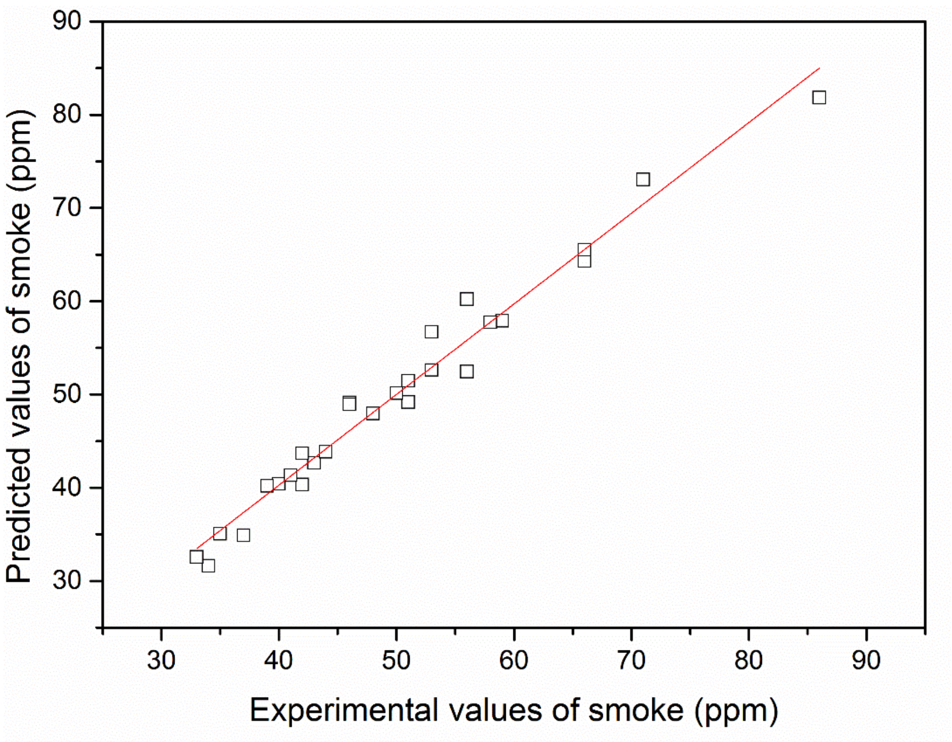


Fig 8.19 Comparison of predicted values and experimental values of smoke

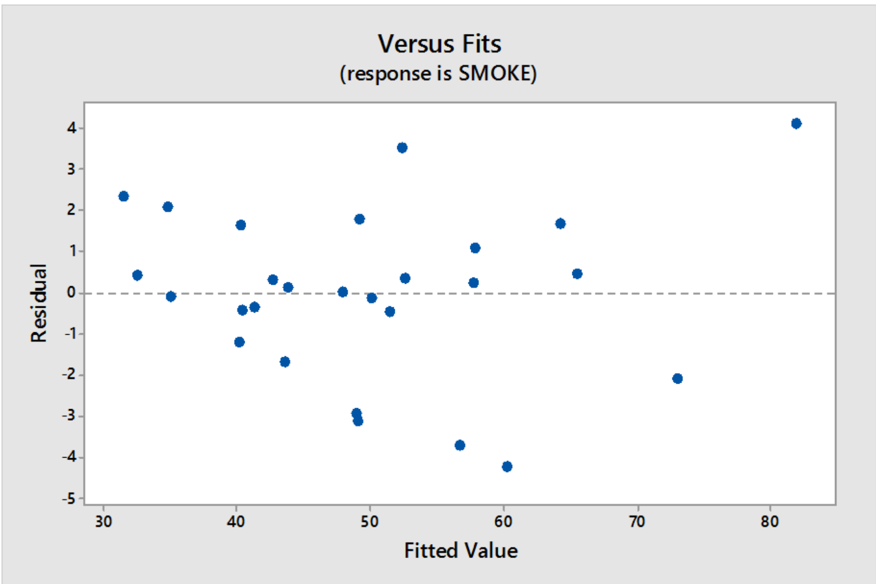


Fig. 8.20 Residual plot for smoke

Table 8.9 presents the results of ANOVA for smoke emission. The fuel blend, compression ratio, injection pressure, injection timing and load are significant for the emission of smoke since all p-values are less than 0.05. The $R^2$ value for the analysis is 0.9723.

Figure 8.21 indicates the signal to noise ratio plot of smoke emission for all the factors considered in the analysis. For optimising the smoke emission smaller is better objective function is used. The levels of the factors corresponding to highest S/N ratio are selected as optimum condition. From the S/N plots the optimum combination of parameter for smoke emission is 40% blend (BK40), 18:1 compression ratio, 220 bar injection pressure, 24.5°BTDC injection timing and at 6 kg load.

Table 8.9 ANOVA results for smoke emissions

| | Degree of freedom | Sum of squares | Mean square | P-value |
|---|---|---|---|---|
| Blend (A) | 1 | 854.2 | 854.22 | 0.000 |
| Compression ratio (B) | 1 | 544.5 | 544.50 | 0.000 |
| Injection pressure (C) | 1 | 382.7 | 382.72 | 0.000 |
| Injection timing (D) | 1 | 696.9 | 696.89 | 0.000 |
| Load(E) | 1 | 1386.9 | 1386.89 | 0.000 |
| Error | 21 | 110.2 | | |
| Total | 26 | 3975.4 | | |

**8.3.7 Determination of Optimum parameters**

By the analysis of S/N ratio and ANOVA the optimum levels of various input factors are determined and tabulated in Table 8.10. From this table the optimum combination of levels of factors for performance and emissions are selected. The blend 30% (BK30), compression ratio 18:1, injection pressure 220 bar, injection timing 24.5°BTDC and load 12 kg are set as optimized parameter levels. At this optimum parameter levels a set of experiments conducted and the results are compared to the predicted values at that condition using the prediction equation. The Table8.11 shows comparison of experimental and predicted values of the responses at the optimum condition. Percentage error between the experimental values and predicted values are determined. It is observed that for all the responses the error is less than 10%.

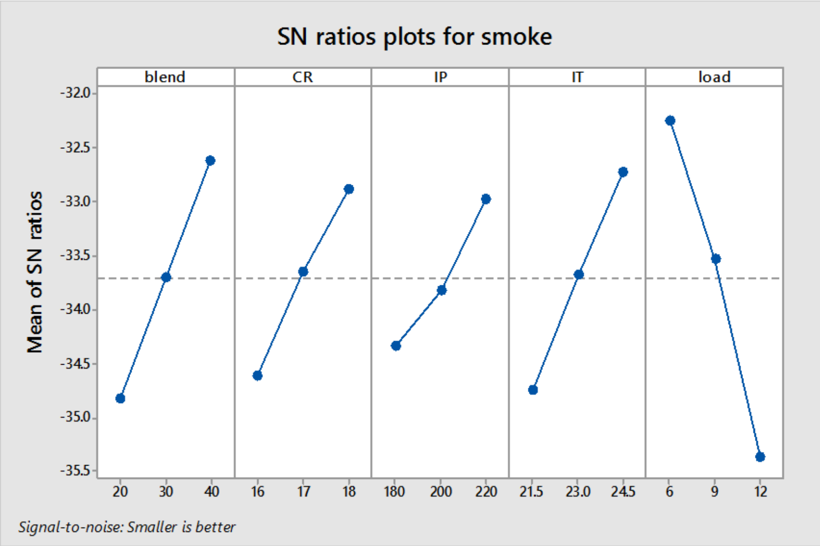


Fig. 8.21 The S/N ratios of smoke for all parameters

Table 8.10 Optimum levels of factors

| Response Variable | Optimum levels of factors | | | | |
|---|---|---|---|---|---|
| | Blend | Compression ratio | Injection pressure (bar) | Injection timing (°BTDC) | Load (kg) |
| Brake thermal efficiency | 30 | 18:1 | 220 | 24.5 | 12 |
| Brake specific fuel consumption | 30 | 18:1 | 220 | 24.5 | 12 |
| Exhaust gas temperature | 30 | 18:1 | 220 | 24.5 | 6 |
| Carbon monoxide | 40 | 18:1 | 220 | 24.5 | 12 |
| Unburned hydrocarbon | 30 | 18:1 | 220 | 24.5 | 6 |
| Oxides of nitrogen | 30 | 16:1 | 180 | 21.5 | 6 |
| Smoke | 40 | 18:1 | 220 | 24.5 | 6 |

Table 8.11 Comparison of predicted and experimental values at optimum condition

| Responses | Experimental value | Predicted value | Error (%) |
|---|---|---|---|
| Brake thermal efficiency (%) | 31.64 | 32.62 | 3.12 |
| Brake specific fuel consumption (kg/kWh) | 0.27 | 0.29 | 7.4 |
| Exhaust gas temperature (°C) | 229.13 | 229 | 0.1 |
| Carbon monoxide (%) | 0.02 | 0.0206 | 3 |
| Unburned hydrocarbon (ppm) | 44 | 45.5 | 3.4 |
| Oxides of nitrogen (ppm) | 383 | 381.5 | 0.4 |
| SMOKE (HSU) | 44 | 42.28 | 3.9 |

# CHAPTER 9

# ECONOMIC ANALYSIS

As discussed earlier, it is a known fact that the fuel price is increasing day by day due to its extensive usage and alarming depletion of resources.

## 9.1 COST CALCULATION

The main objective of this investigation is to use raw Cardanol as an alternative biofuel in diesel engines by considering its economic sustainability. Cost of a biofuel depends on the availability of the feed stock, oil production cost and transportation. The cost of Cardanol kerosene blend is calculated for the optimum blend (i.e. BK30) and compared with diesel fuel, as depicted below:

Table 9.1 gives the detailed cost calculations for the diesel fuel. Similarly, Table 9.2 indicates the detailed cost calculations for BK30 blend.

Table 9.1 Calculation of cost parameters for diesel fuel

| | |
|---|---|
| Market price of diesel fuel per litre | Rs. 68.50 |
| Specific gravity of diesel fuel used | 0.821 |
| Cost of one kg of diesel fuel (i.e. 68.50/0.821) | Rs. 83.43 |
| Brake specific fuel consumption for the diesel fuel | 0.29 kg/kWh |
| Cost of unit power per hour using diesel fuel (i.e. 0.29× 83.43) | Rs. 24.20 /kWh |

Table 9.2 Calculation of cost parameters for BK30 blend

| | |
|---|---|
| Market price of cardanol per litre | Rs. 58 |
| Specific gravity of cardanol used | 0.903 |
| Cost of one kg of cardanol (i.e. 58/0.903) | Rs. 64.23 |
| Market price of kerosene per litre | Rs. 40 |
| Specific gravity of kerosene used | 0.78 |

| | |
|---|---|
| Cost of one kg of kerosene (i.e. 40/0.78) | Rs. 51.28 |
| Cost of one kg of BK30 blend (0.3×51.28+0.7×64.23) | Rs. 60.35 |
| Brake specific fuel consumption for BK30 blend | 0.31 kg/kWh |
| Cost of unit power per hour using BK30 blend (i.e. 0.31× 60.35) | Rs. 18.71 /kWh |

By using 30% kerosene and 70% cardanol blend as an alternative fuel in diesel engine about 22.69 % fuel cost can be reduced.

### 9.2 APPLICATION IN MINING SECTOR

In mining sector Heavy Earth Moving Machinery (HEMM) are mainly powered by diesel engines. This machinery consumes huge amount of diesel fuel. For example, a Volvo 700 BLC excavator, shown in Figure 9.1, having 405 hp rated power consumes 45 litre/hr diesel fuel. It is operated for 8 hours in a shift for which the total diesel consumption was 360 litres. Instead of diesel fuel if cardanol kerosene (BK30) blend is used in this excavator the fuel cost can be reduced by Rs. 5595/- for a shift of 8 hours.


Fig. 9.1 Volvo excavator, Model: 700BLC

Similarly, a Volvo wheel loader L120F, which is shown in Figure 9.2, has rated power of 280 hp. It consumes 16 litre/ hr diesel fuel. If this wheel loader is operated with BK30 blend, the fuel cost can be reduced by Rs. 1989/- for a shift of 8 hours.


Fig. 9.2  Volvo wheel loader, Model: L120F

By using BK30 blend as a fuel in place of diesel, emission of smoke can be reduced by 3.5 %, unburned hydrocarbon by 5.08 % and carbon monoxide by 42%. Use of BK30 blend as a fuel in mining machinery the environmental pollution can also be reduced. This developed biofuel is produced from cashew nut shells. The cashew plants can grow in places which are 700 m above mean sea level wherein temperature does not fall below 20°C. It is suitable for any type of soil but preferably loam soil. Therefore, the cashew plantations can be grown on the dumped overburden of surface mines as reclamation process, so that the environmental pollution can be reduced and the cashew nuts so produced can be processed to get the edible cashew kernels and cashew nut shell, which is the source for the Cardanol oil. By using this biofuel the dependency on diesel fuel can be reduced, to some extent.

# CHAPTER 10

# CONCLUSIONS AND SCOPE FOR FUTURE WORK

## 10.1 SUMMARY OF RESULTS

A single cylinder variable compression ratio diesel engine was operated with various blends (i.e. BK10, BK20, BK30 and BK40) of Cardanol and kerosene. The effect of compression ratio, injection pressure and injection timing on performance and emission characteristics of the engine were analysed. Following were the outcome of the experimental investigations:

- By addition of kerosene to Cardanol the viscosity and density of the blend reduces with increase in percentage of kerosene in the blend. Viscosity of BK10, BK20, BK30 and BK40 blends were 14.2 cSt, 9.3 cSt, 5.9 cSt and 3.8 cSt, respectively, among which the viscosity of BK30 and BK40 blends were within the limits of biofuel standard (ASTM D6751).
  The density of BK10, BK20, BK30 and BK40 blends were respectively, 846 kg/m$^3$, 834 kg/m$^3$, 825 kg/m$^3$ and 811 kg/m$^3$, which are nearer to the density of diesel fuel (i.e. 821 kg/m$^3$).
- Calorific value of the blend increases with increase in the percentage of kerosene. The BK30 and BK40 blends were having calorific values of 41331 kJ/kg and 41712 kJ/kg which is closer to diesel fuel (43580 kJ/kg), whereas the calorific values of BK10 and BK20 blends (i.e. 40598 kJ/kg and 40960 kJ/kg) were much lower than that of diesel fuel.
- Among all the blends tested (i.e. BK10, BK20, BK30 and BK40), BK30 shown the highest brake thermal efficiency of 28.30%, which is very closer to diesel fuel (i.e. 28.72%).
- Brake specific fuel consumptions for BK10, BK20, BK30, BK40 and diesel were 0.38 kg/kWh, 0.34 kg/kWh, 0.31 kg/kWh, 0.32 kg/kWh and 0.29 kg/kWh, respectively. Lowest brake specific fuel consumption was obtained for BK30 blend.
- Exhaust gas temperature of all the tested blends, such as BK10, BK20, BK30 and BK40 were 297.42°C, 281.49°C, 274.96°C and 277.60°C, respectively. The

outcome of this study reveals that the exhaust gas temperature was lower for BK30 blend compared to other blends.

- Emission of carbon monoxide reduced with increase in kerosene percentage in the blend. The carbon monoxide emissions for BK10, BK20, BK30 and BK40 blends were respectively, 0.035%, 0.031%, 0.030% and 0.028%, which is lower than diesel.
- The present study showed that the unburned hydrocarbon emission for BK20, BK30 and BK40 blends were lower than the diesel fuel by 1.69%, 5.08% and 11.8%, respectively. However, for BK10 blend it was more by 5%.
- Smoke emissions for BK30 and BK40 blends were 55 HSU and 53 HSU, which are lower than that of diesel fuel (i.e. 56 HSU), whereas the emission was high with BK10 and BK20 blends (i.e. 78 HSU and 66 HSU).
- Emission of oxides of nitrogen increased by 30.8%, 24.4%, 1.9% and 2.6% for blends BK10, BK20, BK30 and BK40, respectively when compared to diesel fuel, which is the common characteristic of any biofuel, and also one of its drawback. For all the four tested blends BK30 shown the minimum NOx.
- The present study shows that for the increase in compression ratio from 16:1 to 18:1 the brake thermal efficiency increased by 14.3%, 11.3% and 17.8% for blends BK20, BK30 and BK40, respectively. Similarly, increase in injection pressure from 180 bar to 220 bar the brake thermal efficiency increased by 17.3%, 17.9% and 17.1% for blends BK20, BK30 and BK40, respectively. Also, advancement in injection timing from 21.5° BTDC to 24.5° BTDC, the brake thermal efficiency for BK20, BK30 and BK40 blends were increased by 12%, 5% and 7%, respectively. Hence, the results of this study demonstrated that the effect of injection timing on brake thermal efficiency is lower when compared to effect of compression ratio and injection pressure, which corroborates with the results of other biodiesels invented by earlier researchers.
- The effect of injection timing is more on carbon monoxide and unburned hydrocarbon emissions when compared to effect of compression ratio and injection pressure. When injection timing was advanced from 21.5° BTDC to 24.5° BTDC carbon monoxide emission was reduced by 18% and 18.7% for

BK30 and BK40 blends, whereas unburned hydrocarbon reduced by 17.2% and 17.8%.

- The results of this study revealed that the oxides of nitrogen emissions increases with increase in compression ratio, injection pressure and advancement in injection timing, in the same trend, for all the three blends (i.e. BK20, BK30 and BK40).
- For BK20, BK30 and BK40 blends, with increase in compression ratio from 16:1 to 18:1 the smoke emission was reduced by 5.8%, 20.6% and 13.5%, increase in injection pressure from 180 bar to 220 bar smoke emission was reduced by 10%, 23% and 17.5%, and advancement in injection timing from 21.5° BTDC to 24.5° BTDC smoke emission was reduced by 23%, 17% and 15%, respectively.
- The developed prediction model showed a good correlation coefficient ($R^2$) in the range of 0.9438 to 0.9879 for the prediction of performance and emission parameters of the engine.
- With the developed prediction model, BK30 blend (among the three blends considered in the present study i.e. BK20, BK30 and BK40) was proved as the most favourable blend, for optimum engine performance with minimum emissions, under the following operating conditions: compression ratio - 18:1; injection pressure - 220 bar; injection timing - 24.5°BTDC; load - 12 kg.
- By using this developed biofuel blend (BK30) in mining machineries the fuel cost can be reduced by 22.69%.

Table 10 gives a critical comparison of BK30 blend and diesel fuel.

Table 10 Comparison of BK30 blend and Diesel

| Parameters | BK30 Blend | Diesel |
|---|---|---|
| **MERITS** | | |
| Smoke emission | 55 HSU | 56 HSU |
| Unburned hydrocarbon emission | 56 ppm | 59 ppm |
| CO emission | 0.030% | 0.082% |
| **DEMERITS** | | |
| BSFC | 0.31 kg/kWh | 0.29 kg/kWh |
| $NO_x$ emission | 314 ppm | 308 ppm |
| Calorific value | 41331 kJ/kg | 43580 kJ/kg |
| Brake thermal efficiency | 28.30%, | 28.72%, |

## 10.2 CONCLUSIONS

The following conclusions were drawn from the summarized discussion of results:

- The developed biofuel BK30 blend is showing an inspiring result with minimum emissions and its thermal efficiency is well comparable with that of diesel.
- Developed Cardanol and kerosene BK30 blend is an innovative biofuel, which can be commercially used without any engine modification, preferably after detailed study of chemical composition of exhaust gases.
- The developed equations can be used for prediction of performance and emission parameters of the engine operated with this biofuel blend.
- The present research would act as a leading study in the utilization of Cardanol as a biofuel.

## 10.3 SCOPE FOR FUTURE WORK

- For commercial implementation of blend detailed chemical analysis of exhaust gas has to be carried out.
- Tribological study of the engine operated with this biofuel blend can be carried out.
- Studies can be done with super charging and changing the piston types.

# APPENDIX-I

## SPECIFICATIONS OF EXHAUST GAS ANALYSER AND SMOKE METER

Table 1: Multi gas analyser accuracy and range (Netel exhaust gas analyser, Model: NPM-MGA-1)

| Parameter | Accuracy | Range |
|---|---|---|
| HC | ±10 ppm | 0–20,000 ppm |
| CO | ±0.03% | 0–9.9% |
| NOx | ±25 ppm | 0–5000 ppm |

Source: Manufacturer instruction manual (NETEL INDIA LTD)

Table 2: Smoke meter accuracy and range (Netel smoke meter model, NPM-SM-111B)

| Parameter | Accuracy | Range |
|---|---|---|
| Smoke intensity | ± 1% | 0–100% opacity |
| Light absorption coefficient(K) | ± 0.1 $m^{-1}$ | 0–9.9 $m^{-1}$ |

Source: Manufacturer instruction manual (NETEL INDIA LTD)

## APPENDIX-II

## EXPERIMENTAL RESULTS OF PERFORMANCE ANALYSIS

Table 1: Brake Thermal efficiency (%)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | 10.12 | 11.82 | 12.62 | 12.38 | 12.83 |
| 6 | 15.93 | 17.26 | 20.33 | 18.74 | 20.41 |
| 9 | 21.80 | 22.81 | 25.35 | 22.87 | 25.45 |
| 12 | 23.69 | 25.83 | 28.30 | 27.06 | 28.72 |

Table 2: BSFC (kg/kWh)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | - | - | - | - | - |
| 3 | 0.88 | 0.75 | 0.70 | 0.73 | 0.64 |
| 6 | 0.56 | 0.51 | 0.43 | 0.46 | 0.41 |
| 9 | 0.41 | 0.39 | 0.34 | 0.38 | 0.33 |
| 12 | 0.38 | 0.34 | 0.31 | 0.32 | 0.29 |

Table 3: Exhaust gas temperature ($^oC$)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 166.03 | 164.26 | 156.18 | 160.83 | 146.53 |
| 3 | 180.63 | 178.78 | 174.19 | 176.99 | 163.98 |
| 6 | 214.57 | 209.18 | 202.59 | 208.58 | 196.78 |
| 9 | 251.29 | 242.31 | 234.08 | 241.54 | 226.21 |
| 12 | 297.42 | 281.49 | 274.96 | 277.60 | 269.19 |

Table 4: Carbon monoxide in %

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 0.082 | 0.084 | 0.086 | 0.089 | 0.061 |
| 3 | 0.071 | 0.073 | 0.075 | 0.070 | 0.065 |
| 6 | 0.053 | 0.046 | 0.043 | 0.042 | 0.069 |
| 9 | 0.042 | 0.039 | 0.036 | 0.033 | 0.075 |
| 12 | 0.035 | 0.031 | 0.030 | 0.028 | 0.082 |

Table 5: Unburned hydrocarbon (ppm)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 34 | 36 | 37 | 39 | 27 |
| 3 | 43 | 44 | 46 | 46 | 38 |
| 6 | 51 | 48 | 47 | 48 | 45 |
| 9 | 56 | 55 | 54 | 49 | 55 |
| 12 | 62 | 58 | 56 | 52 | 59 |

Table 6: NOx (ppm)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 128 | 119 | 109 | 123 | 92 |
| 3 | 186 | 178 | 152 | 159 | 116 |
| 6 | 267 | 249 | 214 | 221 | 172 |
| 9 | 372 | 324 | 263 | 287 | 268 |
| 12 | 403 | 383 | 314 | 316 | 308 |

Table 7: Smoke opacity (HSU)

| Load(kg) | BK10 | BK20 | BK30 | BK40 | DIESEL |
|---|---|---|---|---|---|
| 0 | 36 | 24 | 21 | 18 | 19 |
| 3 | 45 | 36 | 32 | 24 | 25 |
| 6 | 61 | 48 | 39 | 31 | 32 |
| 9 | 72 | 53 | 46 | 43 | 48 |
| 12 | 78 | 66 | 55 | 53 | 56 |

## APPENDIX III

## EXPERIMENTAL RESULTS AT VARIOUS COMPRESSION RATIOS

Table 8: Brake Thermal efficiency (%) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | 11.13 | 11.82 | 11.91 | 11.76 | 12.62 | 13.23 | 11.64 | 12.38 | 12.82 | 11.72 | 12.83 | 13.43 |
| 6 | 16.24 | 17.26 | 17.9 | 18.14 | 20.33 | 21.86 | 17.86 | 18.74 | 20.78 | 18.35 | 20.41 | 22.16 |
| 9 | 20.74 | 22.81 | 24.21 | 23.42 | 25.35 | 27.84 | 22.15 | 22.87 | 25.36 | 23.54 | 25.45 | 28.52 |
| 12 | 23.87 | 25.83 | 27.3 | 26.83 | 28.3 | 29.87 | 24.28 | 27.06 | 28.62 | 26.86 | 28.72 | 30.36 |

Table 9: BSFC (kg/kWh) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | - | - | - | - | - | - | - | - | - | - | - | - |
| 3 | 0.79 | 0.75 | 0.74 | 0.74 | 0.7 | 0.66 | 0.75 | 0.73 | 0.7 | 0.71 | 0.64 | 0.62 |
| 6 | 0.54 | 0.51 | 0.49 | 0.48 | 0.43 | 0.4 | 0.48 | 0.46 | 0.42 | 0.45 | 0.41 | 0.37 |
| 9 | 0.42 | 0.39 | 0.36 | 0.37 | 0.34 | 0.31 | 0.4 | 0.38 | 0.34 | 0.35 | 0.33 | 0.29 |
| 12 | 0.37 | 0.34 | 0.32 | 0.33 | 0.31 | 0.29 | 0.36 | 0.32 | 0.3 | 0.31 | 0.29 | 0.27 |

Table 10: Exhaust gas temperature ($^{o}C$) at different compression ratios

| Load (kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 169.72 | 164.26 | 159.28 | 162.84 | 156.18 | 151.68 | 164.76 | 160.83 | 157.85 | 152.23 | 146.53 | 140.76 |
| 3 | 185.56 | 178.78 | 171.26 | 179.78 | 174.19 | 168.57 | 183.24 | 176.99 | 171.76 | 170.15 | 163.98 | 158.46 |
| 6 | 223.43 | 209.18 | 201.21 | 216.57 | 202.59 | 193.78 | 221.21 | 208.58 | 197.62 | 201.36 | 196.78 | 192.1 |
| 9 | 262.28 | 242.31 | 228.12 | 256.24 | 234.08 | 221.86 | 259.64 | 241.54 | 228.73 | 234.12 | 226.21 | 221.52 |
| 12 | 302.57 | 281.49 | 264.81 | 294.33 | 274.96 | 258.14 | 297.23 | 277.6 | 263.91 | 276.18 | 269.19 | 265.65 |

Table 11: Carbon monoxide in (%) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 0.087 | 0.084 | 0.081 | 0.089 | 0.086 | 0.082 | 0.092 | 0.089 | 0.084 | 0.063 | 0.061 | 0.058 |
| 3 | 0.078 | 0.073 | 0.07 | 0.081 | 0.075 | 0.072 | 0.073 | 0.07 | 0.068 | 0.068 | 0.065 | 0.063 |
| 6 | 0.049 | 0.046 | 0.041 | 0.047 | 0.043 | 0.039 | 0.045 | 0.042 | 0.038 | 0.073 | 0.069 | 0.064 |
| 9 | 0.042 | 0.039 | 0.036 | 0.041 | 0.036 | 0.032 | 0.038 | 0.033 | 0.031 | 0.08 | 0.075 | 0.069 |
| 12 | 0.033 | 0.031 | 0.026 | 0.032 | 0.03 | 0.028 | 0.03 | 0.028 | 0.027 | 0.087 | 0.082 | 0.076 |

Table 12: Unburned hydrocarbon (ppm) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 39 | 36 | 34 | 41 | 37 | 35 | 42 | 39 | 36 | 31 | 27 | 24 |
| 3 | 48 | 44 | 40 | 50 | 46 | 41 | 52 | 46 | 44 | 42 | 38 | 32 |
| 6 | 54 | 48 | 44 | 52 | 47 | 43 | 53 | 48 | 45 | 51 | 45 | 39 |
| 9 | 59 | 55 | 51 | 58 | 54 | 49 | 56 | 49 | 46 | 59 | 55 | 47 |
| 12 | 62 | 58 | 53 | 60 | 56 | 51 | 62 | 52 | 48 | 65 | 59 | 52 |

Table 13: Oxides of Nitrogen (ppm) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 102 | 119 | 136 | 91 | 109 | 127 | 98 | 123 | 134 | 88 | 92 | 105 |
| 3 | 153 | 178 | 192 | 138 | 152 | 179 | 144 | 159 | 189 | 102 | 116 | 123 |
| 6 | 218 | 249 | 278 | 188 | 214 | 267 | 202 | 221 | 283 | 145 | 172 | 194 |
| 9 | 292 | 324 | 357 | 224 | 263 | 305 | 236 | 287 | 326 | 218 | 268 | 286 |
| 12 | 338 | 383 | 406 | 283 | 314 | 348 | 298 | 316 | 367 | 276 | 308 | 342 |

Table 14: Smoke opacity (HSU) at different compression ratios

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 | 16:1 | 17:1 | 18:1 |
| 0 | 28 | 24 | 22 | 26 | 21 | 20 | 23 | 18 | 17 | 21 | 19 | 18 |
| 3 | 41 | 36 | 33 | 36 | 32 | 29 | 29 | 24 | 22 | 28 | 25 | 22 |
| 6 | 52 | 48 | 46 | 48 | 39 | 34 | 39 | 31 | 28 | 36 | 32 | 28 |
| 9 | 56 | 53 | 49 | 49 | 46 | 41 | 48 | 43 | 39 | 52 | 48 | 42 |
| 12 | 68 | 66 | 64 | 63 | 55 | 50 | 59 | 53 | 51 | 64 | 56 | 52 |

# APPENDIX IV

## EXPERIMENTAL RESULTS AT VARIOUS INJECTION PRESSURES

Table 15: Brake Thermal efficiency (%) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | 10.76 | 11.82 | 12.22 | 11.35 | 12.62 | 13.87 | 10.97 | 12.38 | 13.18 | 11.53 | 12.83 | 14.2 |
| 6 | 16.13 | 17.26 | 18.16 | 17.86 | 20.33 | 23.13 | 17.24 | 18.74 | 22.65 | 18.35 | 20.41 | 23.67 |
| 9 | 20.14 | 22.81 | 24.98 | 22.34 | 25.35 | 28.68 | 21.83 | 22.87 | 25.78 | 23.68 | 25.45 | 29.78 |
| 12 | 23.57 | 25.83 | 27.66 | 26.57 | 28.3 | 31.34 | 25.42 | 27.06 | 29.81 | 26.85 | 28.72 | 31.97 |

Table 16: BSFC (kg/kWh) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | - | - | - | - | - | - | - | - | - | - | - | - |
| 3 | 0.82 | 0.75 | 0.72 | 0.77 | 0.7 | 0.63 | 0.78 | 0.73 | 0.65 | 0.72 | 0.64 | 0.58 |
| 6 | 0.55 | 0.51 | 0.49 | 0.49 | 0.43 | 0.39 | 0.5 | 0.46 | 0.42 | 0.45 | 0.41 | 0.35 |
| 9 | 0.44 | 0.39 | 0.37 | 0.39 | 0.34 | 0.31 | 0.4 | 0.38 | 0.34 | 0.35 | 0.33 | 0.28 |
| 12 | 0.38 | 0.34 | 0.32 | 0.34 | 0.31 | 0.28 | 0.35 | 0.32 | 0.29 | 0.31 | 0.29 | 0.26 |

Table 17: Exhaust gas temperature ($^{o}C$) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 168.71 | 164.26 | 157.26 | 161.84 | 156.18 | 149.67 | 161.61 | 160.83 | 155.74 | 150.12 | 146.53 | 138.62 |
| 3 | 183.47 | 178.78 | 170.14 | 177.88 | 174.19 | 166.45 | 179.26 | 176.99 | 169.18 | 168.95 | 163.98 | 155.86 |
| 6 | 221.83 | 209.18 | 200.16 | 213.71 | 202.59 | 187.88 | 218.22 | 208.58 | 195.92 | 200.86 | 196.78 | 189.8 |
| 9 | 260.17 | 242.31 | 220.2 | 251.22 | 234.08 | 217.94 | 257.44 | 241.54 | 225.61 | 230.32 | 226.21 | 220.12 |
| 12 | 298.87 | 281.49 | 260.11 | 290.23 | 274.96 | 250.15 | 295.23 | 277.6 | 259.61 | 273.11 | 269.19 | 263.45 |

Table 18: Carbon monoxide in (%) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 0.086 | 0.084 | 0.08 | 0.088 | 0.086 | 0.081 | 0.09 | 0.089 | 0.087 | 0.064 | 0.061 | 0.058 |
| 3 | 0.076 | 0.073 | 0.068 | 0.08 | 0.075 | 0.07 | 0.073 | 0.07 | 0.068 | 0.067 | 0.065 | 0.062 |
| 6 | 0.048 | 0.046 | 0.04 | 0.046 | 0.043 | 0.038 | 0.044 | 0.042 | 0.037 | 0.071 | 0.069 | 0.065 |
| 9 | 0.041 | 0.039 | 0.035 | 0.04 | 0.036 | 0.032 | 0.037 | 0.033 | 0.03 | 0.086 | 0.075 | 0.068 |
| 12 | 0.032 | 0.031 | 0.025 | 0.031 | 0.03 | 0.027 | 0.03 | 0.028 | 0.026 | 0.095 | 0.082 | 0.078 |

Table 19: Unburned hydrocarbon (ppm) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 41 | 36 | 33 | 42 | 37 | 30 | 41 | 39 | 34 | 34 | 27 | 23 |
| 3 | 48 | 44 | 38 | 49 | 46 | 38 | 50 | 46 | 43 | 41 | 38 | 30 |
| 6 | 53 | 48 | 42 | 50 | 47 | 40 | 52 | 48 | 45 | 52 | 45 | 40 |
| 9 | 57 | 55 | 49 | 56 | 54 | 47 | 55 | 49 | 44 | 57 | 55 | 46 |
| 12 | 63 | 58 | 52 | 59 | 56 | 48 | 58 | 52 | 49 | 64 | 59 | 50 |

Table 20: Oxides of Nitrogen (ppm) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 101 | 119 | 134 | 90 | 109 | 125 | 96 | 123 | 132 | 86 | 92 | 101 |
| 3 | 152 | 178 | 190 | 136 | 152 | 177 | 141 | 159 | 184 | 101 | 116 | 118 |
| 6 | 216 | 249 | 275 | 185 | 214 | 264 | 199 | 221 | 280 | 142 | 172 | 190 |
| 9 | 291 | 324 | 354 | 223 | 263 | 300 | 234 | 287 | 321 | 215 | 268 | 281 |
| 12 | 335 | 383 | 398 | 280 | 314 | 338 | 294 | 316 | 362 | 272 | 308 | 328 |

Table 21: Smoke opacity (HSU) at different injection pressures

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar | 180 bar | 200 bar | 220 bar |
| 0 | 27 | 24 | 21 | 26 | 21 | 18 | 23 | 18 | 17 | 22 | 19 | 18 |
| 3 | 42 | 36 | 32 | 37 | 32 | 28 | 28 | 24 | 21 | 29 | 25 | 21 |
| 6 | 53 | 48 | 44 | 49 | 39 | 32 | 38 | 31 | 27 | 37 | 32 | 27 |
| 9 | 57 | 53 | 47 | 51 | 46 | 38 | 47 | 43 | 36 | 53 | 48 | 42 |
| 12 | 69 | 66 | 62 | 64 | 55 | 49 | 57 | 53 | 47 | 65 | 56 | 51 |

## APPENDIX V

## EXPERIMENTAL RESULTS AT VARIOUS INJECTION TIMINGS

Table 22: Brake Thermal efficiency (%) at different injection timings

| | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Load(kg) | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 | 0 |
| 3 | 9.82 | 11.82 | 12.28 | 10.75 | 12.62 | 13.13 | 10.11 | 12.38 | 12.68 | 10.73 | 12.83 | 12.17 |
| 6 | 14.02 | 17.26 | 17.96 | 17.16 | 20.33 | 21.6 | 16.23 | 18.74 | 21.05 | 17.85 | 20.41 | 20.24 |
| 9 | 19.15 | 22.81 | 23.17 | 22.84 | 25.35 | 26.12 | 21.81 | 22.87 | 24.62 | 23.76 | 25.45 | 25.37 |
| 12 | 22.43 | 25.83 | 26.96 | 27.17 | 28.3 | 28.64 | 26.16 | 27.06 | 28.1 | 27.88 | 28.72 | 28.23 |

Table 23: BSFC (kg/kWh) at different injection timings

| | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| Load(kg) | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | - | - | - | - | - | - | - | - | - | - | - | - |
| 3 | 0.89 | 0.75 | 0.72 | 0.81 | 0.7 | 0.66 | 0.85 | 0.73 | 0.68 | 0.76 | 0.64 | 0.65 |
| 6 | 0.62 | 0.51 | 0.49 | 0.51 | 0.43 | 0.42 | 0.53 | 0.46 | 0.41 | 0.46 | 0.41 | 0.42 |
| 9 | 0.46 | 0.39 | 0.38 | 0.38 | 0.34 | 0.33 | 0.39 | 0.38 | 0.35 | 0.35 | 0.33 | 0.34 |
| 12 | 0.39 | 0.34 | 0.33 | 0.32 | 0.31 | 0.3 | 0.33 | 0.32 | 0.31 | 0.3 | 0.29 | 0.31 |

Table 24: Exhaust gas temperature ($^{o}C$) at different injection timings

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 170.76 | 164.26 | 154.76 | 162.93 | 156.18 | 147.64 | 162.21 | 160.83 | 153.71 | 149.23 | 146.53 | 137.23 |
| 3 | 186.17 | 178.78 | 169.23 | 180.87 | 174.19 | 163.51 | 184.86 | 176.99 | 166.15 | 166.91 | 163.98 | 153.62 |
| 6 | 224.65 | 209.18 | 198.24 | 216.16 | 202.59 | 182.24 | 223.26 | 208.58 | 192.91 | 198.64 | 196.78 | 186.18 |
| 9 | 265.12 | 242.31 | 217.83 | 255.23 | 234.08 | 211.17 | 263.25 | 241.54 | 221.67 | 228.12 | 226.21 | 218.23 |
| 12 | 300.22 | 281.49 | 259.17 | 295.21 | 274.96 | 244.23 | 299.52 | 277.6 | 256.46 | 270.22 | 269.19 | 258.34 |

Table 25: Carbon monoxide in (%) at different injection timings

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 0.088 | 0.084 | 0.081 | 0.087 | 0.086 | 0.08 | 0.091 | 0.089 | 0.083 | 0.064 | 0.061 | 0.056 |
| 3 | 0.079 | 0.073 | 0.069 | 0.08 | 0.075 | 0.071 | 0.073 | 0.07 | 0.068 | 0.068 | 0.065 | 0.061 |
| 6 | 0.049 | 0.046 | 0.042 | 0.047 | 0.043 | 0.039 | 0.045 | 0.042 | 0.036 | 0.073 | 0.069 | 0.063 |
| 9 | 0.046 | 0.039 | 0.036 | 0.042 | 0.036 | 0.033 | 0.038 | 0.033 | 0.032 | 0.087 | 0.075 | 0.066 |
| 12 | 0.039 | 0.031 | 0.028 | 0.033 | 0.03 | 0.027 | 0.032 | 0.028 | 0.026 | 0.095 | 0.082 | 0.076 |

Table 26: Unburned hydrocarbon (ppm) at different injection timings

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 44 | 36 | 35 | 45 | 37 | 32 | 46 | 39 | 33 | 35 | 27 | 23 |
| 3 | 50 | 44 | 37 | 48 | 46 | 36 | 47 | 46 | 42 | 45 | 38 | 32 |
| 6 | 54 | 48 | 40 | 51 | 47 | 39 | 52 | 48 | 43 | 50 | 45 | 38 |
| 9 | 59 | 55 | 48 | 55 | 54 | 46 | 53 | 49 | 45 | 59 | 55 | 48 |
| 12 | 64 | 58 | 51 | 58 | 56 | 48 | 56 | 52 | 46 | 66 | 59 | 53 |

Table 27: Oxides of Nitrogen (ppm) at different injection timings

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 98 | 119 | 138 | 92 | 109 | 128 | 95 | 123 | 134 | 84 | 92 | 112 |
| 3 | 149 | 178 | 192 | 138 | 152 | 181 | 145 | 159 | 189 | 98 | 116 | 127 |
| 6 | 214 | 249 | 277 | 189 | 214 | 268 | 203 | 221 | 274 | 143 | 172 | 214 |
| 9 | 286 | 324 | 358 | 234 | 263 | 302 | 240 | 287 | 334 | 218 | 268 | 285 |
| 12 | 333 | 383 | 402 | 282 | 314 | 342 | 296 | 316 | 368 | 276 | 308 | 335 |

Table 28: Smoke opacity (HSU) at different injection timings

| Load(kg) | BK20 | | | BK30 | | | BK40 | | | DIESEL | | |
|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° | 21.5° | 23° | 24.5° |
| 0 | 32 | 24 | 20 | 27 | 21 | 19 | 24 | 18 | 16 | 21 | 19 | 17 |
| 3 | 45 | 36 | 31 | 38 | 32 | 29 | 30 | 24 | 22 | 27 | 25 | 22 |
| 6 | 58 | 48 | 42 | 51 | 39 | 34 | 43 | 31 | 28 | 35 | 32 | 29 |
| 9 | 64 | 53 | 45 | 56 | 46 | 41 | 49 | 43 | 39 | 50 | 48 | 44 |
| 12 | 76 | 66 | 60 | 64 | 55 | 53 | 58 | 53 | 49 | 61 | 56 | 54 |

# APPENDIX VI

# REGRESSION ANALYSIS DATA

Welcome to Minitab, press F1 for help.

## Regression Analysis: BTE versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 7 | 447.356 | 63.908 | 0.000 |
| blend | 1 | 45.183 | 45.183 | 0.000 |
| CR | 1 | 7.375 | 7.375 | 0.005 |
| IP | 1 | 38.281 | 38.281 | 0.000 |
| IT | 1 | 43.214 | 43.214 | 0.000 |
| load | 1 | 265.959 | 265.959 | 0.000 |
| blend*blend | 1 | 40.145 | 40.145 | 0.000 |
| CR*CR | 1 | 6.848 | 6.848 | 0.007 |
| Error | 19 | 14.181 | 0.746 | |
| Total | 26 | 461.537 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 0.863923 | 96.93% | 95.80% | 93.77% |

Regression Equation

BTE = -383 + 1.654 blend + 37.7 CR + 0.0729 IP + 1.033 IT + 1.2813 load
- 0.02587 blend*blend
- 1.068 CR*CR

## Regression Analysis: BSFC versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 8 | 0.206519 | 0.025815 | 0.000 |
| CR | 1 | 0.013417 | 0.013417 | 0.000 |
| IP | 1 | 0.036072 | 0.036072 | 0.000 |
| IT | 1 | 0.010755 | 0.010755 | 0.001 |
| load | 1 | 0.010275 | 0.010275 | 0.001 |
| CR*CR | 1 | 0.004091 | 0.004091 | 0.025 |
| IT*IT | 1 | 0.011204 | 0.011204 | 0.001 |

| | | | | |
|---|---|---|---|---|
| load*load | 1 | 0.005202 | 0.005202 | 0.013 |
| CR*IP | 1 | 0.034252 | 0.034252 | 0.000 |
| Error | 18 | 0.012289 | 0.000683 | |
| Total | 26 | 0.218807 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 0.0261288 | 94.38% | 91.89% | 86.63% |

Regression Equation

BSFC = 8.39 - 1.676 CR - 0.06594 IP + 1.224 IT - 0.0831 load + 0.0261 CR*CR
- 0.02716 IT*IT
+ 0.00327 load*load + 0.003778 CR*IP

## Regression Analysis: EGT versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 6 | 35564.6 | 5927.4 | 0.000 |
| blend | 1 | 396.2 | 396.2 | 0.004 |
| CR | 1 | 2485.6 | 2485.6 | 0.000 |
| IP | 1 | 3578.9 | 3578.9 | 0.000 |
| IT | 1 | 5457.0 | 5457.0 | 0.000 |
| load | 1 | 23635.0 | 23635.0 | 0.000 |
| blend*blend | 1 | 379.5 | 379.5 | 0.005 |
| Error | 20 | 752.8 | 37.6 | |
| Total | 26 | 36317.5 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 6.13532 | 97.93% | 97.31% | 95.84% |

Regression Equation

EGT = 811.4 - 4.90 blend - 11.75 CR - 0.7050 IP - 11.608 IT + 12.079 load
+ 0.0795 blend*blend

## Regression Analysis: CO versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 6 | 0.001607 | 0.000268 | 0.000 |
| blend | 1 | 0.000031 | 0.000031 | 0.000 |
| CR | 1 | 0.000014 | 0.000014 | 0.001 |
| IP | 1 | 0.000011 | 0.000011 | 0.004 |
| IT | 1 | 0.000313 | 0.000313 | 0.000 |
| load | 1 | 0.000760 | 0.000760 | 0.000 |
| CR*IP | 1 | 0.000007 | 0.000007 | 0.012 |
| Error | 20 | 0.000020 | 0.000001 | |
| Total | 26 | 0.001627 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 0.0009916 | 98.79% | 98.43% | 97.64% |

Regression Equation

CO = 0.3915 - 0.000167 blend - 0.01367 CR - 0.001017 IP - 0.002778 IT
- 0.002167 load + 0.000050 CR*IP

## Regression Analysis: HC versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 5 | 1504.50 | 300.900 | 0.000 |
| blend | 1 | 18.00 | 18.000 | 0.035 |
| CR | 1 | 512.00 | 512.000 | 0.000 |
| IP | 1 | 213.56 | 213.556 | 0.000 |
| IT | 1 | 320.89 | 320.889 | 0.000 |
| load | 1 | 440.06 | 440.056 | 0.000 |
| Error | 21 | 74.02 | 3.525 | |
| Total | 26 | 1578.52 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 1.87742 | 95.31% | 94.19% | 91.85% |

Regression Equation

HC = 231.6 - 0.1000 blend - 5.333 CR - 0.1722 IP - 2.815 IT + 1.648 load

## Regression Analysis: Nox versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 8 | 93163.2 | 11645.4 | 0.000 |
| blend | 1 | 7171.3 | 7171.3 | 0.000 |
| CR | 1 | 7564.5 | 7564.5 | 0.000 |
| IP | 1 | 856.5 | 856.5 | 0.019 |
| IT | 1 | 517.4 | 517.4 | 0.060 |
| load | 1 | 54230.2 | 54230.2 | 0.000 |
| blend*blend | 1 | 6037.8 | 6037.8 | 0.000 |
| IP*IP | 1 | 763.1 | 763.1 | 0.025 |
| IT*IT | 1 | 627.0 | 627.0 | 0.040 |
| Error | 18 | 2309.6 | 128.3 | |
| Total | 26 | 95472.7 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 11.3274 | 97.58% | 96.51% | 94.43% |

Regression Equation

Nox = 796 - 20.84 blend + 20.50 CR + 11.95 IP - 189.9 IT + 18.296 load
+ 0.3172 blend*blend - 0.0282 IP*IP + 4.54 IT*IT

## Regression Analysis: SMOKE versus blend, CR, IP, IT, load

Backward Elimination of Terms

α to remove = 0.05

Analysis of Variance

| Source | DF | Adj SS | Adj MS | P-Value |
|---|---|---|---|---|
| Regression | 5 | 3865.2 | 773.04 | 0.000 |
| blend | 1 | 854.2 | 854.22 | 0.000 |
| CR | 1 | 544.5 | 544.50 | 0.000 |
| IP | 1 | 382.7 | 382.72 | 0.000 |
| IT | 1 | 696.9 | 696.89 | 0.000 |
| load | 1 | 1386.9 | 1386.89 | 0.000 |
| Error | 21 | 110.2 | 5.25 | |
| Total | 26 | 3975.4 | | |

Model Summary

| S | R-sq | R-sq(adj) | R-sq(pred) |
|---|---|---|---|
| 2.29061 | 97.23% | 96.57% | 95.31% |

Regression Equation

SMOKE = 279.2 - 0.6889 blend - 5.500 CR - 0.2306 IP - 4.148 IT + 2.926 load

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## List of Publications based on PhD Research Work

[to be filled-in by the Research Scholar and to be enclosed with Synopsis submission Form]

| Sl. No. | Title of the paper | Authors (in the same order as in the paper. Underline the Research Scholar's name) | Name of the Journal/ Conference | Year of Publication | Category * |
|---|---|---|---|---|---|
| 1 | Usage of Biofuels for Mining Applications: A review | Ravindra, M. Aruna and Harsha Vardhan | INDOROCK 2016: Sixth Indian Rock Conference | 2016 | 3 |
| 2 | Investigation on the performance of a variable compression ratio engine operated with raw Cardanol kerosene blends | Ravindra, M. Aruna and Harsha Vardhan | BIOFUELS (Taylor and Francis) | 2017 | 1 |
| 3 | Performance testing of Diesel Engine using Cardanol Kerosene oil Blend | Ravindra, M. Aruna and Harsha Vardhan | MATEC Web of Conferences, RiMES 2017 | 2018 | 3 |

* Category: 1: Journal paper, full paper reviewed 2: Journal paper, Abstract reviews 3: Conference/Symposium paper, full paper reviewed
4: Conference/Symposium paper, abstract reviewed 5: others (including papers in Workshops, NITK Research Bulletins, Short notes etc.)

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## *Academic Profile*

| Curriculum | Board/University | Institution | Class Awarded |
|---|---|---|---|
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| B.E in Mechanical Engineering (1995) | Mysore University | S.J.C.E.Mysore | FCD |
| SSLC (1989) | K.S.E.E. Board | S.D.P.T.High School.Kateel | Distinction |

## *Professional Experience*

- Working as Assistant professor since 21-7-2010 in the Dept. of Mechanical Engg. NMAMIT, Nitte.
- Worked as H.O.D Dept. of Mechanical Engg. at SDIT Mangaluru from 15-9-2006 to 20-7-2010
- Worked as H.O.D Dept. of Mechanical Engg. at MITK Kundapura from 1-9-2004 to 11-8-2006
- Worked as lecturer in SNS polytechnic Sunkadakatte from 24-6-1996 to 31-7-2004

## Conference/Journal publications

- Ravindra, Aruna.M and Harsha Vardhan, “Usage of Bio Fuels for Mining Applications:A Review”, Proceedings of 6 th Indian Rock Conference, IndoRock 2016, India, pp. 1001-1009
- Ravindra, M.Aruna and Harsha Vardhan, “Investigation on the Performance of a Variable Compression Ratio Engine Operated with Raw Cardanol and Kerosene Blends”, Biofuels, 2017,( https://doi.org/10.1080/17597269.2017.1389195)
- Ravindra, M.Aruna and Harsha Vardhan “ Performance testing of diesel engine using cardanol kerosene oil blend” MATEC Web of Conferences, RiMES 2017.

## *Conferences, Workshops and Trainings Attended*

1. Workshop on MISSION10X from9th March to11th March 2015 at NMAMIT Nitte.

2. Value orientation program Prajna on 9th September 2014 at Mangalore.
3. Workshop on Introduction to Therapeutic Counselling for Engineering Teachers from 2nd to 5th June2014.
4. Workshop on Effective leadership for excellence from20th and 21st June 2013 organized by IACE Bangalore
5. Faculty Development programme on recent Developments in Alternative Energy Technology at SIT, Mangalore on 18th and 19th Oct 2011
6. Summer School on Recent Developments in Environmental Management in Mining and other core industries at NITK on 27th June to 1st July 2011.
7. Workshop on Application of lab View in Engineering Education at NAMAIT from 10th to 13th August 2010.
8. 2 days workshop on Simulation of Combustion Engine Process for Innovative Engine Design for Future at NMAMIT on 2nd and 3rd March 2009
9. Workshop on Industrial Pollution Control and Advanced IC Engine at NMAMIT on 28th July t o 1st August 2008.
10. Summer School on Emerging Area of Technology in Smart Materials and Manufacturing at NITK from 30th June to 5th July 2008.
11. Workshop on Eco Friendly refrigeration at UBDT on 5th and 6th April 2007.
12. Workshop on Collaborative Mechanical System Modeling and Simulation at NITK, Surathkal on 27th October 2006.
13. Short-term programme on Advances in NDT at NITK, 30th June to 12th July 2003.
14. Workshop on Mechatronics held at KPT Mangalore from 24th April to 3rd May 1997.

Place **:** Nitte