Experimental Studies on Thumba Oil Operated VCR Engine for Long Term Durability

Ph.D. Thesis

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DECLARATION

I, **Narayan Lal Jain**, declare that this thesis titled, "**Experimental Studies on Thumba Oil Operated VCR Engine for Long Term Durability"** and the work presented in it, are my own. I confirm that:

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CERTIFICATE

This is to certify that the thesis entitled "**EXPERIMENTAL STUDIES ON THUMBA OIL OPERATED VCR ENGINE FOR LONG TERM DURABILITY"** being submitted by **Mr. Narayan Lal Jain (2011 RME 7146)** is a bonafide research work carried out under my supervision and guidance in fulfillment of the requirement for the award of the degree of **Doctor of Philosophy** in the department of Mechanical Engineering, Malviya National Institute of Technology, Jaipur, India. The matter embodied in this thesis is original and has not been submitted to any other University or Institute for the award of any other degree.

Place: Jaipur

Date: 12 July 2019

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Dedicated to my revered parents

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(Narayan Lal Jain)

ABSTRACT

The fast depletion of conventional energy sources along with the increasing demand for energy is a matter of grave concern. The exponential growth of automotive industry and the rapid increase in industrialization are further responsible for higher energy demand. The sharp rise in the prices of petroleum products has enhanced the interest of exploring the alternative fuels for internal combustion (I. C.) engines. Furthermore, the dangerous emissions released from the burning of fuels are resulting in the degradation of environment. This situation has created a dire need for clean and renewable alternative fuels.

Vegetable oils are renewable, and biodegradable, and also cause less pollution. Vegetable oil has the potential of adequately replacing the diesel. The developed countries are using edible vegetable oils as alternative fuel. However, in India, the use of edible vegetable oils as fuel is banned. Therefore, the non-edible vegetable oils can only be considered as alternative fuels for C.I. Engines. Vegetable oils like Jatropha, Karanj, Neem, Mahua oil, Sal oil, Rubber oil, Thumba oil, Jojoba oil, Castor oil, and Cottonseed oil are abundant in India. Thumba (Citrullus Colocyntis), a plant widely found in the western region of India, has properties comparable to diesel. Some of the studies are available on the use of Thumba biodiesel as a fuel in compression ignition (C.I.) engine, but very few studies are available on the use of Thumba straight vegetable oil (SVO); therefore, in this study, we selected Thumba SVO as a fuel.

A detailed and comprehensive experimental study has been carried out for assessing the feasibility and potential suitability of Thumba SVO as an alternative fuel for C.I. engine. Four strokes, stationary, single cylinder water cooled, direct injection, naturally aspirated, variable compression ratio multi-fuel engine, was used for the present study.

The present study is carried out in two stages; one is short term and the other is long term. The short-term research covers the comparative study of performance, combustion, and emission characteristics of the engine fuelled with diesel, different blends of the preheated and unheated Thumba oil along with diesel. The engine parameters like injection pressure, injection timing, and compression ratio were optimized for optimized preheated Thumba oil blend with diesel.

For endurance studies (long-term studies), the engine was run for 512 hours (32 cycles of 16 hours per run) as per IS 10000 using the preheated optimized Thumba oil blend with diesel.

After completing the endurance test, the component wear and lubricating oil analysis were carried out to check for any abnormal wear.

Generally, vegetable oils have high viscosity and low volatility, which cause problems such as high carbon deposition, gum formation, piston ring sticking, and injector choking. The viscosity of blended Thumba oil was reduced by heating with the use of engine exhaust gases in the current study.

The results of the short-term study revealed that the engine operating conditions were found optimum at the Compression Ratio of 21, Injection Timing of 23°CA BTDC, and Injection Pressure of 203 bar using diesel as a fuel. The experimental study was further carried out with preheated T10 (10% Thumba oil+ 90% Diesel), T20, T30, T50, and T100 blended Thumba oil along with diesel. The preheated T20 (80% Diesel+20% Thumba oil) Thumba oil blend with diesel showed better performance than the other blends of Thumba oil, therefore, the preheated T20 Thumba oil blend with diesel was found as the optimized blend. The engine parameters were further optimized using the preheated T20 Thumba oil blend along with diesel as a fuel. The engine operating condition was found optimum at the Compression Ratio of 22, Injection Timing of 23°CA BTDC, and Injection Pressure of 203 bar using T20 Thumba oil blend along with diesel. The brake thermal efficiency and brake specific fuel consumption of the engine operating with preheated T20 Thumba oil blend was observed to be very close to diesel.

Brake thermal efficiency of the engine was improved by 1.27% and smoke opacity, CO, HC emissions were reduced by 2.6%, 0.2%, and 5 PPM, respectively, because of preheating the T20 Thumba oil blend along with diesel as a fuel.

The results of the long-term study reported high carbon deposition on different internal parts of the engine. After the endurance test, the running surfaces of the cylinder liner, piston rings, and valves were found in good condition. After 450 hours test run of engine, it was observed that the dissolved Copper and Silicon found in the used lubricating oil were more than the permissible limit, but the viscosity of all the used lubricating oil gradually decreased to the bottom of the allowable limit.

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ABBRIVIATION

BTE	Brake Thermal Efficiency (%)
BSFC	Brake Specific Fuel Consumption (kg/ kW)
EGT	Exhaust Gas Temperature
UTODB	Unheated Thumba Oil Diesel Blend
PTODB	Preheated Thumba Oil Diesel Blend
T10	10% Thumba Vegetable Oil+ 90% Diesel
T20	20 % Thumba Vegetable Oil+ 80 % Diesel
T50	50% Thumba Vegetable Oil+ 50% Diesel
T100	100% Thumba Vegetable Oil+ 0% Diesel (Pure Thumba Oil)
CI	Compression Ignition
CR	Compression Ratio
CO	Carbon Monoxide
CO_2	Carbon Dioxide
O_2	Oxygen
HC	Hydrocarbon
NO_X	Oxides of Nitrogen
H_2SO_4	Sulphuric Acid
KOH	Potassium Hydroxide
HCs	Hydrocarbons
PPM	Parts Per Million
SPM	Smoke Particulate Matters
PM	Particulate Matter
ID	Ignition Delay
SOC	Start of Combustion
HRR	Heat Release Rate

CA	Crank Angle
CN	Cetane Number
LPG	Liquefied Petroleum Gas
CNG	Compressed Natural GAS
BTU	British thermal unit
MTCE	Million Tonnes Coal Equivalent
MTOE	Million Tonnes Oil Equivalent
PAH	Poly Aromatic Hydrocarbons
VCR	Variable Compression Ratio
OPEC	Organization of Petroleum Exporting Countries
GHG	Green House Gas
UBHC	Unburned Hydro Carbon
DEE	Di Ethyl Ether
HSDI	High Speed Direct Injection
DC	Direct Current
PSZC	Partially Stabilized Zirconia Coating
LHR	Low Heat Rejection
ETO	Epoxy Thumba Oil
TME	Thumba Methyl Ester
IP	Injection Pressure (bar)
IT	Injection Timing (°CA BTDC)
SVO	Straight Vegetable Oil
CI	Compression Ignition

Nomenclature

°CA BTDC	Degree Crank Angle before Top Dead Centre
°CA ATDC	Degree Crank Angle after Top Dead Centre
Kb /d	Kilo Barrel per Day
mb /d	Million barrels per Day
TWh	Terra watt hour
GW	Giga Watt
kCal / SCF	Kilo calorie per standard cubic foot
kJ/kg	Kilo Joule per kilo gram
kg /m3	Kilogram per meter cube
MJ/kg	Mega Joule per kilogram
kHz	Kilo Hertz
kW	Kilowatt
RPM	Revolutions per Minute
EJ	Exajoules (10^{18} Joules)

CHAPTER 1

INTRODUCTION

1.1 INTRODUCTION

Energy demand is increasing at an exponential rate because of urbanization, modernization, and incessant growth in the transportation vehicle. Energy sources are insufficient worldwide mostly in the developing countries like India. Energy demand and consumption in India is still below the world average. Indian population is 18% of the world population and consumes only 6% energy of the world [1, 2]. India's energy demand has further doubled since 2000, and its global share increased from 4.4% in 2000 to about 5.7% in 2013 [3, 4]. Moreover, energy plays a vital role in the economic growth of any country. India's economy is growing very fast; hence, there is huge demand of energy, which being expressed on a per-capita basis shows an increase of 46% since 2000 [5, 6, 7].

The existing conventional energy sources are nonrenewable and are depleting at a faster rate. Moreover, the conventional fuels emit hazardous elements, which result in the degradation of the environment. These emissions have –perilous impact such as acid rains, photochemical smog (HC, NOx), toxicity (CO), respiratory ailments (SPM), and neurological damage (Pb). The emission control requirements and renewable fuel developments have resulted in significant changes in the design and operation of internal combustion engines [8, 9, 10].

The two major problems energy crisis and environmental degradation are occurring because of the utilization of conventional fuels as the primary energy sources. Hence, there is a dire need to explore the alternative energy sources, which are renewable, green, and clean. Vegetable oils may be used as an alternative to liquid petroleum fuel. The developed countries are already using edible vegetable oils as fuel. However, in India, the use of edible vegetable oils as fuel is banned. Moreover, plenty of non-edible vegetable plants grow in India and the oil content in their seeds is also excellent. The non-edible vegetable oils like Jatropha oil, Karanja oil, Mahua oil, Sal oil, Neem oil, Rubber oil, Thumba oil, Jojoba oil, Castor oil, and Cottonseed oil are available in abundance in India [11-16].

Pure vegetable oils cannot be used as fuel in the compression ignition engines owing to their higher viscosity, which can create problems like injector coking, piston ring sticking, and hard and heavy carbon deposition. Therefore, it is necessary to reduce the viscosity of these oils before utilizing them as fuel [17-23].

1.2 ENERGY SCENARIO

1.2.1 Conventional Energy Scenario

World energy demand is expected to rise from 633 EJ in 2015 to 774.37 EJ in 2040. The share of global emissions is expected to increase from 12% to 27% in 2040. The world average per capita energy consumption was found to be approximately 1.92 tonnes oil equivalent (toe) in 2013, and the average per capita energy consumption for India was about 0.65 toe in 2013, and was 6.8 toe and 2.2 toe, respectively, for the US and China [3, 4].

The energy demand of India increased from 441 Mtoe in 2000 to775 Mtoe in 2013. India imports a large quantity of crude oil to fulfill the domestic need. The need of domestic crude oil is approximately 4.4 Mb/d; however, the production of 900-kilo barrels per day (kb/d) is not sufficient to meet the requirements. Oil consumption in 2014 was about 3.8 million barrels per day (Mb/d), of which 40% is utilized in the transport sector. Crude oil consumption and demand are steadily increasing in India. Therefore, the net oil imports will be 9.3 Mb/d by 2040 and India's dependency on oil import will be over 90%. Electricity demand increased from 376 terawatt-hours in 2000 to 897 terawatt-hours in 2013. The annual growth rate of India's electricity demand is about 6.9%, and electricity accounts for 15% of the final energy consumption [4-7, 24-26]. On the supply side, India's power generation capacity is 290 gigawatts. The generation and distribution of energy for the year 2000 and 2013 from different sources are as shown in Figure 1.1. The figure shows that maximum power is generated using coal, whereas the minimum power is generated through nuclear sources [4, 27, and 28].

1.2.2 Non-Conventional Energy Scenario

1.2.2.1 Solar and Wind Energy

World renewable energy production capacity is 673 GW, of which the contribution of wind energy is approximately 58%. In Figure 1.2., the world and India's renewable energy scenario is presented. India occupies the fifth position in the worldwide production of wind energy and eleventh position in the worldwide production of solar energy [28]. The contribution of renewable power generation reaches a share of 50% in the European Union, and is approximately 30% in China and Japan, and more than 25% in the United States and India [29, 30].

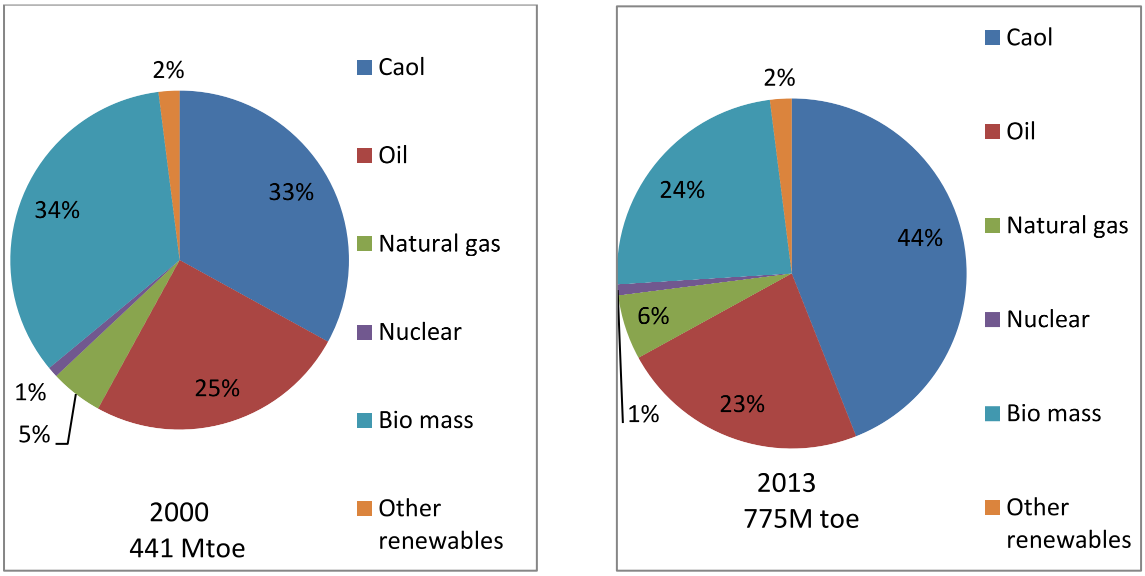


Figure 1.1 Generation and distribution of conventional energy in India

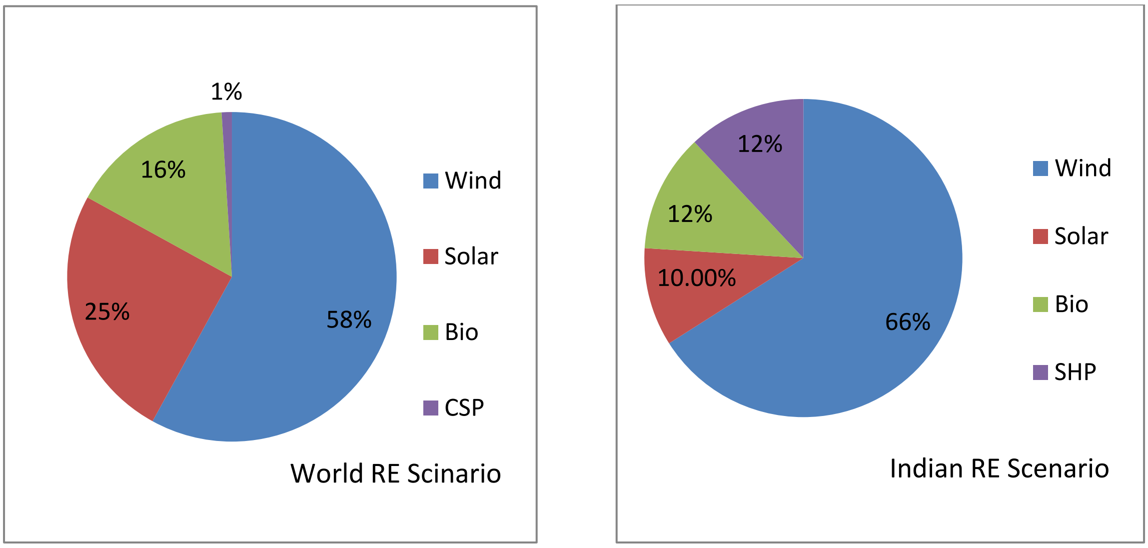


Figure 1.2 Generation and distribution of nonconventional energy of world and India

1.2.2.2 Bioenergy

The one-fourth of India's energy demand is meeting out through the use of bioenergy, and the maximum bioenergy is used for household cooking. The source of this bioenergy in India is a residue of the agriculture sector. The emission from household cooking creates an adverse effect on the health of users. In 2014, the power generation capacity through biomass was around 7 GW. The share of modern bioenergy is insignificant in India; hence, efforts are being made by the Indian government to enhance the share of the overall bioenergy by initiating National Bio Energy Mission [4].

Biofuels are another source of bioenergy in India. The government of India announced in 2009 that it would be mandatory to blend bioethanol and biodiesel with 20% diesel by 2017. However, its implementation is very slow than planned and blending of only 5% ethanol has started at many places in India. In 2017, the biodiesel fuel pump has been installed in Rajasthan and Uttarakhand states of India [30].

1.2.3 Rural Energy Scenario

Most of the population of the developing countries resides in the rural areas. In India, almost 70% of the population is living in the rural areas, and their energy demand is deficient as compared to the urban areas. For energy, the rural population depends on the state grid, diesel fuel, and traditional bioenergy. Diesel prices are unaffordable for them, and the use of bioenergy in conventional style for cooking is harmful. Further, the electricity supply is irregular and inadequate. Therefore, it becomes mandatory to find the alternative of the existing energy resources [31-33].

1.3 ALTERNATIVE ENERGY SOURCES

The unconventional energy sources are called as the alternative energy sources, which are long-lasting, clean, and bio-degradable. The principal alternative energy sources are solar, wind, and bioenergy in India. The implementation of wind and solar power generation technologies are costlier in the rural areas. The alternative fuels have all the required qualities of replacing the conventional petroleum fuels—wholly or partially. Many studies have already been carried out on different alternative fuels, but these fuels are only partial used, even though they have huge potential worldwide. The different alternative fuels are liquefied petroleum gas (LPG), compressed natural gas (CNG), hydrogen, alcohols, and vegetable oils [34-38].

LPG is a mixture of lower hydrocarbons that are used in the Indian household for cooking purposes, and it emits very less pollution in comparison to the conventional fuels. It produces less noise during combustion and increases engine life. Owing to its higher molecular weight, LPG settles down being exposed and may form explosive; hence, it is risky to use [39].

Compressed natural gas (CNG) is a mixture of lower hydrocarbon in which 80% to 90% is methane. It also causes less pollution than conventional fuels, but is costlier than diesel. Though this gas is already in use for vehicles, it is riskier than others and requires heavy-duty cylinders for storing and dispensing owing to its high pressure. CNG cylinders are filled with pressure of 200 kg/cm^2 to 225 kg/cm^2 and LPG cylinder are filled with pressure of 9 bar and gasoline at atmospheric pressure. High pressure storage is always dangerous than low pressure whenever blast occurs lot of area is covered in damage done. A leaking CNG tank will give rise to a gas jet which becomes combustible when the gas mixes with the air of the environment. When ignited, the gas jet turns into a jet fire. The rupture of a CNG tank will result in a physical explosion characterized by a blast wave if the natural gas released is not ignited. If the gas is ignited at the time of release one will experience a gas explosion. If the ignition of the gas is delayed a gas cloud fire will occur.

Ethanol and methanol are probably a better substitute for petrol and diesel. Alcohols produced through the fermentation of biomass in the absence of oxygen can be mixed with up to 22% petrol and 15% diesel. Further, the government of India had made the policy of mixing up to 20% ethanol with gasoline by 2017 [41].

Hydrogen is a completely clean fuel being produced from water, and it does not emit any harmful emissions like carbon monoxides, hydrocarbon, smoke, and particulate matters. It does not emit sulphur and lead compounds and is better than natural gas and other gaseous fuels used in many combustion engine applications, but its energy content (82 Kcal/SCF) is less than natural gas (262 Kcal/SCF). The handling, storing, and producing hydrogen are expensive; hence, there is need for further research to remove this complexity. Upon resolving this problem, hydrogen can become the future fuel of the world [42].

Vegetable oils are green, clean, renewable, and available at all places in the rural areas across the world. The viscosity of these oils is very high; hence, it becomes mandatory to reduce their viscosity before using these oils as fuel [43-46]. A transesterified form of vegetable oils is known as biodiesel [47]. In the rural and remote areas of the developing countries, grid power is unavailable. Therefore, vegetable oils can play a vital role in decentralized power generation for irrigation, electrification, and transportation [48-51].

1.4 VEGETABLE OILS AS ALTERNATIVE FUELS

Vegetable oils are composed of triglycerides (90% to 98%), monoglycerides, and diglycerides (2%-10%). Triglycerides contain three fatty acids and a glycerol molecule. The quantity of fatty acid in the vegetable oil depends on the length of carbon chain and the number of double bond present in their molecular structure. Further, these oils contain free fatty acid up to 5%. Palmitic, Linoleic, Stearic, and Linolenic acids are found in the fatty acid of vegetable oils. The suitability of these oils as fuels for diesel engines depends on their physical, chemical, and combustion characteristics [52-54].

Vegetable oils have comparable energy density, heat of vaporization, cetane number, stoichiometric air/fuel ratio with mineral diesel. Furthermore, these oils have the potential for significant reduction in emissions like sulfur oxides, carbon monoxide, Poly Aromatic Hydrocarbons (PAH), and particulate matter [55, 56].

Considering all the above properties, vegetable oils have one major problem, i.e., higher viscosity, which leads to unsuitable fuel spray. Further, lower volatility, poor atomization, and improper mixing of fuel with air result in incomplete combustion. Further, it increases carbon deposition, gum formation, particulate emission, and unburned fuel in the lubricating oil. These features make vegetable oils less suitable as fuel for compression ignition engine because they do not help in the reduction of viscosity [57-60]. Viscosity reduction is necessary to increase the suitability of the vegetable oils as fuels for diesel engines. The four common techniques used to reduce viscosity of vegetable oils are blending/dilution, heating/ pyrolysis, microemulsion, and transesterification. After modification, these oils become suitable for use as fuel in compression ignition engine [61-63].

1.4.1 Problem with Biodiesel

Biodiesels are usually not suitable for storage. Oxidative stability is a crucial issue, which is generally predicted by an iodine number (ASTM D 1510). The iodine number measures the presence of a double bond of carbon in the oil structure that is prone to oxidation [64-71]. The thumb rule for instability is increase in the number of the double bond. Poor stability results in the increase of gum formation and sedimentation [72]. Biodiesel is a mild solvent of painted surfaces and may deface the paints. It tends to clean out the storage tanks like vehicle fuel tanks and system storage tanks. The sludge gets accumulated in the storage system, which becomes thicker with time. Biodiesel dissolves these sediments and carries them into the fuel system of the engine that may lead to clogging of the filter and whole fuel system [73].

1.4.2 Suitable Method of Viscosity Reduction for Present Study

Transesterification requires a lot of chemicals and other items. The procurement of these items is not feasible in the rural area owing to logistic problems hence, this technique is inappropriate for these regions. In this study, it is found that the combination of blending and preheating is the most suitable process for reducing the viscosity of the vegetable oil. Waste engine exhaust gases used for heating vegetable oil and it is also blended with diesel for reducing viscosity.

1.5 SELECTION OF FUEL FOR PRESENT STUDY

On the basis of the availability and suitability of vegetable oils, the different countries have distinct varieties of vegetable oils.. The nations of tropical climate like Malaysia and Indonesia are mainly producing coconut and palm oil [74, 75]. The United States prefers to utilize soybean oil, and many European countries are focusing on the production of rapeseed oil [76]. However, the significant potential of the non-edible and toxic oilseeds remain untapped. The seed oils of these plants have diverse physical and chemical properties. India has about 80 different types of oilseeds that are borne by varied trees like Thumba Neem, Mahua, Sal, Karanj, Kusum, Kokam, Undi, Pisa, Dhupa, etc. [77]. The suitability of some of these oilseeds as fuel has been assessed for utilizing them in compression ignition engine.

Thumba oil is one of the non-edible vegetable oils, which is known as Indrayan in Hindi and Bitter Apple in English. This plant is a native of Turkey and is also found in the different parts of Asia and African countries. In India, it grows mostly in the desert region of Rajasthan and Gujarat. It is a creeper type of plant and grows well in sandy soil. The crop cycle of this plant is only six months and grows with other crops without any particular care. Presently, the local soap industries consume this oil. The properties of this plant oil are closer to that of diesel as compared to Jatropha and Pongamia Piñnata vegetable oils [78].

Jatropha oil has been assessed and approved as fuel by the government of India. However, Thumba oil is better than Jatropha oil. Comparison of Thumba and Jatropha oils is given in Table 1.1 below. Thumba plant of small crop cycle and various uses can play a vital role in the Indian rural economy, but it has not been examined comprehensively. Therefore, the fuel selected for the current study is Thumba oil.

Table 1.1: Comparison of Jatropha and Thumba oil [78, 79]

Sr. No.	Characteristics	Plant	
		Jatropha Curcas (Ratanjyot)	Thumba (Citrullus Colocyntis)
1	Plant Type	Shrub (3-5 m)	Creeper
2	Crop cycle	3-30 Years	6 months

3	Soil type	Any Type	Dry and desert
4	Saplings	1200 plants/ Acre	Separate land is not required
5	Maturity harvesting	After five years	Six months
6	Usage	Oil with medicinal value, dyes, feed, insecticide, pesticides, an alternative to diesel	Similar to Jatropha, but edible for animals and medicinal value
7	Viscosity	Higher than Thumba oil	Lower than Jatropha oil

1.6 DURABILITY STUDY

Both the short-term and long-term studies have been carried out for assessing the suitability of vegetable oils as fuel for compression ignition engine. An experimental short-term study was conducted to observe the performance, emission, and combustion behavior of the engine using these fuels for a short duration. However, this study does not address the implication of using the oil as fuel during the long run of the engine.

In the long-term study, the engine was made to run for 512 hours as per specific guidelines (IS-10000 part-9) with variable load conditions, and per day, the engine ran for sixteen hours as per the given cycle time. The test was conducted without a break for 32 days. The durability study is a long-term study that comprises wear analysis, lube oil analysis, and visual analysis; hence, it is also known as endurance testing of an engine [80-81].

1.6.1 Lube Oil Analysis

The lube oil analysis presents the total history of wear for internal parts of the engine. Lubricating oil reduces the friction among the sliding components of the engine. Therefore, the wear of internal parts is minimal. The cooling and entertainment of contaminants is also a function of the lube oil. In the present scenario, the lubricating oils are highly engineered products that comprise a base with a complex set of additives. These additives contain various constituents, which act as detergents, dispersants, wear inhibitors, antioxidants, and viscosity modifiers. These additives are used for maintaining the required quality of lubricating oils while running the engine. Viscosity is an essential property of lubricating oils. It should be within the allowable limit and high enough to keep the operation out of the boundary. Viscosity further decreases owing to the mixing of vegetable oil and dissolved wear materials in the lubricating oil. Different viscosity modifier additives are employed to stabilize the viscosity of the lubricating oil. The lubricating oil is replaced after a specific time for different fuels depending on the duration in which the quality of the oil blended with fuel remains good. The recommended hours of oil change with diesel fuels are 150 hours. The wear of different engine parts are evaluated by testing the lubricating oil for dissolved materials after a specific interval of time like 100 hours or 150 hours. The oil viscosity and dissolved materials in the oil provide details about the health and wear specifications of the engine.

1.6.2 Wear Analysis

Wear analysis is the measurement of wear for different sliding parts. The wear of each engine parts is evaluated on the basis of the difference in the measured dimensions both before and after the test. It is conducted to check the wear of engine parts after the test and to find whether it is within the allowable limit or not.

1.6.3 Visual Analysis

In the visual analysis, the engine parts are visualized to evaluate their condition both before and after the test. It provides details of carbon deposition, gum formation, and workability of the different internal components of the engine along with damage of specific engine parts. Moreover, the analysis provides specifications related to any particular observation while conducting the test.

CHAPTER 2

LITERATURE REVIEW

2.1 INTRODUCTION

Fossil fuel reserves are limited and depleting very fast; hence, it has become necessary to find the alternative of fossil fuels [81, 82, and 83]. Therefore, a comprehensive review has been carried out on the utilization of vegetable oils as alternative fuels.

2.2 HISTORY OF VEGETABLE OIL

In the late nineteenth century, vegetable oils were tested for their utilization as fuel in the compression ignition engine before invention of diesel engine. Dr. Rudolf Diesel invented the diesel engine in 1895 and presented his compression ignition engine fueled with peanut oil at the international exhibition in Paris [84]. He believed that the operation of engine with vegetable oils is a better substitute for the steam engine. Researchers from different countries conducted research on the use of vegetable oil during the 1920s and 1930s, although petroleum fuels were admitted as the fuel for engine [20, 85, 86].

By the 1970's, India began relying on the imported oil from middle-east, but after the twin crisis—1973 and 1978—owing to OPEC controlling and rapid growth of diesel vehicle, the potential of bio-fuels reentered the public consciousness [86]. A lot of research was conducted on different edible and non-edible vegetable oils to analyze their suitability as a fuel for internal combustion engine in the last four decades. Most of the research on the vegetable oil has been carried out on methyl ester of vegetable oil (biodiesel), and very few studies are available on the non-modified vegetable oil.

2.3 CHARACTERISTICS OF VEGETABLE OIL

Vegetable oils are triglycerides with branchless chains of different lengths and different degrees of saturation. These oils have a comparable energy density, heat of vaporization, cetane number, and stoichiometric air/fuel ratio with mineral diesel [59, 87]. Vegetable oils are heavy and bulky in their molecular weight and structure. The viscosity of these oils is very high owing to their massive molecular structure as compared to the existing mineral conventional fuel. The volatility of these oils is lower than the petroleum diesel fuel. The kinetic viscosity of these vegetable oils is between 25 cSt and 70 cSt at 40°C and the calorific value varies from 35, 000 kJ/kg to 40, 000 kJ/kg. The density and cetane number of vegetable oils vary from 900 kg/m^3 to 940 kg/m^3 and from 30 to 60, respectively. The density of vegetable oils is approximately 12% higher than diesel and the calorific value is about 10% lower than diesel [14]. The properties of different vegetable oils are given in Table 2.1.

Table 2.1 Properties of different vegetable oil [80, 88, and 89]

Oils	Kinematic viscosity (cSt 40° C)	Density (kg/m^3)	Heating Value (MJ/kg)	Cloud point (° C)	Pour point (° C)	Flash point (° C)	Cetane Number	Carbon Residue (w/w)
Diesel	2.75	835	42.00	-15	-20	66	47	0.001
Jatropha	49.9	921	39.7	16	8	240	40-45	0.64
Karanja	46.5	929	38.8	13.2	6	248	40	0.64
Rapeseed	37	911	39.7	-3.9	-31.7	246	37.5	0.30
Neem	57	938	39.4	8	2	295	47	0.96
Sunflower	33.9	916	39.6	7.2	-15	274	37.1	0.23
Soybean	32.6	914	39.6	-3.9	-12	254	38	0.27
Coconut	27.7	915	37.1	-	-	281	52	0.13
Peanut	39.6	902	39.7	12.8	-6.7	271	42	-
Linseed	27.2	923	39.3	1.7	-15	241	34.6	-
Palm	39.6	918	36.5	27	-15	271	42	0.043
Corn	34.9	909	39.5	-1.1	-40	277	37.6	0.24
Thumba	31.52	905	39.7	-	-	201	45	-
Babassu	30.3	946	-	20	-	150	38	-
Tallow	-	-	40	-	-	201	-	6.1

The cetane number shows the ignition quality of fuel; hence, higher cetane number implies shorter ignition delay and higher efficiency. Flashpoint shows the overall flammability hazard in the presence of air; hence, higher flash point indicates safe handling and storage. The pour point determines the measure of performance for fuels in cold conditions. It is lower for edible vegetable oil than diesel fuel but higher for non-edible vegetable oil [14]. The carbon residue indicates the deposition of carbon in the combustion chamber and fuel injector, which is always higher for vegetable oil than diesel fuel.

Pramanik [13] investigated the properties of Jatropha oil and its suitability as a potential alternative fuel for compression ignition engine. The properties of vegetable oils are much similar to mineral diesel, but the viscosity of these oils is higher than diesel (approximately ten times); hence, it would not be feasible and viable to directly use vegetable oil in the diesel engine without modification.

2.4 VEGETABLE OIL MODIFICATION TECHNIQUES

The four techniques—transesterification, heating/pyrolysis, microemulsion, and blending—are commonly used to reduce the viscosity of vegetable oils. A lot of research has been carried out on transesterification in comparison to other methods [90].

2.4.1 Transesterification (Bio-Diesel Formation)

Transesterification is a well-known technology for the reduction of viscosity of vegetable oil. Biodiesel is a generic name for the Transesterified vegetable oils. This method converts fatty acids of these oils into simple esters by using different chemicals (alcohols) and processing these oils through heat in the biodiesel conversion machine. The conversion process is lengthy and expensive. Many studies are available on the transformation of vegetable oil into biodiesel and the optimization of the conversion process.

Biodiesel obtained from rice bran oil contains high free fatty acids, which is obtained using an efficient two-step acid-catalyzed process [91]. Jatropha oil is transesterified using supercritical methanol in the absence of the catalyst under different temperature conditions [92]. Rubber seed oil is processed for biodiesel conversion through the utilization of an acid catalyst, which is followed by alkali catalyst in a single process and further developed a biodiesel conversion process for cheaper feedstock [93].

Antolin et al. [94] analyzed biodiesel production from sunflower oil. Many researchers have also suggested different methodologies for the production of biodiesel from the transesterification of different vegetable oils [95].

2.4.2 Blending/Dilution

Blending is a process in which vegetable oils are mixed with diesel to reduce the viscosity of the vegetable oil. The blended vegetable oil has properties almost similar to diesel depending upon the concentration of vegetable oil in the blend.

Forson et al. [45] conducted a performance study of Jatropha oil by blending it with diesel in a compression ignition engine. The results showed that the blending of 97.4% diesel with 2.6% Jatropha fuel produces maximum brake power and brake thermal efficiency.

Pryor et al. [96] carried out the short term and long term tests by blending soybean vegetable oil with diesel. Both the crude-degummed soybean oil and soybean ethyl ester were found suitable substitutes for diesel fuel. The long-term study showed various implications in association with compression ignition engine.

2.4.3 Heating/Pyrolysis

Heating is another viscosity reduction method. The viscosity of the vegetable oils decreases as the temperature of vegetable oils increases; and the viscosity of these oils comes close to diesel at 80-100°C. Few studies are available on the use of preheated vegetable oil as a fuel in the compression ignition engine.

Agrawal et al. [60] carried out a study on the preheated Jatropha oil. The viscosity of Jatropha vegetable oil was reduced by heating it using the waste heat of exhaust gases coming out from the engine. The performance and emission characteristics were found quite similar to diesel.

Nagaraja et al. [97] reported the performance and emission characteristics of the VCR engine by fuelling it with the preheated palm oil that is blended with diesel and further investigated the effect of compression ratio on the performance of the engine. Engine performance was better upon using 20% palm oil with 80% diesel than other blends of palm oil. The preheated vegetable oils have comparable viscosity, flash point, and pour point with diesel [98-99].

2.4.4 Micro Emulsion

The micro-emulsion is a method in which water is added at the time of extraction of oil to reduce the viscosity of plant oil. The plant oil modified through micro-emulsion is thermodynamically stable oil and has lower volumetric heating values than diesel fuels because of alcohol content. Alcohols have high latent heat of vaporization and tend to cool the combustion chamber, which results in reducing nozzle coking [63].

2.5 POTENTIAL OF NON-EDIBLE OIL IN INDIA

The use of edible vegetable oils as a fuel is banned in India as discussed in the previous section, so the scope is only for the non-edible vegetable oils as a fuel in compression ignition engine. The annual production of various non-edible vegetable oil in India is given in Table 2.2.

Table 2.2 Annual Production of Non-edible Oil Seeds in India [41, 69, 100]

Species	Annual Production (Thousand tonnes)	Oil Percentage

Neem	500	30
Kusum	80	34
Pilu	50	33
Wild Walnut	----	60-70
Undi	04	50-73
Thumba	100	21
Castor	250	45-50
Jatropha	200	50-60
Mohua	200	35-40
Sal	200	10-12
Linseed	150	35-45
Pongamia	60	30-40

2.6 USE OF VEGETABLE OIL AS A FUEL

A lot of research has been carried out with different edible and non-edible vegetable oils as fuel for compression ignition engine.

Ramadhas et al. [59] studied a compression ignition engine with rubber seed oil. Result revealed that the rubber oil blend has 5 % concentration of rubber oil along with diesel can be used directly as a fuel without any engine modification.

Pramanik [13] investigated the properties of transesterified Jatropha vegetable oil and the performance and emission characteristics of the engine fuelled with Jatropha vegetable oil. The result showed that the properties of Jatropha biodiesel are very close to diesel, except that of viscosity. Engine performance is also better with the use of Jatropha biodiesel than diesel.

Vellguth [37] presented the performance of a single cylinder DI diesel engine with different vegetable oils. The result showed that vegetable oils can be used directly as fuels in diesel engines for a short-term with little loss in efficiency, but operational difficulties will arise in the long-term operation of the engine with vegetable oils.

Forsion et al. [45] studied the effect of emulsified vegetable oil, which improves the atomization and lower combustion chamber temperatures and reduces the NOX emissions. This experiment was carried out on a single cylinder diesel engine and the performance of the engine was found to gradually become better with 2.6% blending of Jatropha oil with diesel.

Rao et al. [54] studied the performance of a compression ignition engine that is fueled with pure vegetable oils. The result revealed that the performance of the engine with these oils was very close to diesel and suggested that straight vegetable oils can be used as fuel in compression ignition engine without any significant problems.

Agarwal et al. [12] examined the effect of temperature on the viscosity of Jatropha vegetable oil and observed that with increase in temperature, there is significant reduction in viscosity. The effect on engine performance, emission, and combustion characteristics were also experimentally evaluated using the preheated (using the waste heat of the exhaust gases) Jatropha oil and its different blends along with diesel.

Suresh kumar et al. [18] studied the performance of compression ignition engine using Pongamia Pinata biodiesel, and the result revealed that the performance of engine using 40% of Pongamia Pinata along with diesel was found to be very close to the performance of engine using diesel.

Hossain et al. [14] presented a life cycle analysis of the use of different preheated pure vegetable oils as fuel in the compression ignition engine. The output to the input energy ratio of raw plant oils was approximately six times higher than fossil diesel.

Ramning et al. [16] analyzed the transesterification process for the production of Neem oil methyl ester using Central Composite Design and Response Surface Methodology and optimized the process parameters for reducing viscosity. To maximize yield and minimize the acid value, the optimum condition for the Esterification process is 16.832 cSt against 1:9 Alcohol to Oil Molar Ratio, 10 (minutes) Reaction Time and 3 ml H2SO4 Catalyst Concentration.

Jacobus et al. [102] studied a single cylinder compression ignition engine using many vegetable oils and compared the engine performance and emissions characteristics with respect to the utilization of each of these fuels. The result revealed that the engine performance using these fuels is found very close to diesel.

Bhatt et al. [103] studied a compression ignition engine using Karanja oil along with diesel. The results of using T10, T20, T30, and T40 blends of Karanj oil along with diesel as fuel were closer to that of diesel. The engine performance was found to improve at higher compression ratio.

Alten et al. [104] investigated a DI engine using sesame oil biodiesel. The result showed that using sesame oil-diesel blends, the engine performance and emission characteristics were nearer to that of using diesel as fuel. Lower blends of Sesame oil with diesel can substitute diesel in a compression ignition engine without any modification and change in the maintenance schedule.

Herchel et al. [105] studied the performance and emission characteristics of a single cylinder diesel engine using different coconut oil blends along with diesel. Smoke and NOx emissions were found to decrease with the increase in coconut oil concentration in diesel.

Varaprasad et al. [106] conducted a comparative study of the performance and emission characteristics of a single cylinder compression ignition engine using Jatropha oil and Jatropha biodiesel. Brake thermal efficiency of Jatropha biodiesel was found to be superior to that of pure Jatropha oil and inferior to diesel. In comparison to pure Jatropha oil, it was observed that the use of Jatropha biodiesel resulted in higher NOx emission and lower smoke.

Murugesan et al. [107] presented the opportunities and prospects of introducing vegetable oils and their derivatives as fuel in diesel engines. Performance, combustion, and emission analysis were also evaluated in this study. Biodiesel conversion technique was further optimized and the result showed that biodiesel fuel causes less emission in comparison to diesel.

Norbert et al. [108] carried out a study on compression ignition engine using pure rapeseed oil. The torque and engine power output using straight rapeseed oil was found inferior to diesel, but emissions like HC and CO were higher for rapeseed oil than diesel.

Sahabuddin et al. [109] reviewed the studies conducted for the last 10 years related to the combustion characteristics of bio-fuels. The investigation results reported that as compared to diesel, biodiesel has early start of combustion and shorter ignition delay owing to higher cetane number and lower compressibility of vegetable oil. Lower heat release rate has been found for vegetable oil due to the inferior calorific value of these oils.

Rakopoulos et al. [15] studied the performance and emission characteristics for a single cylinder high-speed direct ignition engine using the cottonseed oil and blending its biodiesel with Diethyl Ether and n-butanol. The result revealed that the addition of n-butanol and Diethyl Ether in the cottonseed oil and its biodiesel improved the performance.

2.7 THUMBA VEGETABLE OIL AS A FUEL

Citrullus Colocyntis is a species of family Cucurbitaceous, which is commonly known as ‘Bitter Apple’ (in English), Thumba, (in Marathi), and Indrayan (in Hindi). Thumba plant grows naturally in the rainy season, and its fruits are available in winter. This plant is a native of turkey and is also found in Africa, Sahara, Egypt, and southern European countries. In India, it grows in various states like Madhya Pradesh, Rajasthan, Gujarat, Haryana, and Uttar Pradesh. Thumba is a creeper type of plant and grows well in sandy soil. Crop cycle of this plant is only six months. Moreover, it grows with other crops; hence, there is no need of any special care. The average yield is about 2500-3500 kg seeds per hectare with minimum inputs and these seeds contain 20-25% of the golden yellow-brown oil.

Thumba oil is one of the non-edible vegetable oils. Local soap industries consume this oil as a soap binder and lubricant. In comparison to Jatropha and Karanj oils, the properties of this plant oil are closer to the properties of mineral diesel [110]. Thumba plant of small crop cycle and various uses can play a vital role in the Indian rural economy. Huge research has been conducted on Thumba biodiesel, but very few studies are available on Thumba as a vegetable oil. The properties and fatty acid composition of Thumba oil are as given in Table 2.3 and Table 2.4. Linoleic acids are maximum (55.9 %) in Thumba oil [96, 111, 112, 113]. The biophysical limit of thumba oil has been presented in Table 2.5.

Table 2.3 Properties of Thumba Oil and Diesel [96, 111]

Property	Thumba Oil	Diesel
Molecular weight	872.61	200
State	Liquid	Liquid
Colour	Golden Yellow Brown	Light Brown
Cetane No.	45	45-50
Net Calorific Value kJ/kg	39, 700	42, 000
Viscosity at 40°C (cSt)	36.9	2.5
Sp. Gravity	0.905	0.835
Free fatty acid	>1%	NA
Pour Point (°C)	-5	-23
Flashpoint (°C)	201	80

Fire point	122	64
Iodine Value	126.4	N.A
Saponification Value	187.9	N.A.
Unsaponifiable Matter (wt %)	2.25	_
Acid Value	2.2	_
Peroxide Value	0.1	_

Table 2.4 Fatty acid profile of Thumba oil [20, 96, 111]

Sr. No.	Fatty acids	wt%
1	Palmitic (C_{16} H_{34})	10.3
2	Stearic(C_{18} H_{38})	8.00
3	Oleic (C_{18} H_{36})	24.5
4	Linoleic (C_{18} H_{34})	55.9
5	Other	1.03

Table 2.5 Biophysical limits of Thumba Plant

Sr. No.	Elevation range	Lower: sea level upper: about 1,500 m (4,921 ft)
1	Mean annual rainfall	Lower: 250 mm (10 in) upper: 4,000 mm (160 in)
2	Rainfall pattern	A crop adapted to arid zones
3	Dry season duration	A desert plant, giving evidence of the dominion of life even in such arid regions
4	Mean annual temperature	Lower: 15°C upper: 48°C

Pal et al. [78] extracted Thumba biodiesel by low-frequency ultrasound energy process and evaluated the physical and chemical properties of Thumba oil and its biodiesel. The ultrasound energy process is industrially viable, timesaving, relatively simple, efficient, and eco-friendly method.

Figure 2.1 Thumba tree with fruit

Figure 2.2 Thumba seed

Jain et al. [111] presented the performance and emissions characteristics of single cylinder diesel engine fuelled with Thumba biodiesel. The experiment carried out using T40 and T20 Thumba biodiesel blends with diesel. The results showed that the performance of T40 and T20 are better than diesel, and it was further observed that in comparison to diesel, the emissions of Thumba blends are inferior.

Mathur et al. [20] optimized biodiesel production from Thumba oil through the transesterification process. The maximum yield of Thumba methyl ester was obtained at 6:1 molar ratio, 0.75% of KOH at 65°C temperature with 250 R.P.M and 70 Min. of .reaction time.

Harari et al. [113] studied the performance and emission of a single cylinder compression ignition engine using Thumba biodiesel as an alternative fuel to diesel.

Mathur et al. [88] reported the economics of Thumba biodiesel and its formulation technique and suggested to conduct further research of using Thumba oil as fuel on multi-cylinder engines and other diesel engines used in the agriculture and transport sector. Furthermore, the long-term test is necessary to evaluate the durability of the engine using vegetable oils as fuel. Moreover, the production of Thumba biodiesel needs to be improved in the future to promote the properties of Thumba biodiesel, which in turn will encourage more research and development in biodiesel resources and engine design [20, 88 and 115].

Lal et al. [116] studied the performance of a diesel engine fuelled with Thumba biodiesel blends with diesel. The study was conducted using B5 to B25 with the difference of 5 % concentration of Thumba biodiesel and the results of these blends were compared with Jatropha biodiesel, mustard biodiesel, and castor biodiesel.

Sharma et al. [117] studied the Thumba methyl ester production process through transesterification. Methanol and potassium hydroxide was used as catalyst in this biodiesel development process. The physical and chemical properties of the Thumba biodiesel were evaluated and the characteristics were found comparable to diesel.

Kumar et al. [118] evaluated the effect of temperature, catalyst concentration, amount of methanol, and reaction time on the biodiesel production process and optimized these parameters to maximize yields.

Sharma et al. [119] studied the biodiesel conversion technologies and various experimental test rigs such as power ultrasound, hydrodynamic cavitation, and depending on the gathered information from different studies, the supercritical methanol processes were designed to produce biodiesel using Thumba oil. The rate of the alkyl ester formation using low-frequency and high-intensity ultrasound under ambient conditions was found to be higher than the mechanical stirring method. Cavitation condition identical to acoustic cavitation was generated in hydrodynamic cavitation (HC), and it had better impact on the mixing of immiscible liquids, thus resulting in reducing the reaction time and increasing yield. Further, it was observed that scale-up of hydrodynamic cavitation has better opportunities than the ultrasonic reactor because of the simple process and geometric details of the reactor. Hydrodynamic cavitation was found to be a cheaper alternative than the conventional mechanical stirring method for the consumption of energy, as it required approximately half the total cost associated with the traditional method.

Ganapathi et al. [120] improved the brake thermal efficiency of engine by modifying the engine. Engine Piston, cylinder liner, and cylinder head were insulated with the Partially Stabilized Zirconia (PSZ) coating to reduce the engine heat losses.

2.8 DURABILITY STUDY

The vegetable oils as a fuel for compression ignition engine were assessed successfully for short-term use. Nevertheless, many problems were observed during the long-term use of vegetable oils as fuel; therefore, it is compulsory to conduct a durability test before successful completion of the testing of vegetable oil as a fuel.

Wander et al. [21] presented a long run study (durability study) of 1000 hours for three single cylinder diesel engines using pure diesel oil, pure soya methyl ester, and pure castor oil methyl ester. It was observed that the use of pure methyl ester fuels is acceptable for long-term use without any engine modification.

Basinger et al. [43] studied the longevity implications of using pure plant oil in an indirect injection slow speed agro-processing stationary engine. A model was also developed to characterize the engine wear and estimate the change in lube oil frequency.

Shano et al. [121] compared the results of durability test on compression ignition engine with Jatropha biodiesel and diesel fuel. The performance of engine was found better with Jatropha biodiesel than diesel owing to lower engine operating temperature.

Sharma [87] reported the long-term implications of a single cylinder diesel engine using Neem oil. The results of elemental analysis showed normal engine wear and it was observed from the lubricating oil analysis that the decrease in the viscosity of the lubricating oil was within limits (less than 25% at 40°C).

Charles et al. [122] reported the performance and durability analysis of eight biodiesel fuels. The long-term performance testing conducted consisted of 200 hours test for each fuel. Problems like excessive wear filter clogging, piston ring seizure, excessive oil dilution, or excess injector tip coking were observed. Injector coking was least for diesel followed by ethyl ester with slight coking differences.

Agarwal et al. [123] assessed wear in compression ignition engine fuelled with Thumba biodiesel, and found no excessive wear, except the deposition of heavy carbon on different parts of the engine.

2.9 MOTIVATION AND RESEARCH GAP

Conventional fuels are scarce and depleting very fast. Vegetable oils are economical, green, and renewable fuels; but these oils cannot be used directly as fuel owing to higher viscosity of these oils. Therefore, the modification of these oils is necessary. Transesterification is the most popular technology for modifying these oils, but it is not feasible in the rural areas of developing countries due to logistic problems. Preheating is another technology to reduce the viscosity of vegetable oils. Heat is available in the form of waste heat of exhaust gases of the same compression ignition engine in which these oils are used as fuel. Few studies are also available on the use of preheated vegetable oil as an alternative fuel for compression ignition engine.

Thumba oil is a non-edible vegetable oil, and the physical and chemical characteristics of Thumba oil are close to diesel. Thumba vegetable oil is an underutilized and under assessed fuel as per the reviewed literature. However, very few comprehensive studies are available with the use of Thumba oil as a fuel on a single cylinder VCR engine.

Huge research is available on the characterization of Thumba oil and Thumba biodiesel along with the most advantageous technology for Thumba oil conversion. The available studies on engine performance, combustion, and emission characteristics are associated with Thumba biodiesel; and only few of them focus on the use of pure Thumba oil and its blend with diesel.

The long-term study is to be carried out for regular use of any vegetable oil as a fuel in compression ignition engines. The available experimental studies on Thumba vegetable oil are short-term studies. Studies conducted on the use of Thumba oil show that none of them addresses endurance testing as well as lubricating oil analysis.

2.10 OBJECTIVES OF CURRENT RESEARCH

The present research work was carried out to comprehensively assess the suitability of Thumba vegetable oil as a fuel. Combination of heating and blending is used to reduce the viscosity of Thumba oil. Waste engine exhaust gases were used to preheat Thumba oil blends with diesel. The performance, combustion, and emission behavior of a diesel engine fuelled with Thumba oil blends with diesel were carried to optimize the Thumba oil blend. The engine durability test using the optimized Thumba blend with diesel was further carried out to observe the longevity implications.

The major objectives of the research study were as follows.

1. To develop exhaust gas operated fuel heating system for a single cylinder, and variable compression ratio (VCR) compression ignition engine operated on Thumba oil-diesel blends.
2. To carry out combustion, performance, and emission studies for optimizing the engine operating parameters for the developed system.
3. To carry out an endurance test on the optimized engine for assessing the feasibility of long-term use.

CHAPTER 3

EXPERIMENTAL SETUP AND RESEARCH METHODOLOGY

For accomplishing the research objective, we used a four-stroke, water cooled, naturally aspirated, direct injection, and variable compression ratio single cylinder diesel engine. An eddy current dynamometer was attached with this setup for the application of load. Piezoelectric sensors were used for the measurement of the cylinder and fuel line pressure. The temperature at various points was measured using Thermocouples, and Rotameter was used for measuring the water flow rate at multiple inlet and outlet positions. A compact diagnostic five-gas analyzer was used to measure smoke opacity, CO, CO2, HC, and NOx.

3.1 EXPERIMENTAL SETUP

A four stroke, water cooled, stationary, direct injection, naturally aspirated, variable compression ratio (VCR) single cylinder diesel engine was used for this study, and it can run with multiple fuels. The basic engine with capacity of 5 kW and model is TV1 was manufactured by Kirloskar brothers limited. It is modified for analyzing VCR engine of size 3.5 kW using apex innovation. The engine considered for research was equipped with an eddy current dynamometer for load application, an encoder for measuring the RPM, the pressure and temperature transducers for measuring pressure and temperature, an air and fuel measuring system, and a separate panel with the computerized measuring system. The set up for testing is attached to an alternate fuel supply system, which is equipped with shell and tube type heat exchanger to heat the fuel using waste engine exhaust gases. The photographic view of the experimental set up is as shown in Figure 3.1.

3.1.1 Diesel Engine

For this study, we employed a 3.5 kW VCR diesel engine and modified it into VCR engine with the change in the inclination of the cylinder head. Due to the tilt of the cylinder head, the clearance volume gets changed; hence, we changed the compression ratio by raising and lowering the bore. After making the modification, the compression ratio can vary from 12 to 22, which was earlier from 12 to18. Specification of VCR diesel engine is given in Table 3.1. Kirloskar research engine is also shown in Figure 3.2.

Figure 3.1 Photographic front view and rear view of test engine setup

Figure 3.2 Kirloskar engine (model TV1)

Table 3.1 Engine Specification

Sr. No.	Description	Specification
1	Model	TV1
2	Make	Kirloskar Oil Engine Ltd., Pune
3	Type	Four stroke, water cooled, Diesel
4	No. of cylinder	One
5	Bore	87.5 mm
6	Stroke	110 mm
7	Cubic capacity	0.661 liters
8	Peak pressure	77.5 kg/cm^2
9	Max. speed	2000 rpm
10	Direction of rotation	Clockwise (looking from flywheel end side)
11	Minimum idle speed	750 rpm
12	Minimum operating speed	1200 rpm
13	Fuel timing for engine	23°CA BTDC

14	Inlet valve open	4.5°CA BTDC
15	Inlet valve close	35.5°CA ATDC
16	Exhaust valve open	35.5°CA BTDC
17	Exhaust valve close	4.5°CA ATDC
18	Inlet valve clearance	0.18 mm
19	Exhaust valve clearance	0.20 mm
20	Bumping clearance	0.046 ”- 0.052”
21	Lubricating system	Force feed system
22	Break mean effective pressure at 1500 rpm	6.35 kg/cm^2
23	Lubricating oil pump	Gear type
24	Lubricating oil pump delivery	6.5 liters/ minute
25	Sump capacity	2.7 liter
26	Connecting rod length	234 mm

3.1.2 Dynamometer

A water-cooled eddy current dynamometer was attached to the engine for the application of load. A strain gauge type of load cell was attached to the dynamometer. The load cell was used for measuring the exerted rotational torque. A rotor was mounted on a shaft that rotates in a casing with the support of bearing in the unit of the dynamometer. A water supply system is attached to the dynamometer to prevent its’ overheating. The specifications of the dynamometer are given in Table 3.2.

Table 3.2 Technical specification of dynamometer

Sr. No.	Description	Specification
1	Model	AG10
2	Make	Saj Test Plant Pvt. Ltd.
3	Pressure lbf/in2	23
4	Air gap mm	0.77/0.63
5	Torque Nm	11.5
6	Hot coil voltage max.	60
7	Continuous current amps	5.0
8	Cold resistance ohms	9.8

9	Speed max.	10000rpm
10	Load	3.5kg
11	Weight	130kg

3.1.3 Alternate Fuel Supply System

Thumba vegetable oil and its blends were preheated before their inoculation into the cylinder. The two fuel tanks were already equipped with a separate panel. One of the tanks was for diesel fuel and the other was for petrol or additional fuel, but the supply line was common for both the fuels.

An alternate fuel supply line was developed for supplying the heated Thumba oil and its blend as a fuel into the engine cylinder. For heating the Thumba oil and its blend, a shell and tube type heat exchanger was attached with alternate fuel supply line.

A silicon pipe was used to supply the preheated Thumba oil and its blend with diesel. The silicon pipes can withstand the temperature of 250°C. The alternate fuel supply line with the heat exchanger is shown in Figure. 3.3.

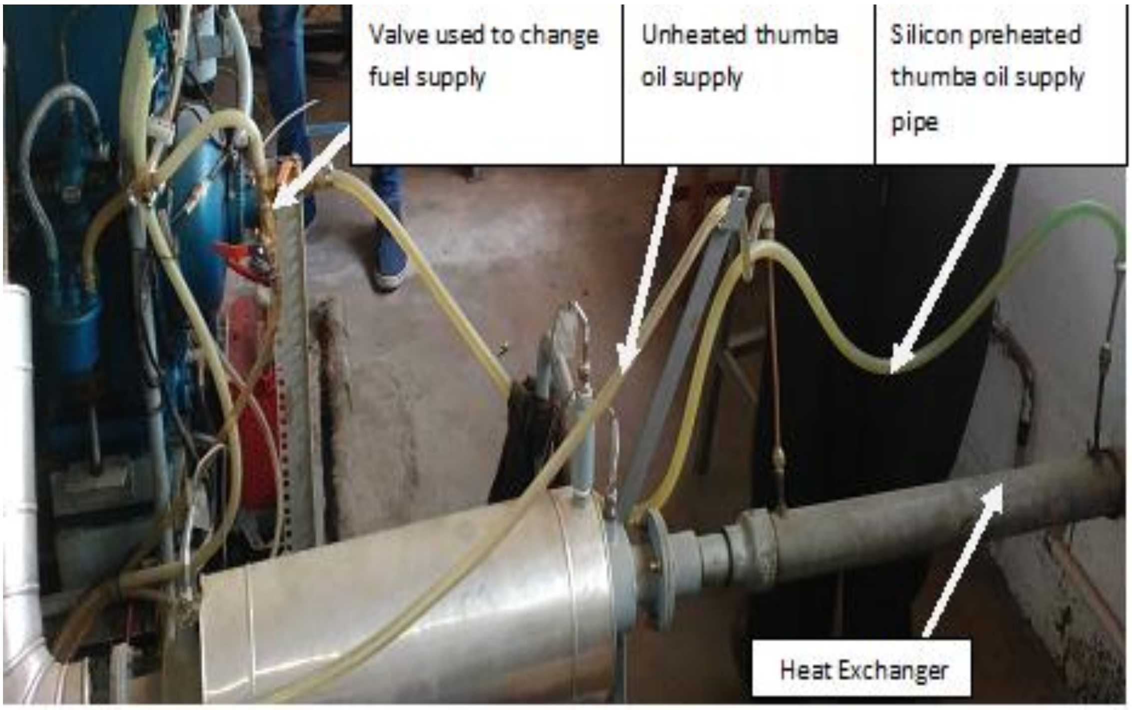


Figure.3.3 Alternate fuel supply system with attached heat exchanger

3.1.3.1 Design and Development of Heat Exchangers

Five different sizes of shell and tube type heat exchangers were developed in the institute's workshop to opt the suitable heat exchangers for heating the fuel. The material of shell and tubes of the heat exchanger are galvanized iron and copper, respectively. The designed shell of heat exchanger was 2 Inches in diameter and 1 meter in length. A tube of 5 mm diameter was used in the heat exchanger. The developed heat exchangers are shown in Figure. 3.4.

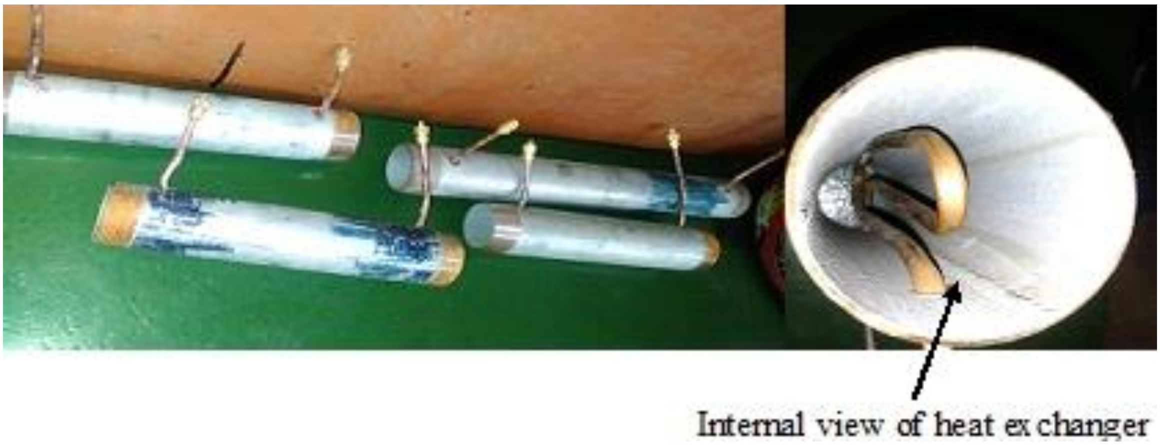


Figure 3.4 Developed heat exchangers

3.1.4 Air Flow Measurement

An airbox (0.5*0.5*0.6 m3) was used to measure the airflow and it was mounted on the separate panel that was used to make the air flow steady for carrying out the accurate measurement of airflow. An orifice of 25 mm diameter was fitted on the side wall of boxes, and the CD of the hole was 0.6. An air inlet pipe of the engine was connected to the outlet of the airbox. Air was induced in the airbox owing to the pressure difference between the atmospheric air and the air present inside the box. An air filter was attached between the engine inlet and box. The quantity of air induced into the box was measured with the help of a manometer mounted on the front of a separate panel. A U-type manometer was used within the range 100-0-100 mm (Make –Apex, Model – MX-104). The air induced could be sensed through the sensor based hotwire flow meter that works on the principle of change in resistance while heating. The passing of air over the wire resulted in cooling. The current required for maintaining the constancy of wire's electrical resistance was directly proportional to the mass of air flowing. The current signals were stored on the computer using a data acquisition system, which can also be calculated manually depending on the following relation.

Air induced / second = Cd * A * (2 * g * H * ρw / ρa) 1/2 (m3/s)

Where, Cd = 0.6

A = Cross section area of orifice m3

H = Manometer reading (meters)

w = Density of water (1000 kg/cubic meter)

a = Density of air (1.157 kg/cubic meter)

3.1.5 Fuel Flow Measurement

Two fuel tanks were mounted on the top of the separate panel of this experimental setup. A pipe connected both the tanks through a tee valve, and one fuel could only flow at a time through this pipe, which was further attached to a calibrated glass burette system for measuring the flow. Another tee valve was used for connecting the fuel tank and burette to regulate the flow either from a tank or burette. For measuring the fuel flow quantity, the valve was directed toward the burette for ensuring the flow of fuel through it. The fuel flow rate was digitally measured through a multi-function microprocessor-based fuel system. This flow sensed again by the sensor attached perceived the flow rate of fuel and sent signals to the computer for acquiring data. Fuel flow can even be manually measured with the help of a stopwatch.

3.1.6 Water Flow Measurement

Water flow was measured for heat calculations. Water was circulated in the cylinder liner to cool the engine and in the calorimeter to reduce the temperature of exhaust gases. With the help of the centrifugal pump, water was supplied to the calorimeter and dynamometer, and rotameter (Make-Eureka Model-PG 6, Range 40-400 liter per hour) was used for measuring the water flow.

3.1.7 Pressure Measurement System

Piezoelectric sensors were used for the measurement of fuel line and cylinder pressure. These miniature sensors (make: PCB Piezotronics, Inc. New York, Models SM111A22 and M108A02) are intended to measure pressure at various points. This versatile transducer measured compression, combustion, explosion, pulsation, cavitations, blast, pneumatic, hydraulic, and other such pressures. It is necessary to supply the constant current of 2 mA to 20 mA at +20 VDC to +30 VDC through a current regulating diode or equivalent circuit to the sensor. Most of the signal conditioners manufactured by PCB have adjustable current features that permit the flow of input currents from 2 mA to 20 mA. In general, the lower current ranges are chosen for the lowest noise (best resolution). When driving long cables (up to several thousand feet), use the higher current, up to 20 mA maximum.

3.1.8 Temperature Measurement

The K type thermocouples (Calibration: 0-1200°C.) and Pt100 type RTD (Calibration: 0-100°C) were used to measure the temperatures at different points in the engine. The output of these points was attached to the computer through the data acquisition system.

3.1.9 Data Acquisition System

The data acquisition system acquires relevant data and stores it on a personal computer for further analysis. This data is the sampling of the real word for generation and manipulation of the data. The data stored in signals and waveform can be processed to obtain the desired information. Sensors are the component of the data acquisition system that can convert any measured parameter to an electrical signal. A device used for it is NI USB-6210 Bus Powered M Series.

3.1.10 Exhaust Gases Measurement

Exhaust gases were measured using Five Gas Analyzer combined with Smoke Meter, which is known as a Compact Diagnostic system (Model: AVL DITEST GAS 1000 BL-2014). Two separate probes were used for measuring the smoke opacity and five gases discharged from the exhaust. CO, CO_2, and HC were measured using the non-dispersive infrared analyzers. The quantity of infrared energy absorbed in a sample cell is proportional to the concentration of the compound in the cell. O_2 and NOX were measured using the electrochemical method. Visible light absorbed from the exhaust gases gives the measure of smoke opacity. The Compact Diagnostic system is shown in Figure 3.5.

Figure 3.5 AVL Five Gas Analyzer combined with Smoke Meter

3.2 RESEARCH METHODOLOGY AND EXPERIMENTAL PROCEDURE

3.2.1 Research Methodology

- Conducting a comprehensive literature survey and literature review to identify the research objective
- Procurement of research engine for carrying out the research work along with accessories such as combined Smoke Meter and Five Gas Analyzer
- Design and development of shell and tube type heat exchanger for preheating the fuel using the waste heat of engine exhaust gases
- Development of the experimental set up with alternate fuel supply system and fuel preheating arrangement
- Development of various concentrations of Thumba oil blends with diesel in the laboratory for its use as fuel
- Formation of Thumba oil blends through the variations in the concentration of Thumba oil in diesel fuel from 10% to 100%; and therefore, the variations were designated as T10 and T100, respectively. Blends were formulated with the difference of 10% concentration. T10 means 10% Thumba oil and 90% diesel and T100 means 100% Thumba oil and 0% diesel {Tx = x% Thumba oil+ (100-x) % diesel}, where the values of x are 10, 20, 30, …, 100. The Thumba oil blends with diesel are shown in Figure 3.6.
- Experimentally evaluated the engine performance, combustion, and emission characteristics with diesel for the base case
- Optimization of engine operation parameters like compression ratio, injection pressure, and injection timing to find the optimum parameters using diesel

- Using the different concentrations of Thumba oil with diesel, the engine test run was conducted to identify an optimized concentration of Thumba oil with diesel.
- Optimization of engine operation parameters using the preheated optimized Thumba oil blend with diesel
- Experimentally evaluated the engine performance, combustion, and emission characteristics with preheated and unheated optimized concentration of Thumba oil blend with diesel
- Comparison of the engine performance, combustion, and emission characteristics using diesel and the preheated and unheated optimized Thumba-diesel blend.
- Subsequent to the performance, combustion and emission analysis, the engine test run was conducted for long-term durability (endurance testing) to identify the long-term implications of the optimized engine for using the preheated T20 Thumba oil and diesel blend.
- Documentation of the findings and analysis of the detailed experimentation and report writing of the research conducted

Figure 3.6 Developed Thumba oil blends with diesel

3.2.2 Experimental Procedure

We experimentally evaluated the engine performance, combustion, and emission characteristics using diesel for the base data. Brake thermal efficiency, brake specific fuel consumption, and exhaust gas temperature were studied to evaluate the performance of the engine. Cylinder pressure, heat release rate, and rate of pressure rise were observed to assess the combustion characteristics of the engine. Smoke opacity, CO, CO_2, HC, and NOx emissions were identified to evaluate the emission characteristics of the engine. Several tests were conducted for analyzing the performance, emission, and combustion characteristics of the engine fuelled with diesel as well as the preheated and unheated T20 Thumba oil-diesel blends.

The engine operation was optimized using the diesel fuel for different engine parameters such as compression ratio, injection pressure, and injection timing. For optimum compression ratio, the engine was tested for different compression ratio, i.e., 16, 17, 18, 19, 20, 21, and 22. For each compression ratio, the test run of engine was done with varying loads from 0% to 100%; and in each test run, the load was increases by 25% keeping constant the injection pressure, injection timing, and engine speed at 203 bar, 23° Crank Angle (CA) Before Top Dead Centre (BTDC), and 1500 RPM, respectively. The manufacturer prescribed these optimized parameters in the experiment manual.

After the optimization of the compression ratio, the engine was tested for injection pressures from 170 bar to 230 bar (170, 180, 190,203, 210, 220, and 230 bars), while keeping constant the injection timing, compression ratio, and engine speed. Injection timing and engine speeds were 23° CA BTDC and 1500 RPM, respectively. The obtained optimized compression ratio was considered as constant compression ratio in the optimization of injection pressure and injection timing. For each injection pressure, the engine test run was conducted with varying load from 0% to 100%—the load was increases by 25% in each test run (0%, 25%, 50%, 75%, and 100% load).

Finally, the engine was tested at different injection timings, i.e., 18°, 20°, 23°, 27°, and 30°CA BTDC with the same variation of the load, while keeping constant the compression ratio, injection pressure, and engine speed.

After the initial testing of some of the engine parameters, the results were found to be very poor, therefore, the compression ratio of 16 and 17, the injection pressure of 170, 180, 220, and 230 bar, and the injection timing of 18° and 30° CA BTDC were not considered in this study. The final used operating parameters are listed below in Table 3.3. The same experimental procedure was followed to optimize the engine parameters using the optimized preheated optimized Thumba oil blend with diesel.

Table 3.3 Engine parameters considered for the study

Sr. No.	Name of parameter	Range	Unit
1	Load	0% 25% 50% 75% 100%	-
2	Compression Ratio	18 19 20 21 22	-
3	Injection Pressure	190 203 210	Bar
4	Injection Timing	20 23 27	˚CA BTDC
5	Engine Speed	1500	RPM

The diesel fuel was tested in the Variable Compression Ratio (VCR) engine to accomplish the base data. The preheated Thumba oil and its blends with diesel were further tested in the engine for optimizing the preheated Thumba oil blend with diesel. The Thumba oil blends were heated using engine waste exhaust gases before these oils entered the engine fuel pump.

The primary study revealed that some of the Thumba oil blends T40, T60, T70, T80, and T90 demonstrated inferior results; hence, those blends were not considered within the scope of this study. Pure Thumba oil (T100) was used to compare it with other blends and diesel.

Various engine input parameters and their ranges to be used for optimizing Thumba oil blend have been listed below in Table 3.4. Finally, the performance, combustion, and emission characteristics of the engine were evaluated using the unheated T20 Thumba oil blend with diesel.

Table 3.4 Various engine input parameters and their range to be used to optimize Thumba oil blend

Sr. No.	Parameters	Unit	Range Values
1	Thumba oil blends	-	T10, B 20, T30, T50, and T100
2	Load Range	% of rated load	0, 25, 50, 75 and 100
3	Engine Speed	RPM	1500
4	Compression Ratio	-	22
5	Injection Pressure	Bar	203
6	Injection Timing	°CA BTDC	23

The engine durability test was carried out to observe the long-term implications of using the preheated T20 Thumba oil blend in the optimized engine. The engine was dismantled for measuring the dimensions of the internal components, and the engine sump was filled with fresh lubricating oil before commencement of the durability test.

The engine run of 512 hours required to be completed in 32 days (sixteen hours per day) without a break as per the guideline of IS-10000 (Part-9). The engine was started at 6 AM and stopped at 10 PM. For the initial20 minutes, diesel was used to warm up the engine and then the preheated T20 Thumba oil blend was used to run the engine for whole day with different loads as per the IS- 10000 (Part-9). It was again shifted to the diesel mode for the last 20 minutes to purge the blended oil from the line and filter, and subsequently, the engine was shut down for the day.

The used lubricating oil was replaced with fresh lubricating oil after every 150 hours of test runs, and 100 ml sample of the lubricating oil was tested from ISO Certified and Accredited Laboratory (CEG, Test laboratory) for scrutinizing the dissolved materials. The viscosity degradation of used lubricating oil was evaluated after every 50 hours of running the engine. The engine was dismantled again to estimate the wear of components after completion of the test. The wear of parts was shown by the difference in dimensions before and after the test.

CHAPTER 4

RESULTS AND DISCUSSION

(PART 1- Performance, Emission and Combustion studies)

4.1 INTRODUCTION

The findings of the experimental study are documented in two different sections. The comparison of the performance, combustion, and emission behavior of the engine fueled with diesel, preheated and unheated Thumba oil Diesel blends are presented in this chapter. The findings of the long-term studies (Durability studies) of the engine fuelled with preheated T20 (Thumba oil diesel blend) are presented in the next chapter. The results are represented through graphs and tables.

4.2 ENGINE CHARACTERISTICS FUELED WITH DIESEL

The brake thermal efficiency, brake specific fuel consumption, and exhaust gas temperature are measured to indicate the performance. The measurement of Smoke, NOx, CO, CO_2, and HC was conducted to compare the emissions. The combustion characteristics are indicated by various parameters such as cylinder pressure, heat release rate, delay period, combustion duration, and the start of combustion and injection. The engine parameters are optimized for obtaining the characteristics of engine at the optimized operating conditions.

4.2.1 Optimization for Compression Ratio

The experimental study was conducted with different compression ratio (18, 19, 20, 21 and 22) while keeping constant injection pressure, injection timings, and engine speed at 203 bar, 23°CA BTDC, and 1500 RPM, respectively.

4.2.1.1 Performance Studies

Brake Thermal Efficiency (BTE) is a measure of net shaft power output of an engine developed from the supplied fuel heat. The variation of BTE with the load at different compression ratios is shown in Figure 4.1. BTE increases with increase in the percentage of load while keeping the compression ratio constant. BTE also increases with increase in compression ratio, but BTE at the compression ratio of 22 was found to be inferior to the compression ratio of 21, which is possibly owing to increase in friction losses at high compression ratio. Maximum BTE was observed at the compression ratio of 21 probably because of superior and complete combustion of fuel.

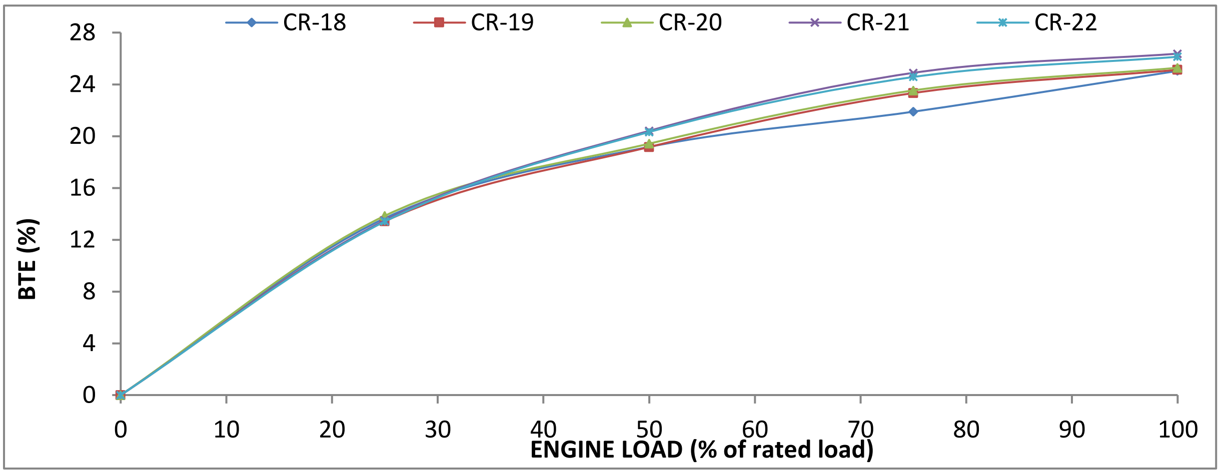


Figure 4.1Variation of Break Thermal Efficiency with percentage increase in Load at different Compressions Ratios upon fuelling the engine with diesel.

Brake Specific Fuel Consumption (BSFC) is a parameter that measures the conversion of fuel energy for enhancement in work. The variation of the BSFC with the load at different compression ratios has been shown in Figure 4.2. BSFC decreases with increase in the percentage of load while keeping the compression ratio constant. BSFC even decreases with increase in compression ratio, but it was found to be the lowest for the compression ratio of 21 possibly owing to proper mixing of fuel and atomization with superior combustion quality.

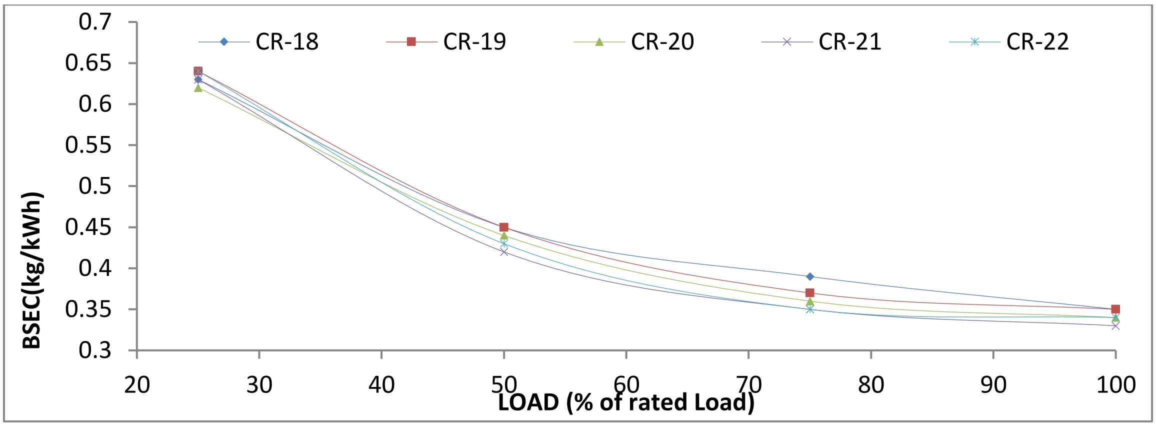


Figure 4.2 Variation of BSFC with percentage increase in Load for different Compression Ratios upon fuelling the engine with diesel

Exhaust Gas Temperature (EGT) is another performance parameter, which provides information about combustion and average cylinder temperature. The variation of EGT with the load at different compression ratios is shown in Figure 4.3. The figure shows that EGT increases with the increase in the load while keeping the compression ratio constant owing to decrease in the air-fuel ratio with the increase in the load. The graph further indicates that EGT decreases as the compression ratio increase, but it was found to be the lowest for the compression ratio of 21 because of the efficient utilization of fuel energy at this compression ratio.

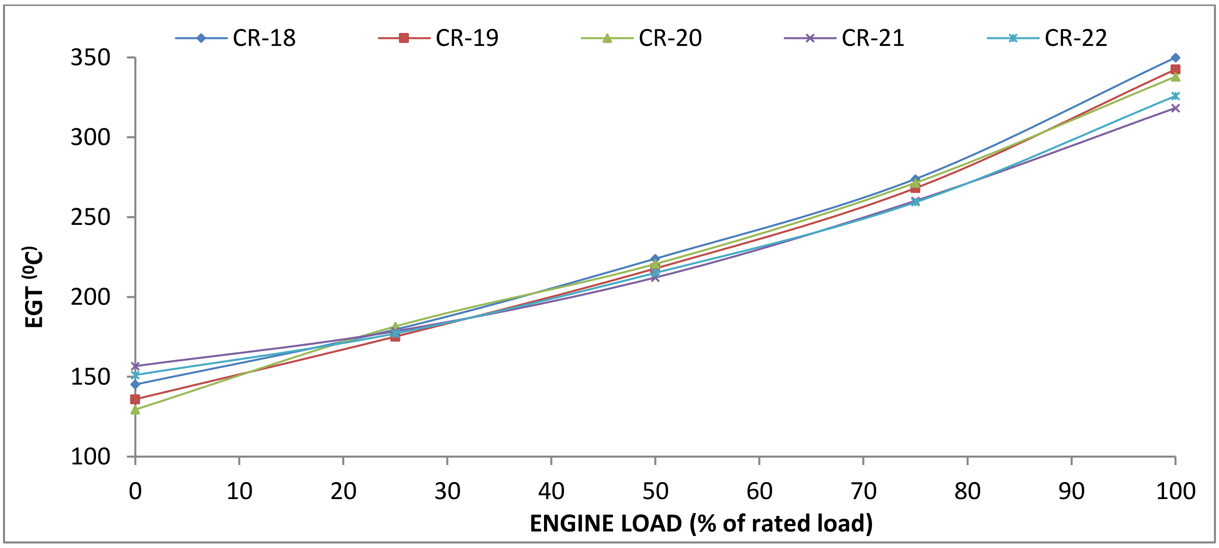


Figure 4.3 Variation of EGT with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel

4.2.1.2 Emission Studies

Smoke opacity, CO, CO_2, HC, and NOx emissions are analyzed and presented in this part of the study. Mostly, smoke opacity and NOx are of severe concern in the compression ignition engines that are operated with diesel fuel.

Smoke is generally produced during full load and overloading engine operating conditions. It is black in color and graphite structure soot, which is produced during thermal cracking. The variation of smoke opacity with the load at different compression ratios is shown in Figure 4.4. Smoke opacity increased with increase in the percentage of load while keeping the compression ratio constant, probably because of higher combustion temperature at higher load. Higher combustion temperature helps in the formation of smoke. Smoke opacity at the compression ratio of 21 was found to be minimum owing to superior combustion and better mixing of air and fuel.

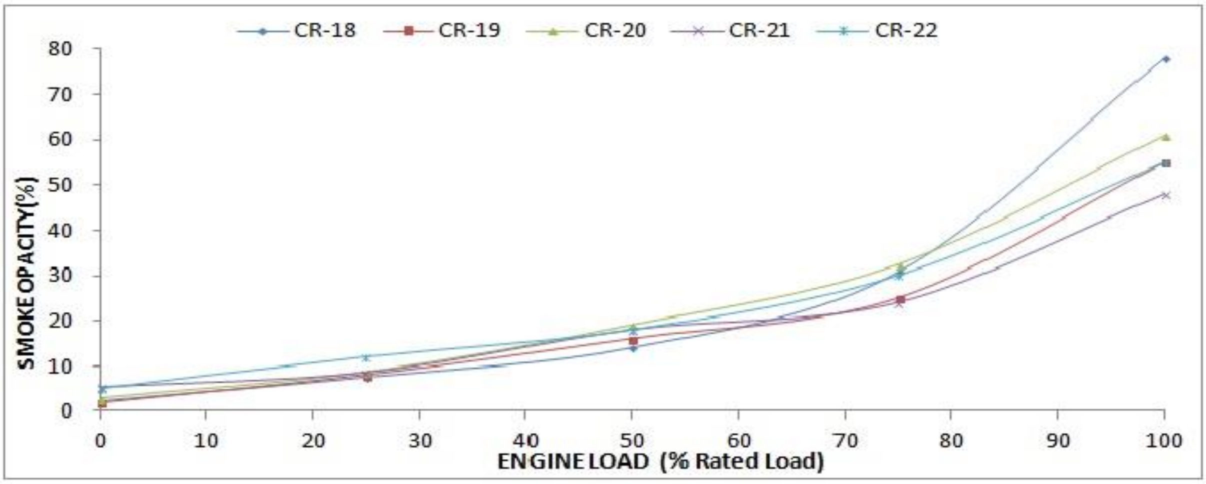


Figure 4.4 Variation of smoke density (opacity) with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel.

Carbon monoxide is an intermediate product, which is formed because of lower operating temperature, improper mixing of fuel, and incomplete combustion of fuel. If complete combustion of fuel takes place in the combustion chamber, then carbon monoxide gets converted to carbon dioxide. The variation of CO emissions with the load at various compression ratios is shown in Figure 4.5, which depicted that CO emission initially decreased with the increase in the percentage of load, but increased latter. CO emission was found to be minimum at the compression ratio of 21.

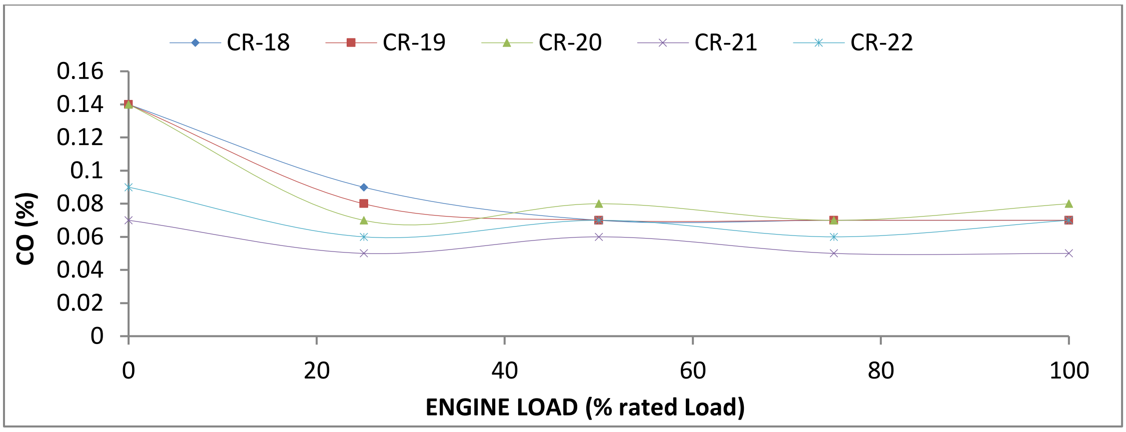


Figure 4.5 Variation of CO with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel

$CO_2$is formed due to complete combustion of fuel. The significant emissions of CO_2 in exhaust gases indicate the complete combustion of fuel. The variation of CO_2 emissions with the load at various compression ratios is shown in Figure 4.6. The carbon dioxide emissions increase with increased in the percentage of load. It was observed that at the compression ratio of 21, the carbon dioxide emissions were the highest in exhaust gases.

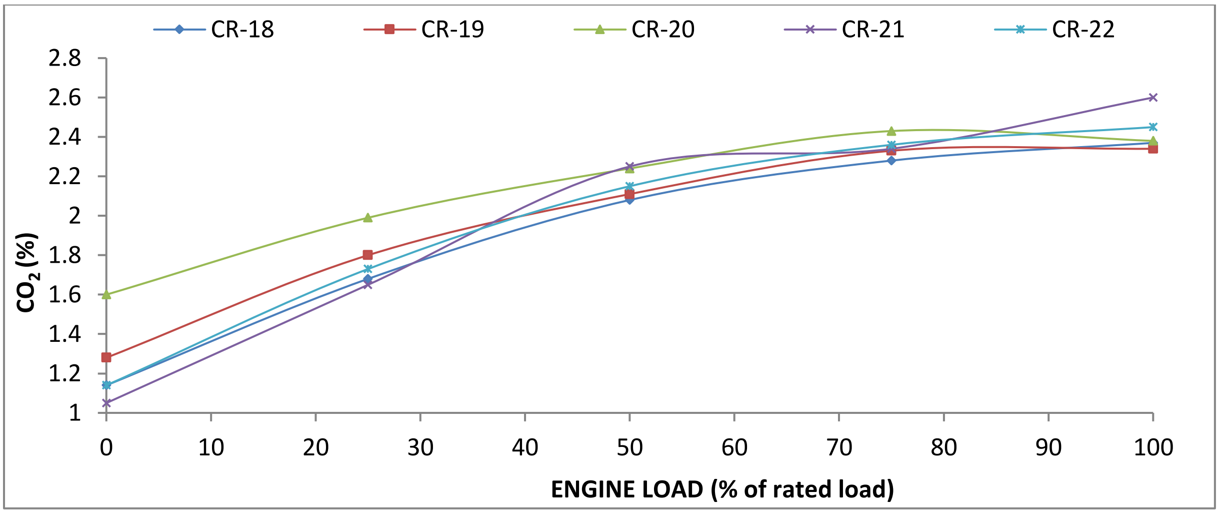


Figure 4.6 Variation of CO_2 with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel

HC emissions are the result of incomplete combustion. Diesel engines are always lean burned engine and involve excess air; therefore, these engines release less unburned HC as compared to the gasoline engines. The variation of HC emissions with the load at different compression ratios is shown in Figure 4.7. HC emissions initially decreased, but subsequently increased with increase in the percentage of load at constant compression ratio. The figure shows that

HC emissions were minimum at the compression ratio of 21 owing to complete combustion and better mixing of fuel and air.

NOx emissions are caused because of the peak cycle temperature and duration for which this temperature is maintained in the combustion chamber. The variation of NOx emissions at different compression ratios with the load is shown in Figure 4.8. NOx emissions increased with increase in the percentage of load, but decreased latter. Moreover, NOx emissions increased with increase in compression ratio, possibly owing to higher combustion duration at a higher compression ratio.

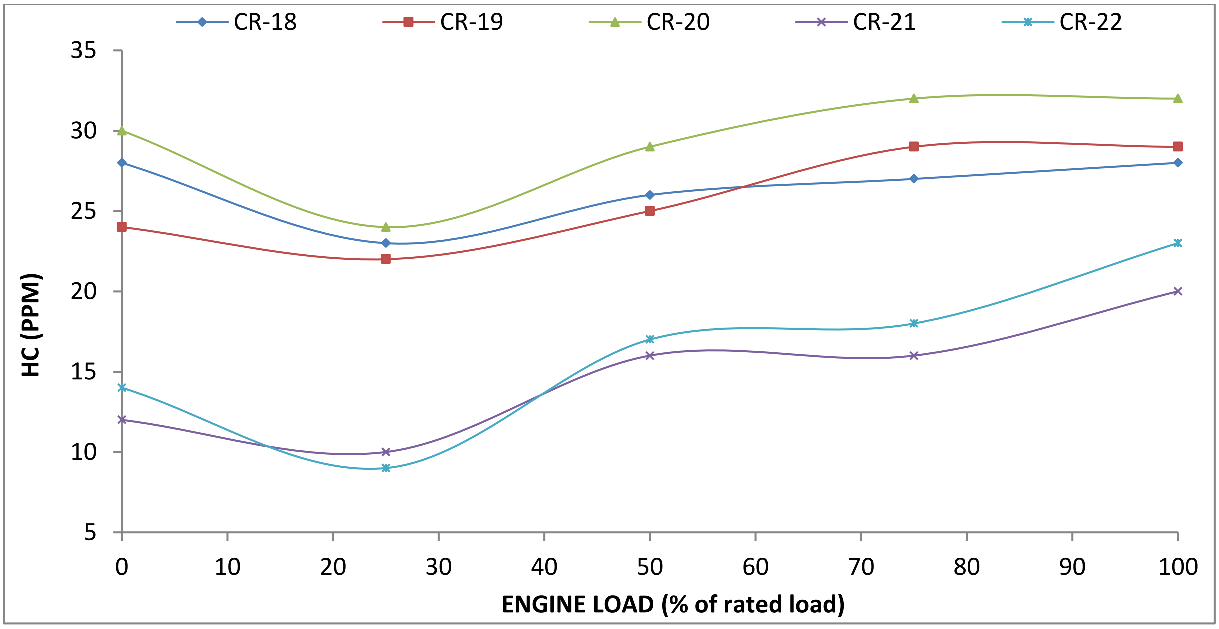


Figure 4.7 Variation of HC with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel

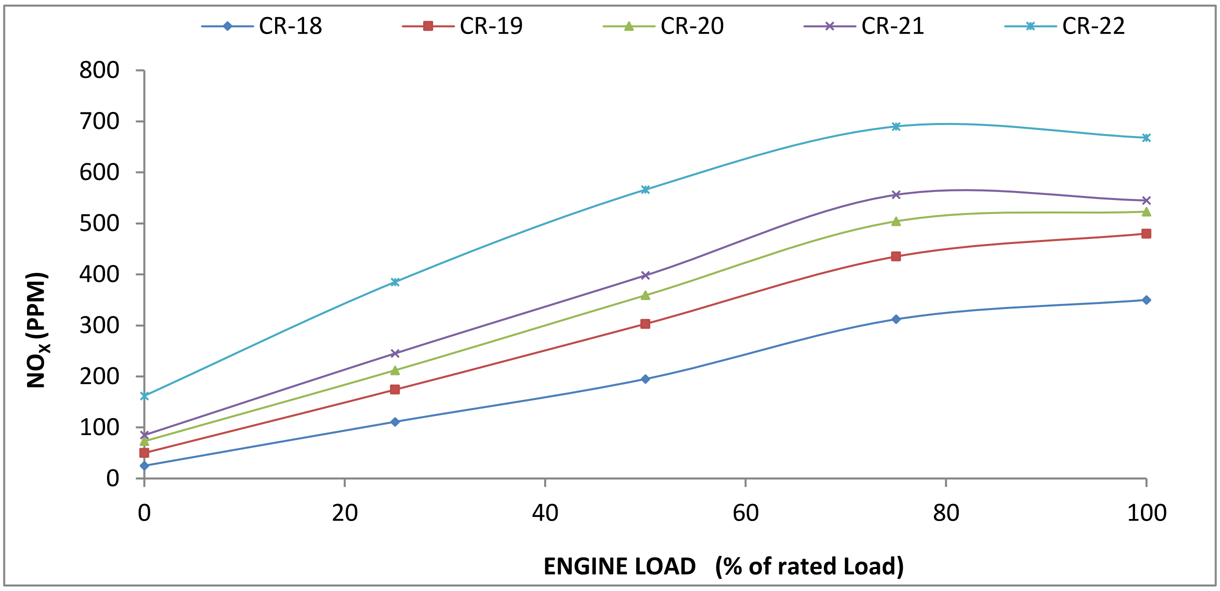


Figure 4.8 Variation of NOx with percentage increase in Load at different Compression Ratios upon fuelling the engine with diesel

4.2.1.3 Combustion Studies

The combustion in C.I. engines depends upon several factors such as the type of fuel, injection pressure, injection advance, compression ratio, intake air temperature, jacket water temperature, fuel temperature, supercharging, speed, engine size, type of combustion chamber, and load. The peak cycle pressure during combustion is the significant parameter, and its value depends upon the delay period. The shorter delay period leads to smooth pressure rise and minute increase in peak cycle pressure because under these conditions most of the fuel burns in the third phase of combustion (Controlled combustion). The variation of cylinder pressure with the engine crank angle is shown in Figure 4.9. The peak pressure at the compression ratio of 21 and 22 was found to be very close. The data related to pressure were taken at peak load (100% load) because peak load is the optimized load. The maximum rate of pressure rise for different compression ratio is shown in Figure 4.10 and the rate of pressure rise was the highest at the compression ratio of 21. Other combustion parameters at different compression ratios are summarized in Table 4.1.

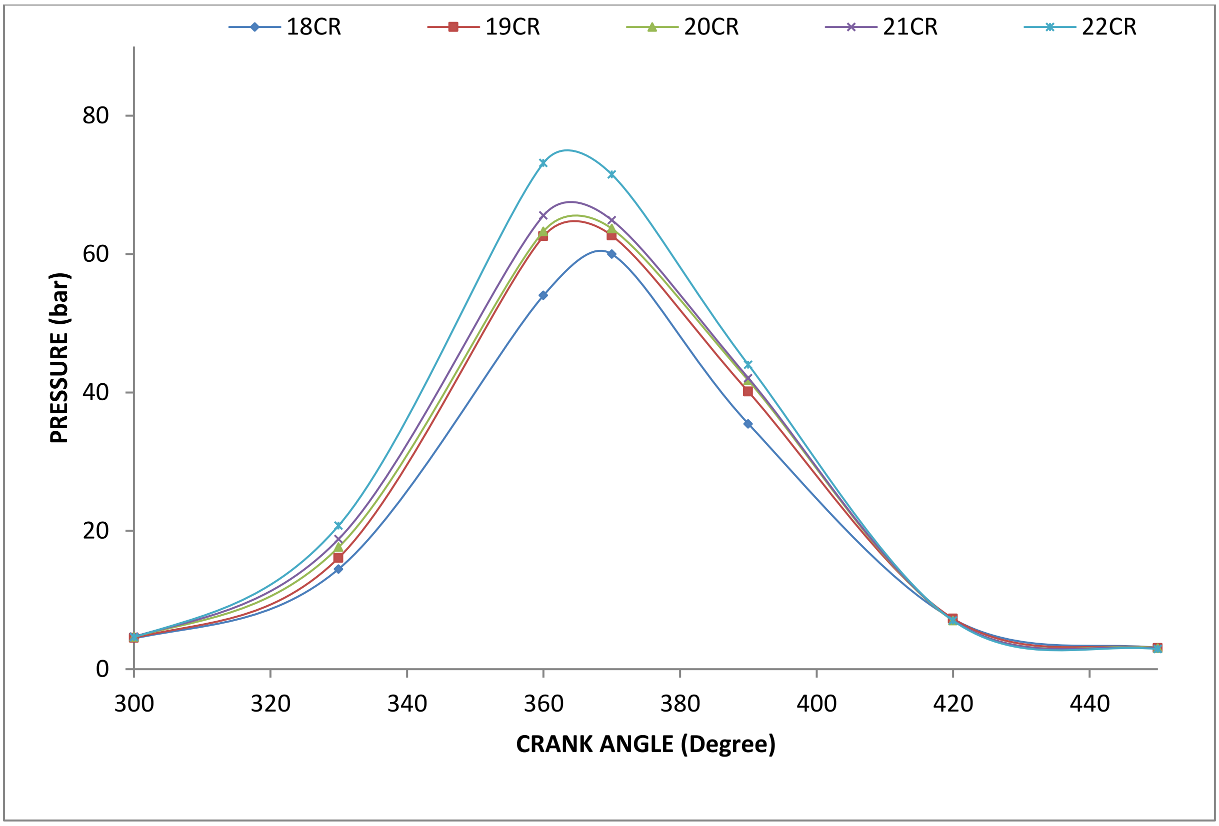


Figure 4.9 Variation of cylinder Pressure with crank angle at different Compression Ratios upon fuelling the engine with diesel

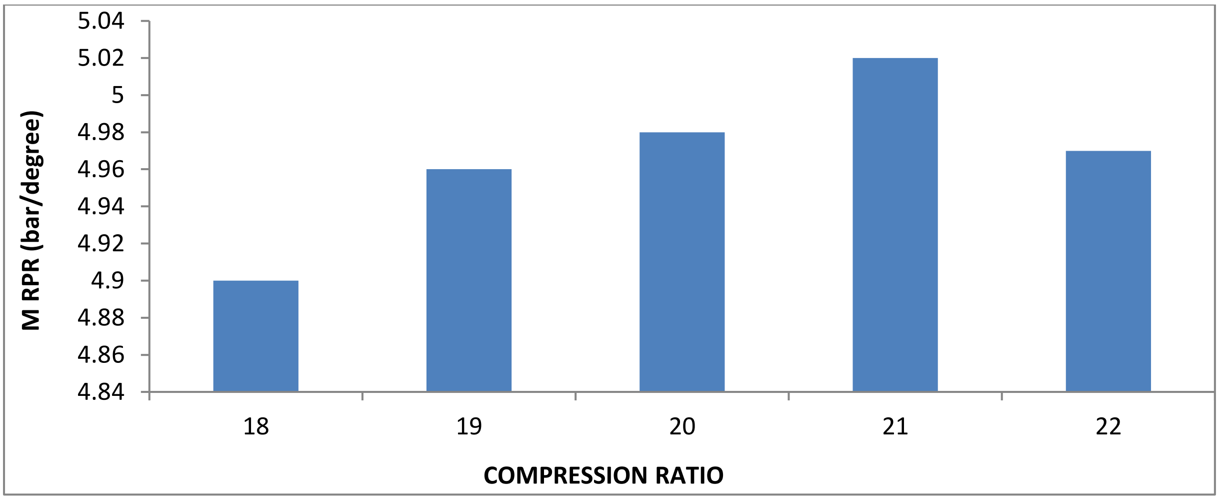


Figure 4.10 Variation of the maximum rate of pressure rise at different compression ratio

Table 4.1 Combustion parameters at different Compression Ratios when engine operated with diesel

Combustion Parameters	18CR	19CR	20CR	21CR	22CR
Maximum Rate of Pressure Rise (bar/°CA)	5.85	5.9	6.1	6.4	6.2

Maximum Net Heat Release MJ	43.2	43.6	44	46.4	45.3
Ignition Delay (°CA BTDC)	14.8	14.6	14.5	13.9	14.1
Start of Combustion (°CA)	11.3	10.25	9.5	9	9.1
Maximum Pressure (bar)	62.4	65.83	69.5	71.56	76.11
Angle for Maximum Pressure (°CA ATDC)	7	7.5	8	8.5	9

4.2.2 Optimization for Injector Needle Lift Pressure (Injection Pressure)

The test run was conducted at the injection pressures of 190 bar, 203 bar, and 210 bar, while keeping constant the compression ratio of 21, the injection timing of 23°CA BTDC, and the engine speed of 1500 RPM.

4.2.2.1 Performance Studies

The variation of BTE with the load at different injection pressures is shown in Figure 4.11. The graph shows that the BTE increases with an increase in the percentage of load. BTE was found to be maximum at the injection pressure of 203 bar, possibly owing to better atomization, vaporization, mixing of fuel, and better combustion quality. With lower percentage of load, the BTE for each injection pressure was very close.

BSFC is an important parameter for measuring the performance of the engine. The variation of BSFC with the load at different injection pressures is shown in Figure 4.12. BSFC decreased with increase in the percentage of load and the same trend was found for all the injection pressures. Minimum BSFC was found for the injection pressure of 203 bar.

The variation of exhaust gas temperature with load at different injection pressures is shown in Figure 4.13. Exhaust gas temperature increased with increase in the percentage of load and was found to be minimum for the injection pressure of 203 bar, possibly owing to proper combustion and better utilization of fuel.

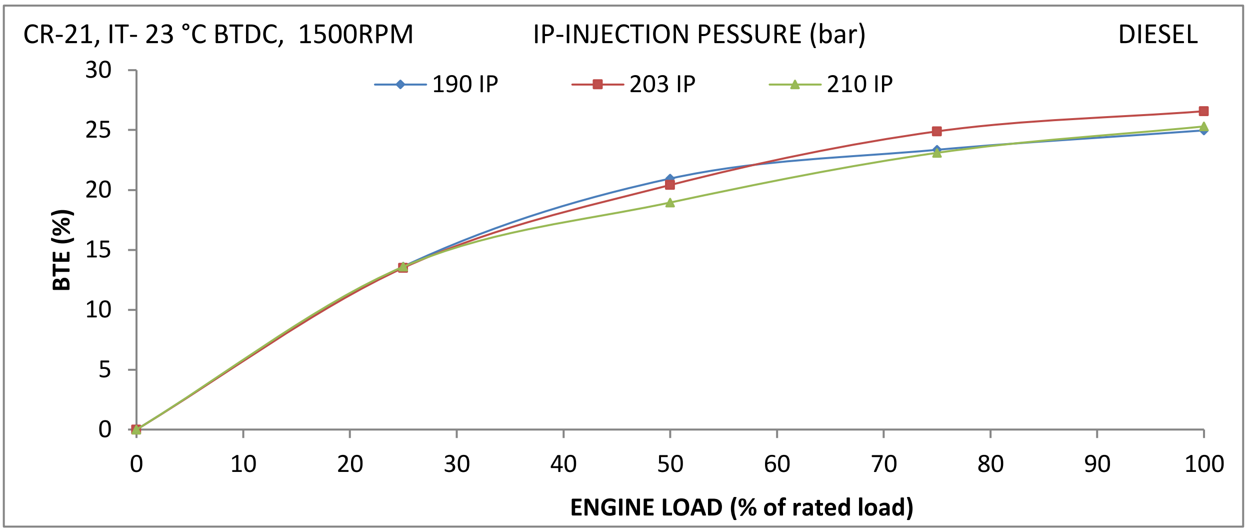


Figure 4.11 Variation of brake thermal efficiency with percentage increase in load at different injection pressures upon fuelling engine with diesel

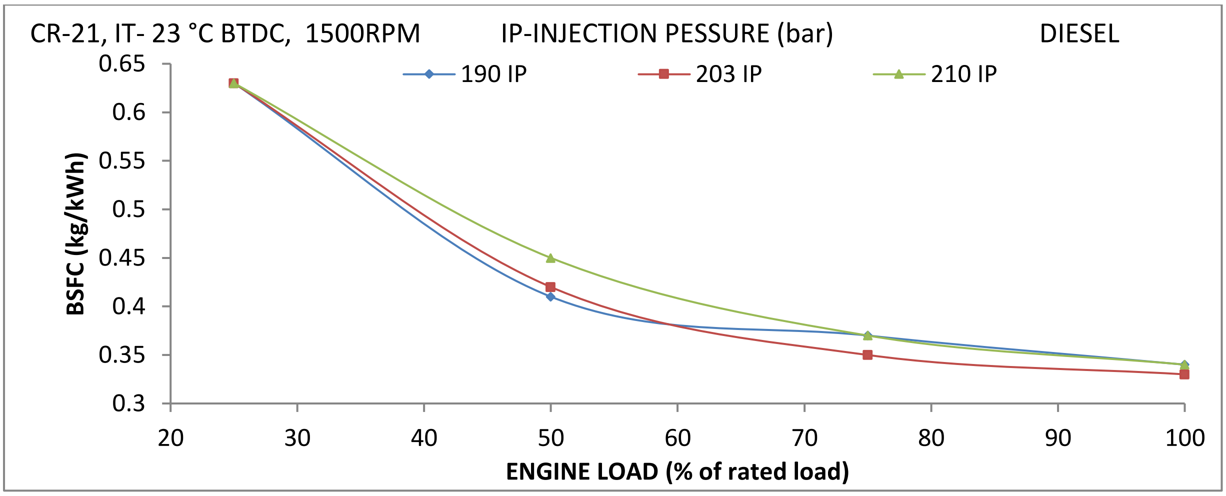


Figure 4.12 Variation of brake specific fuel consumption with percentage increase in load at different injection pressures

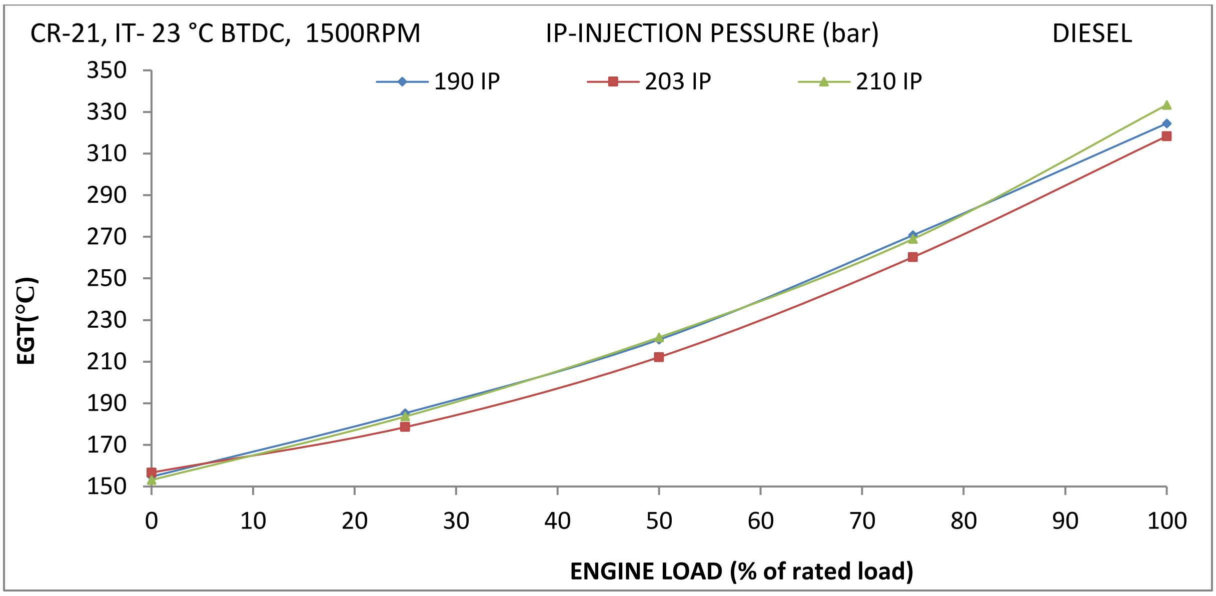


Figure 4.13 Variation of Exhaust Gas Temperature with percentage increase in load at different injection pressures upon fuelling the engine with diesel

4.2.2.2 Emission Studies

The variation of emission parameters with load at different injection pressures is presented in this part of the analysis. The parameters of smoke opacity, CO, CO_2, HC, and NOx emissions were considered for emission analysis.

The variation of smoke opacity with the load at different injection pressures is shown in Figure 4.14. Smoke opacity is found to increase with increase in the percentage of load possibly owing to increase in fuel air ratio. Lowest smoke opacity was observed for injection pressure of 203 bar because of superior combustion, atomization, and mixing of fuel.

The variation of CO emissions with the load at different injection pressures is shown in Figure 4.15. CO emissions are found to decrease initially with increase in the percentage of load, but it subsequently increases. The lowest CO emission was observed for injection pressure of 203 bar owing to better combustion.

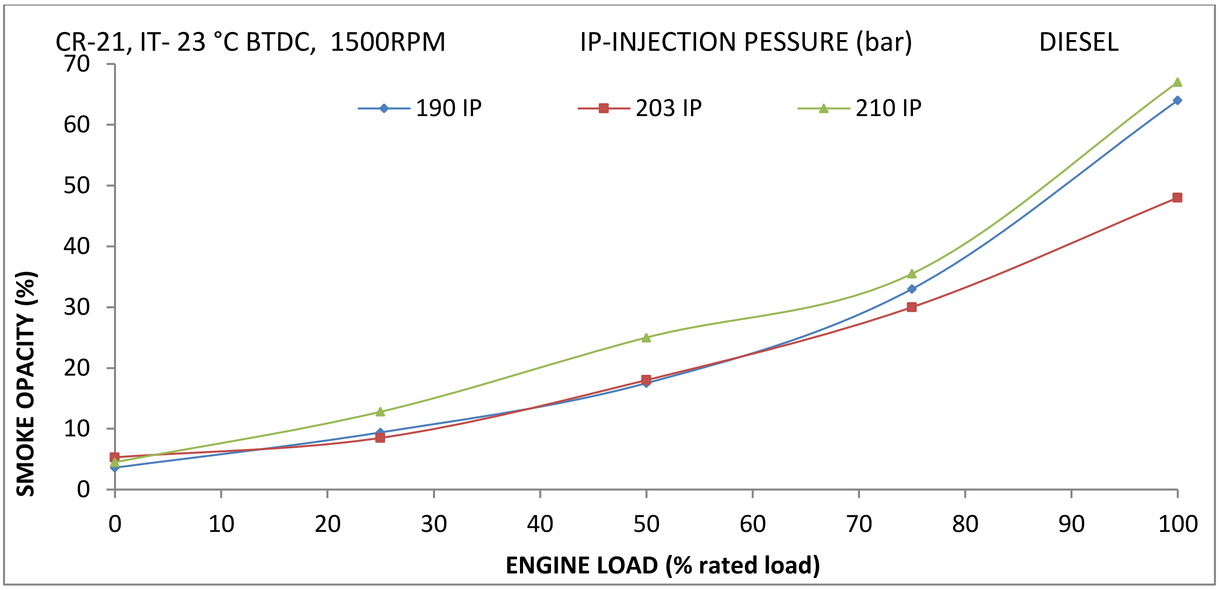


Figure 4.14 Variation of smoke opacity emissions with percentage increase in the load at different injection pressures upon fuelling the engine with diesel

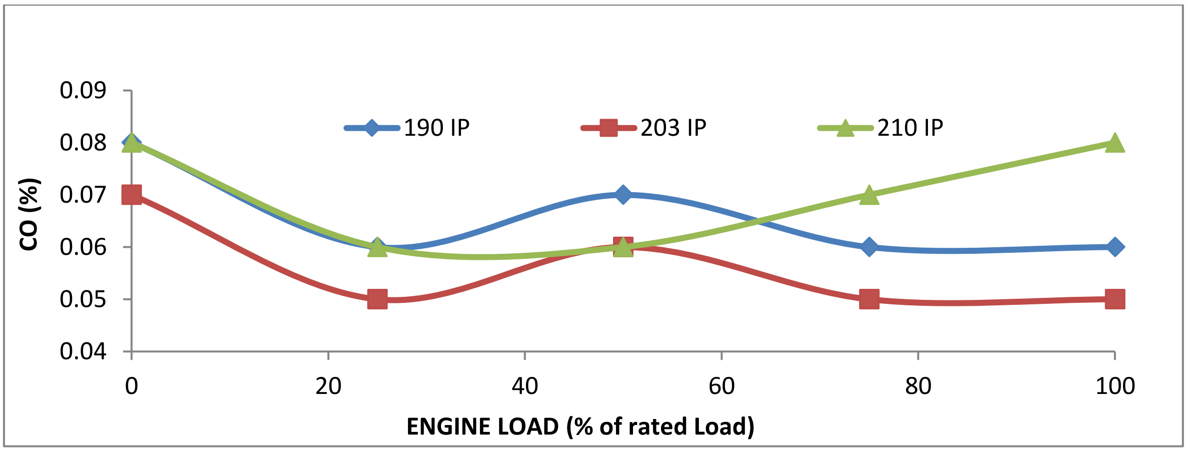


Figure 4.15 Variation of CO emissions with percentage increase in load at different injection pressures upon fuelling the engine with diesel

The variation of CO_2 emissions with load at different injection pressures is shown in Figure 4.16. The graph shows that CO_2 emissions increase with the increase in the percentage of load. The similar trend followed in the plot for all the injection pressures. The highest CO_2 emission was observed for injection pressure of 203 bar owing to better and complete combustion.

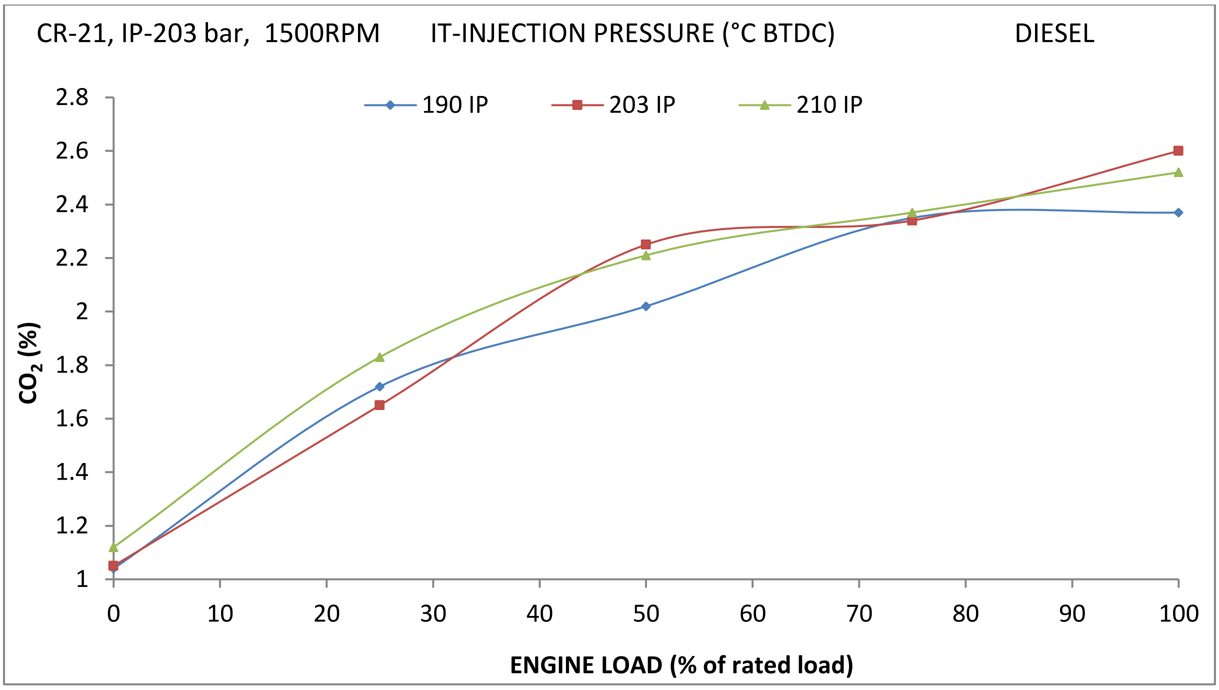


Figure 4.16 Variation of CO_2 emissions with % load at different injection pressures when engine fuelled with diesel

HC emissions are the result of incomplete combustion of the fuel. The variation of HC emissions with the load at different injection pressures is shown in Figure 4.17. The plot shows that initially HC emissions slightly decrease and then gradually increase with increase in the percentage of load. HC emissions are minimum for injection pressure of 203 bar owing to complete combustion.

The variation of NOx emissions with the load at different injection pressures is shown in Figure 4.18. NOx emissions are found to increase with the increase in the percentage of load, but these emissions tend to decrease if the percentage of load becomes higher. A similar pattern followed for all the injection pressures. NOx emissions were minimum for injection pressure of 203 bar at full load because of less combustion temperature and duration.

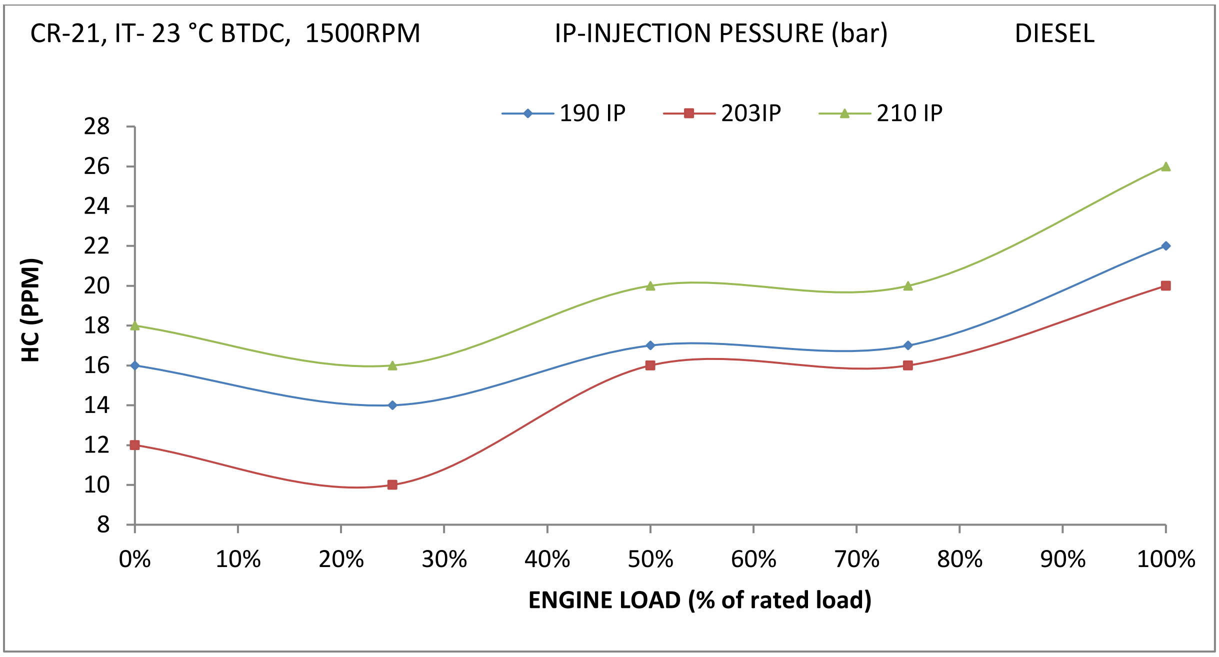


Figure 4.17 Variation of hydrocarbons with percentage increase in load at different injection pressures upon fuelling the engine with diesel

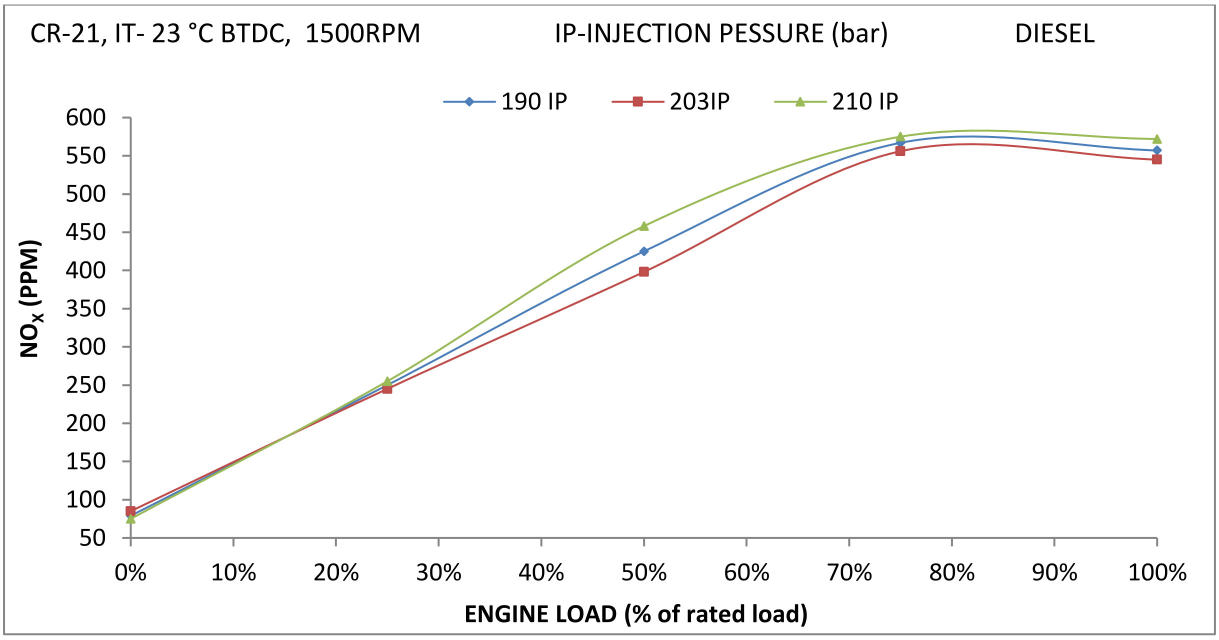


Figure 4.18 Variation of NO_X with percentage increase in load at different injection pressures upon fuelling the engine with diesel

4.2.2.3 Combustion Studies

The effect of injection pressure on combustion parameters like cylinder pressure, maximum rate of pressure rise, heat release rate, and other combustion parameters was observed. Cylinder pressure depends on the ability of the fuel to mix with air and its burning. Peak pressure and the rate of pressure rise are dependent on the amount of fuel burn in the third stage of combustion. The variation of cylinder pressure with crank angle at different injection pressures is shown in Figure 4.19. The peak cylinder pressure increases with increase in injection pressure. The Figure shows that the crank angle for peak pressure shifted to left with the increase in injection pressure, but the maximum rate of pressure rise was found to be the highest (6.4 bar) at an injection pressure of 203 bar as shown in Figure 4.20. Moreover, the net heat release rate was the highest for injection pressure of 203 bar. The shorter delay period (13.9°CA) was observed for injection pressure of 203 bar as compared to the injection pressure of 190 (14.5°CA) and 210 bars (14.1°CA) as per Table 4.2.

Table: 4.2 Combustion parameters at different injection pressures when engine fuelled with diesel

Combustion Parameters	IP-190 bar	IP-203 bar	IP-210 bar
Maximum Rate of Pressure Rise (bar/°CA)	6.1	6.4	6.3
Maximum Net Heat Release (MJ)	45.2	46.4	45.6
Ignition Delay (°CA BTDC)	14.5	13.9	14.1
Start of Combustion (°CA)	4.87	9	14.3
Maximum Pressure (bar)	70.2	71.56	73.2
Angle for Maximum Pressure (°CA ATDC)	9	8.5	8

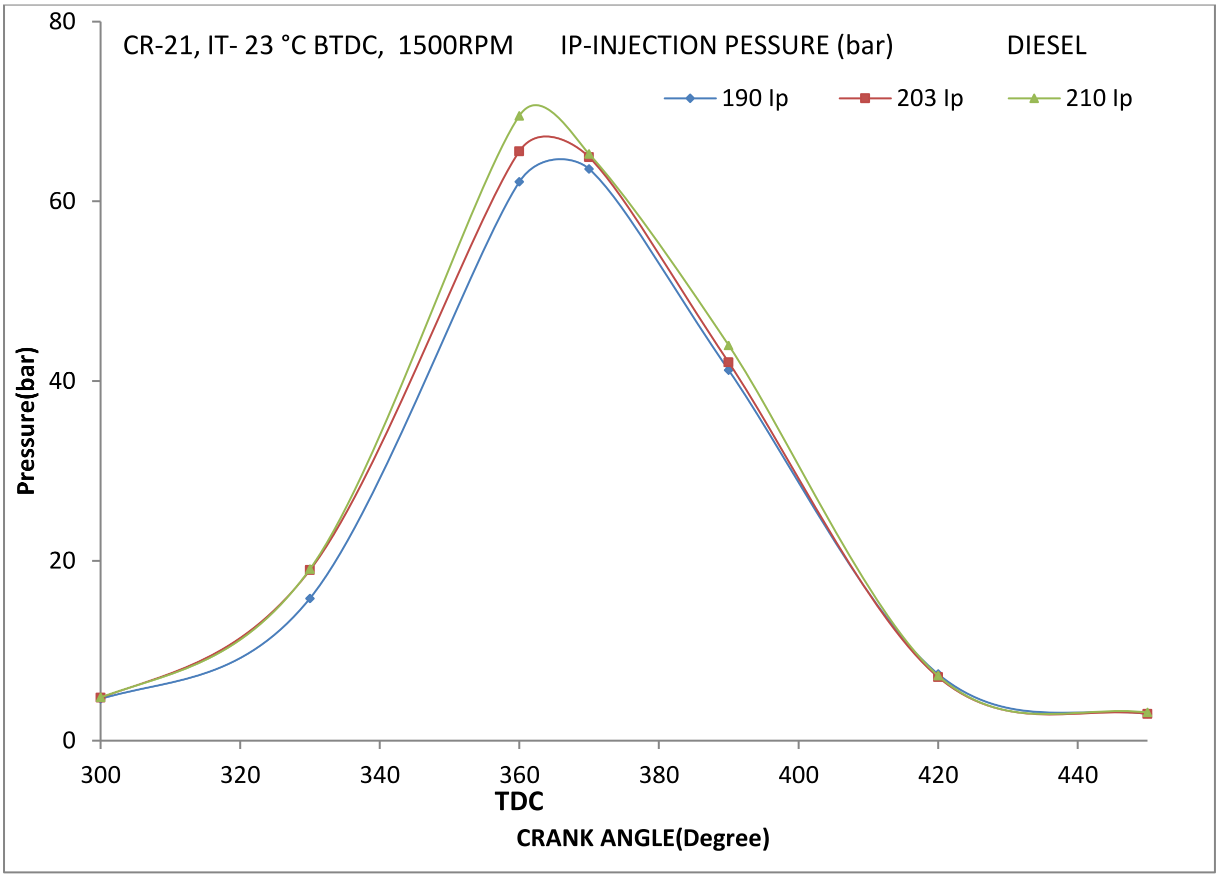


Figure 4.19 Variation of cylinder pressure with crank angle at different injection pressures upon fuelling the engine with diesel

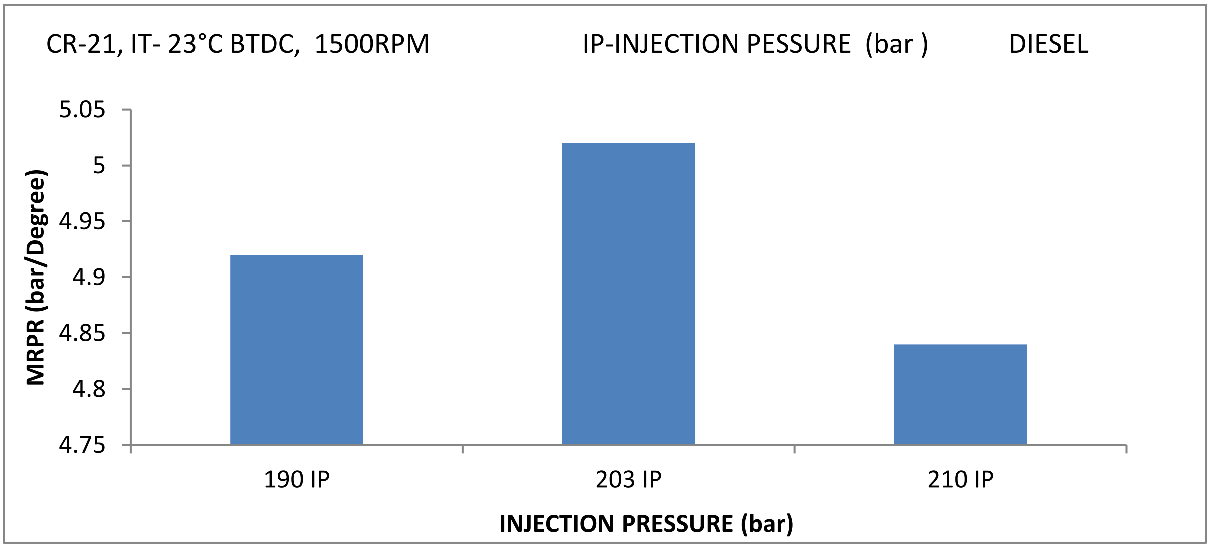


Figure 4.20 Variation of the maximum rate of pressure rises at different injection pressures upon fuelling the engine with diesel

4.2.3 Optimization for Injection Timing

The experimental study was conducted to observe the effect of injection timing on performance, combustion, and emission characteristics. The engine operated for injection timing of 20, 23, and 27°CA BTDC while keeping the other engine parameters constant.

4.2.3.1 Performance Studies

The variation of BTE with the load at different injection timings is shown in Figure 4.21. The plot shows that BTE increases with increase in the percentage of load. BTE was found to be maximum at 23°CA BTDC injection timing owing to proper atomization and mixing of air and fuel. BTE of 20°CA and 27°CA BTDC for all load conditions were inferior to 23°CA BTDC.

The variation of BSFC at different injection timings with the load is shown in Figure 4.22. BSFC decreases with increase in the percentage of load, and BSFC was found to be minimum for 23°CA BTDC at full load condition. BSFC at 20°CA BTDC was very close to 23°CA BTDC. The highest BSFC was achieved at 27°CA BTDC throughout the load range, which is probably due to longer ignition delay and combustion duration.

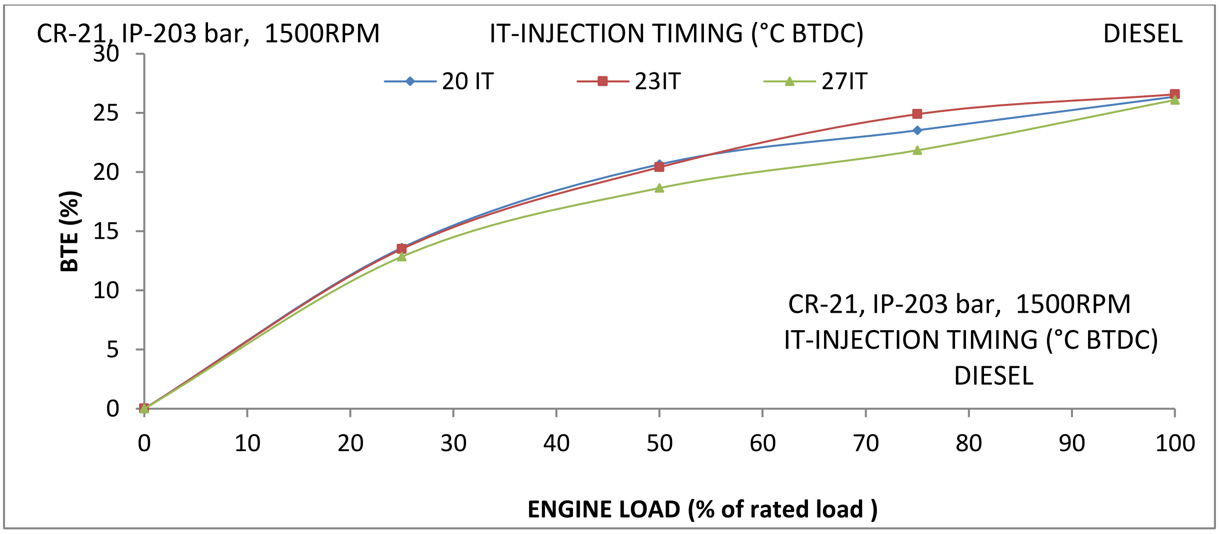


Figure: 4.21 Variation of brake thermal efficiency with percentage increase in load at different injection timings upon fuelling the engine with diesel

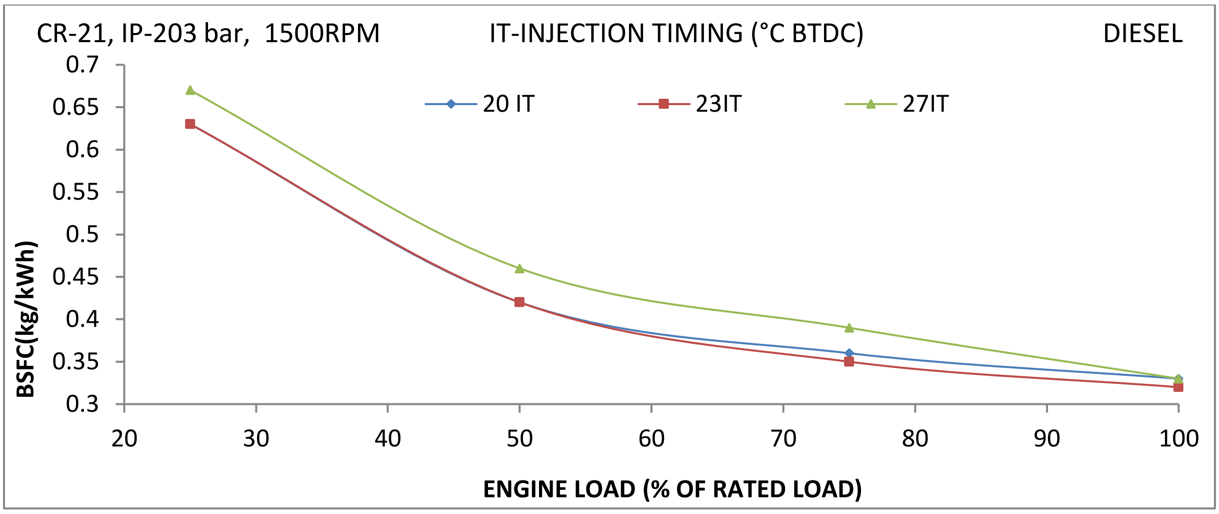


Figure: 4.22 Variation of brake specific fuel consumption with percentage increase in load at different injection timings upon fuelling the engine with diesel

The variation of exhaust gas temperature with the load at different injection timings is shown in Figure 4.23. Exhaust gas temperature increases with increase in the percentage of load, and it was found minimum at 23°CA BTDC. Exhaust gas temperature at 20°CA and 27 °CA BTDC were higher than 23°CA BTDC possibly owing to higher ignition delay and combustion duration at these injection timings.

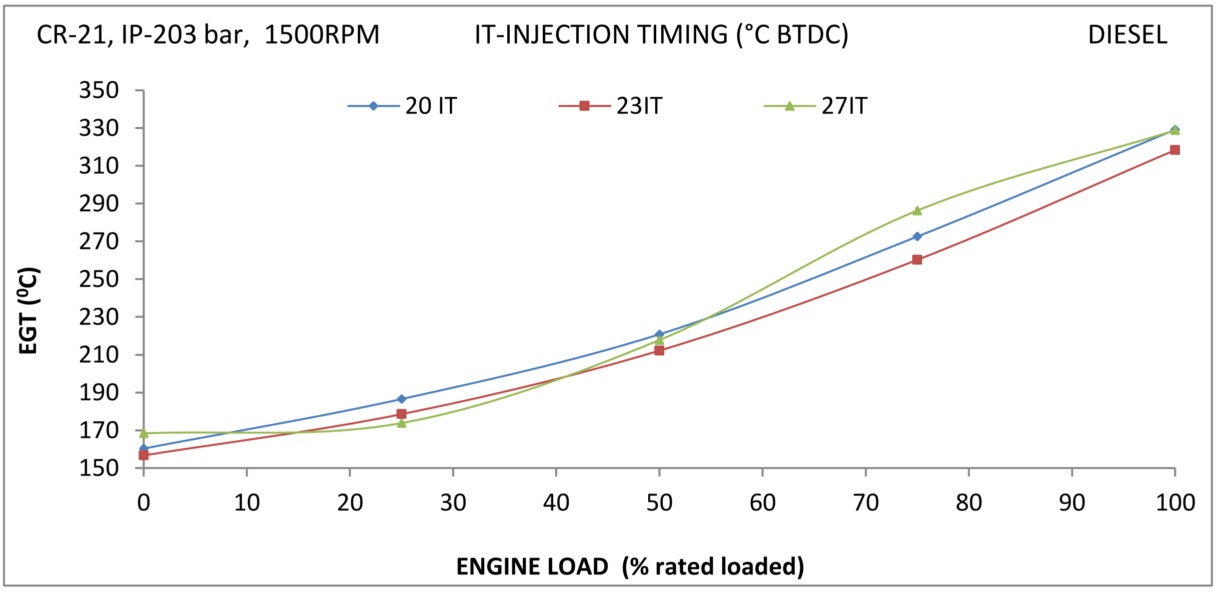


Figure: 4.23 Variation of EGT with percentage increase in load at different injection timings upon fuelling the engine with diesel

4.2.3.2 Emission Studies

The variation of smoke opacity with the load at different injection timings is shown in Figure 4.24. Smoke opacity increases with increases in the percentage of load at different injection timings. The smoke opacity was lowest for injection timing of 23°CA BTDC because of better intermixing of fuel and combustion quality compared to of 20°CA and 27°CA BTDC at higher load.

CO emissions were found in the exhaust gases because of incomplete combustion and variation of CO emissions with load at different injection timings is shown in Figure 4.25. The figure shows that CO emissions decrease initially and then gradually increase. CO emission was found to be lowest at the injection timing of 23°CA BTDC owing to complete combustion. The variation of CO_2 emissions with the load at different injection timings is shown in Figure 4.26. CO_2 emission was found to be maximum with higher load condition at 23°CA BTDC because of complete combustion and efficient burning of fuel at this injection timing.

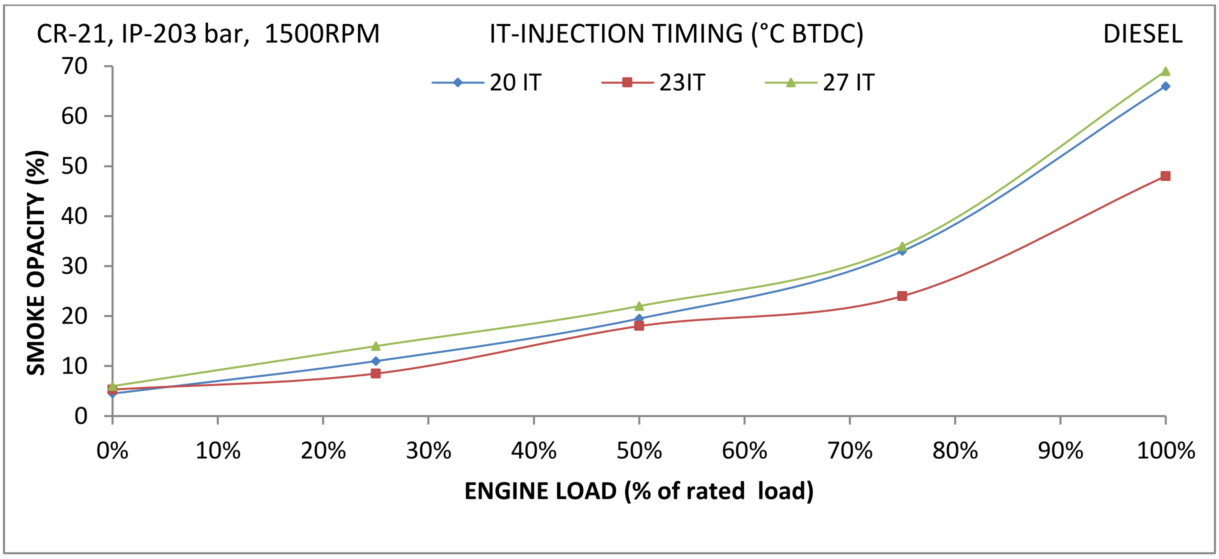


Figure: 4.24. Variation of Smoke opacity with percentage increase in load at different injection timings upon fuelling the engine with diesel

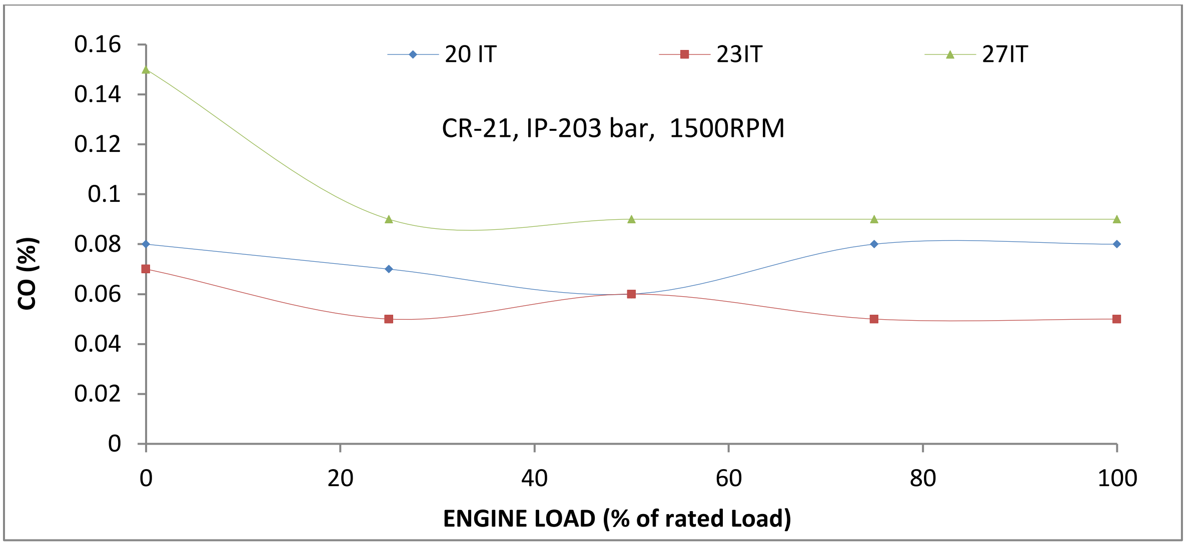


Figure: 4.25 Variation of CO with percentage increase in load at different injection timings upon fuelling the engine with diesel

The variation of HC emissions with the load at different injection timing is shown in Figure 4.27. HC emissions increase with increase in the percentage of load at different injection timing as shown in the figure. HC emissions were found minimum at 23°CA BTDC because of better combustion quality at this injection timing. HC emissions for 20°CA and 27°CA BTDC were marginally higher than that of 23°CA BTDC owing to lower and incomplete combustion quality.

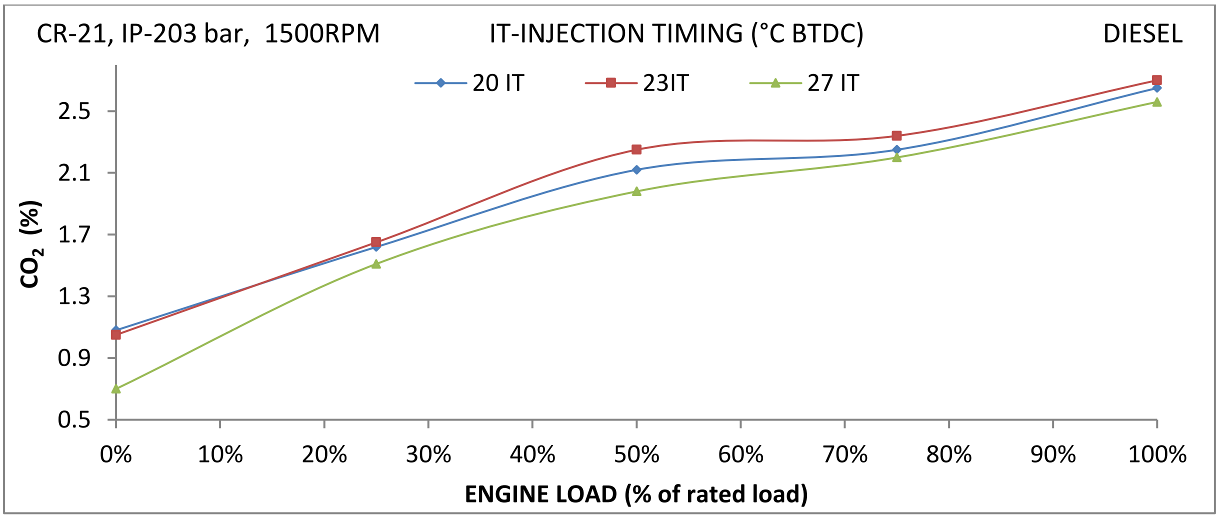


Figure: 4.26 Variation of CO_2 with percentage increase in load at different injection timings upon fuelling the engine with diesel

The variation of NOx emissions with the load for different injection timings is shown in Figure 4.28. It is evident from the figure that NOx emissions increase with the increase in the percentage of load. These emissions were found in exhaust gases because of large combustion temperature and combustion duration of fuel in the combustion chamber. NOx emissions are lowest at 23°CA BTDC owing to less combustion duration and exhaust gas temperature.

From the above analysis, it can be concluded that the results improved for 23°CA BTDC injection timing at full load condition; hence, this is the optimum injection timing for the engine operated with diesel.

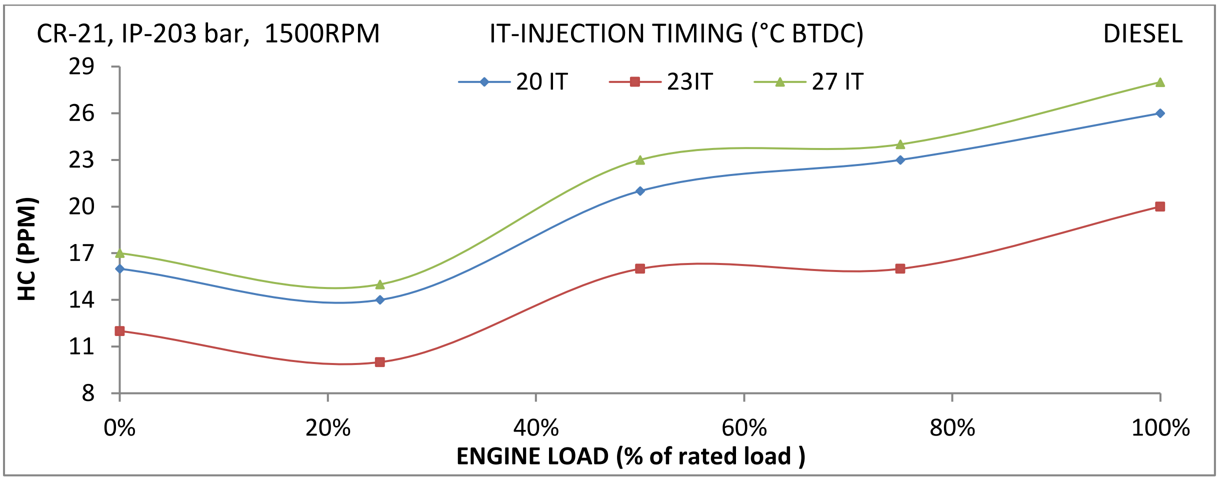


Figure: 4.27 Variation of Hydrocarbon with percentage increase in load at different injection timings upon fuelling the engine with diesel

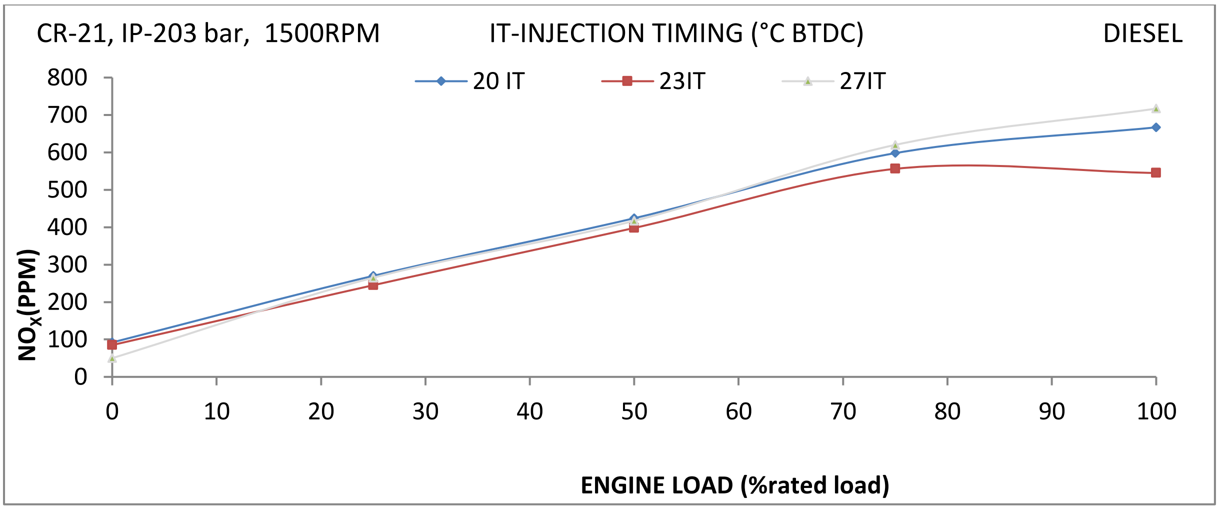


Figure: 4.28 Variation of NOx with percentage increase in load at different injection timings upon fuelling the engine with diesel

4.2.3.3 Combustion Studies

The combustion characteristics of the VCR engine were analyzed for the different injection timings. The variation of cylinder pressure with crank-angle for different injection timings is shown in Figure 4.29. It is observed from the graph that the peak cylinder pressure increases with the increase in injection pressure, and the angle of peak cylinder pressure decreases with the rise of injection pressure. Peak pressure at injection timings of 20°CA, 23°CA, and 27°CA BTDC were 67.4 bar, 71.56 bar, and 73.9 bar, respectively. The maximum rate of pressure rise for different injection timing is shown in Figure 4.30. It is observed from the plot that the maximum rate of pressure rise and net heat release rate were highest at 23°CA BTDC. The ignition delay is shorter for injection pressure of 23°CA BTDC. Other combustion characteristics are summarized in Table 4.3 at different injection timings.

Table: 4.3 Combustion parameters at different injection timings when engine operated with diesel

Combustion Parameters	IT-20 (˚CA BTDC)	23 IT (˚CA BTDC)	27 IT (˚CA BTDC)
Maximum Rate of Pressure Rise (bar/˚CA)	6.3	6.4	6.2
Maximum Net Heat Release (MJ)	44.9	46.4	45.8
Ignition Delay (˚CA BTDC)	14	13.9	14.4
Start of Combustion (˚CA)	6.95	9	11.23
Maximum Pressure (bar)	67.4	71.56	73.9
Angle for Maximum Pressure (˚CA ATDC)	9.5	8.5	8

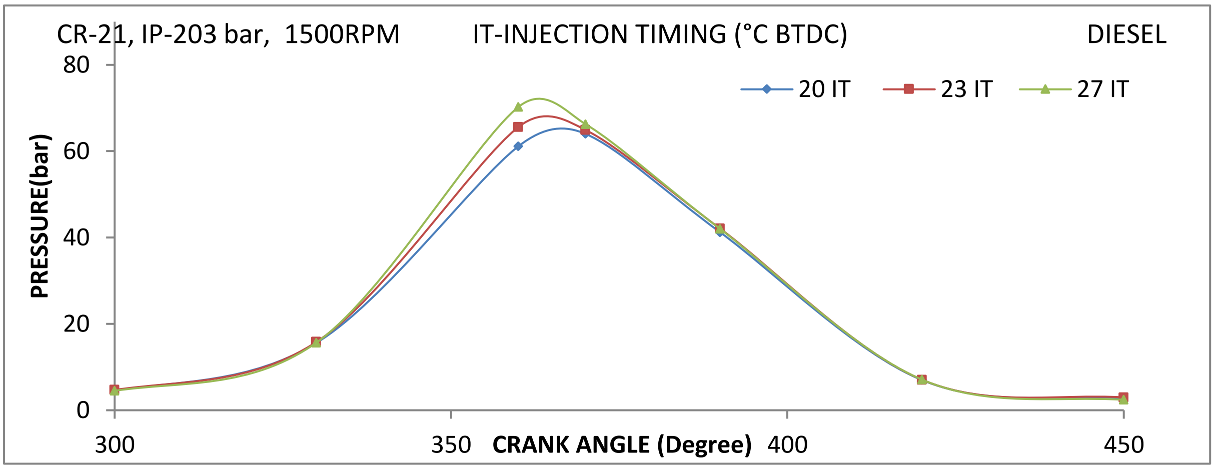


Figure 4.29 Variation of cylinder pressure with crank angle at different injection timings upon fuelling the engine with diesel

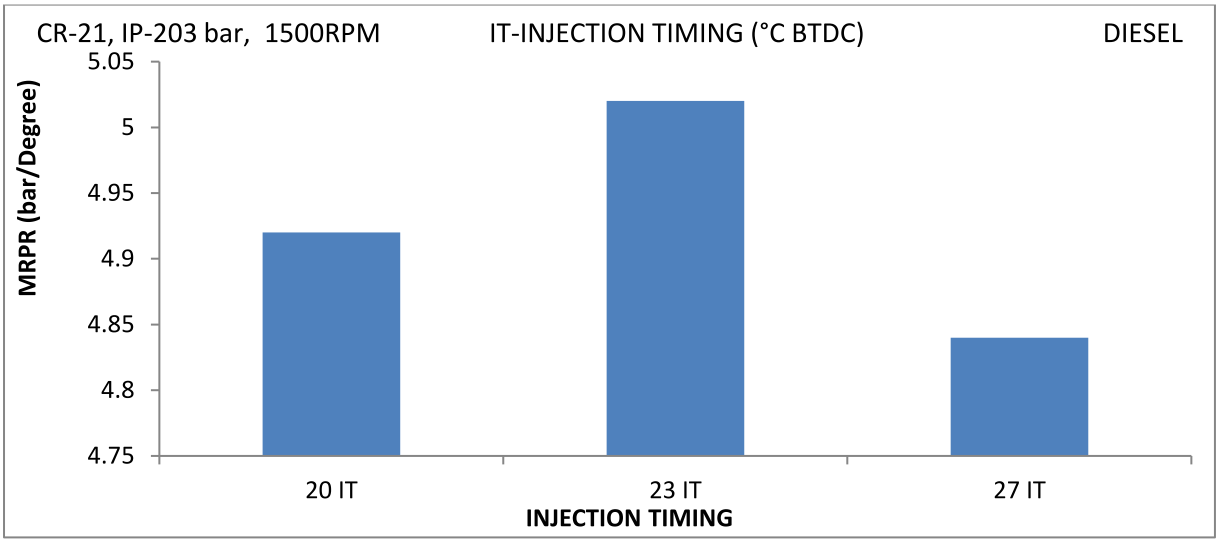


Figure: 4.30 Variation of the maximum rate of pressure rise with Crank Angle at different injection timings upon fuelling the engine with diesel

4.2.3.4 Summary of Results

In the optimization of compression ratio process, BTE was maximum at the compression ratio of 21. Hence, it was considered as an optimum compression ratio for multi-fuel engine with VCR. The compression ratio of 22 and 20 exhibited marginally lower BTE compared to the optimum compression ratio. Similarly, an injection pressure of 203 bar and the injection timing of 23°CA BTDC exhibited maximum BTE through use of diesel as fuel.

The engine parameters compression ratio of 21, an injection pressure of 203 bar, and injection timings of 23°CA BTDC were considered as optimized parameters in the results of the experimentation engine fuelled with diesel. Optimized Engine parameters are summarized in Table 4.4.

Table: 4.4 Optimized Engine parameters when engine operated with diesel

S. No.	Engine parameter	Optimum value
1	Compression Ratio	21
2	Injector Pressure (bars)	203
3	Injection Timing (°CA BTDC)	23
4	Load	100%

4.3 OPTIMIZATION OF PREHEATED THUMBA OIL BLEND WITH DIESEL

The use of diesel as fuel was evaluated for multi-fuel engine with VCR for the base case data. Similarly, Thumba oil and its blends with diesel were also assessed for examining the optimized Thumba oil blends with diesel for the engine. All the combinations were preheated using engine waste exhaust gases in the developed preheating arrangement before entering into the engine fuel pump. After the primary study, T40, T60, T70, T80, and T90 Thumba oil-diesel blends were not considered probably owing to marginally inferior results. Though the results of fuelling pure Thumba oil were not comparable, it was compulsory to observe the engine characteristics for using pure Thumba oil, so that it can be compared with other blends. Various engine input parameters and their range to be used for finding the optimized Thumba oil-diesel blend is given in Table 4.5.

Table 4.5 various engine input parameters used to optimize the Thumba oil blend

Sr. No.	Parameters	Unit	Range Values
1	Thumba oil blends	-	T10, B 20, T30, T50, and T100
2	Load Range	% of rated load	0, 25, 50, 75 and 100
Constant Parameters			
3	Engine Speed	RPM	1500
4	Compression Ratio	-	22
5	Injection Pressure	bar	203
6	Injection Timing	°CA BTDC	23

4.3.1 Performance Studies

The variation of the brake thermal efficiencies with the load for various Thumba oil-diesel blends is shown in Figure 4.31. BTE was found to be maximum for the preheated T20 Thumba oil-diesel blend because of better combustion quality and higher lubricity of Thumba oil.

The variation of BSFC with the load for different preheated Thumba oil-diesel blends is presented in Figure 4.32. The figure shows that the BSFC decreases with increase in the percentage of load, and BSFC was found to be minimum for T20 preheated Thumba oil-diesel blend. BSFC further increased with increase in the concentration of Thumba oil in the Thumba oil-diesel blend owing to poor combustion and atomization. BSFC for T30 preheated Thumba oil diesel blend was very close to T20 preheated Thumba oil diesel blend.

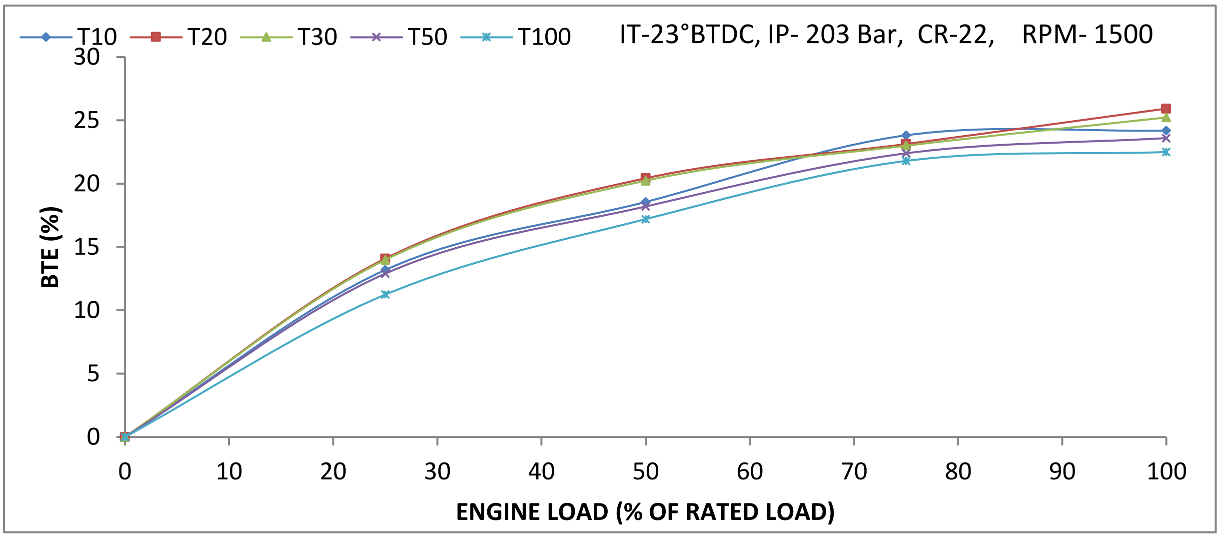


Figure 4.31Variation of brake thermal efficiency with percentage increase in load for various preheated Thumba oil-diesel blends

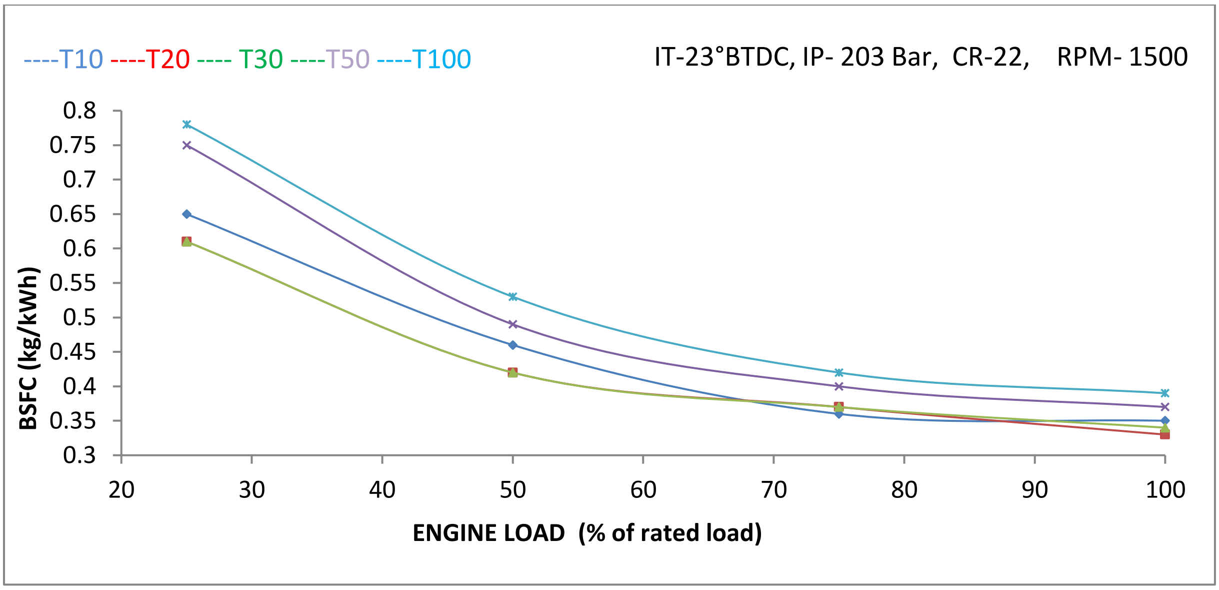


Figure 4.32 Variation of brake specific fuel consumption with percentage increase in load for various preheated Thumba oil-diesel blends

The variation of exhaust gas temperature with the load for various preheated Thumba oil diesel-blends is shown in Figure 4.33. The exhaust gas temperature was minimum for T20 preheated Thumba oil-diesel blend because it is the optimized blend. Exhaust gas temperature for T20 preheated Thumba oil blend was very close to T10 and T30 Thumba oil blend. Exhaust gas temperature increased with the increase in load as shown in the figure.

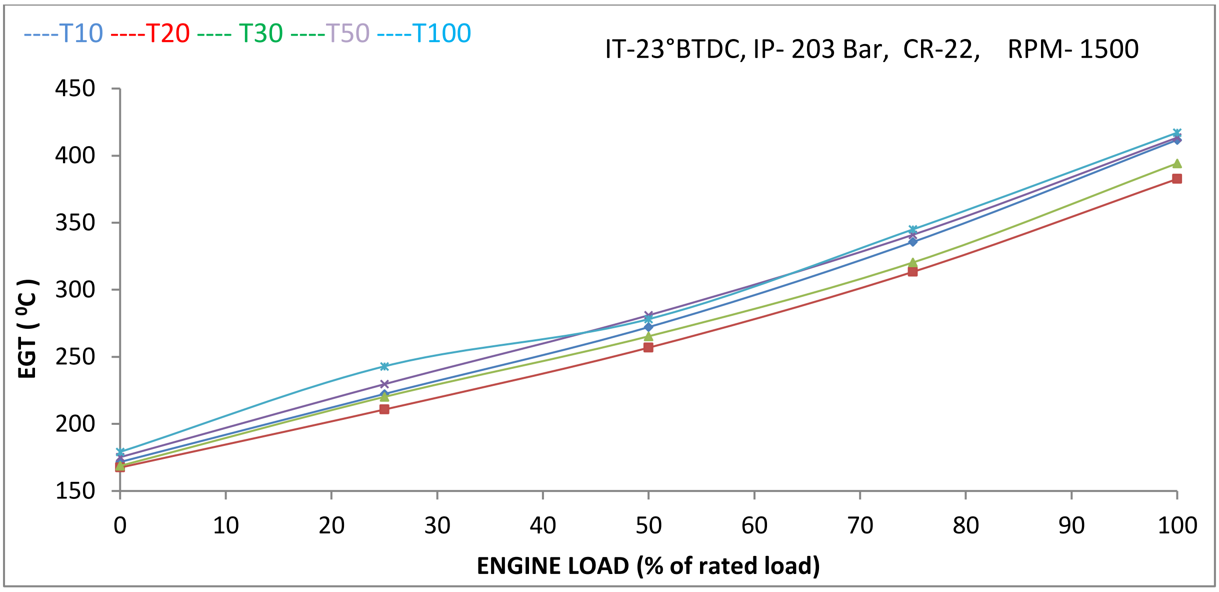


Figure 4.33Variation of exhaust gas temperature with percentage increase in load for various preheated Thumba oil-diesel blends

4.3.2 Emission Studies

The presence of smoke opacity in the exhaust gases depends upon the combustion quality of fuel. The variation of smoke opacity with the load for different preheated Thumba oil-diesel blends is shown in Figure 4.34. Smoke density increased with increase in load owing to the rich fuel-air ratio at higher percentage of load. Smoke opacity further increased with increase in the concentration of Thumba oil in the blend because of the poor atomization, higher viscosity, and more significant size of fuel molecules owing to the specific physical and chemical properties of Thumba oil. The smoke level was found to be minimum for 20% preheated Thumba oil-diesel blend, followed by 30% and 10% concentration of Thumba oil in the mixture due to better combustion in comparison to other combinations.

The formation of CO depends upon the combustion quality, fuel viscosity, vaporization, and atomization. The variation of CO emissions with the load for various Thumba oil blends is shown in Figure 4.35. CO emissions was minimum for T20 preheated Thumba oil blend in diesel, although the higher concentration of Thumba blends contained less carbon in the mixture. Poor vaporization and atomization led to increase in CO at high concentration blends.

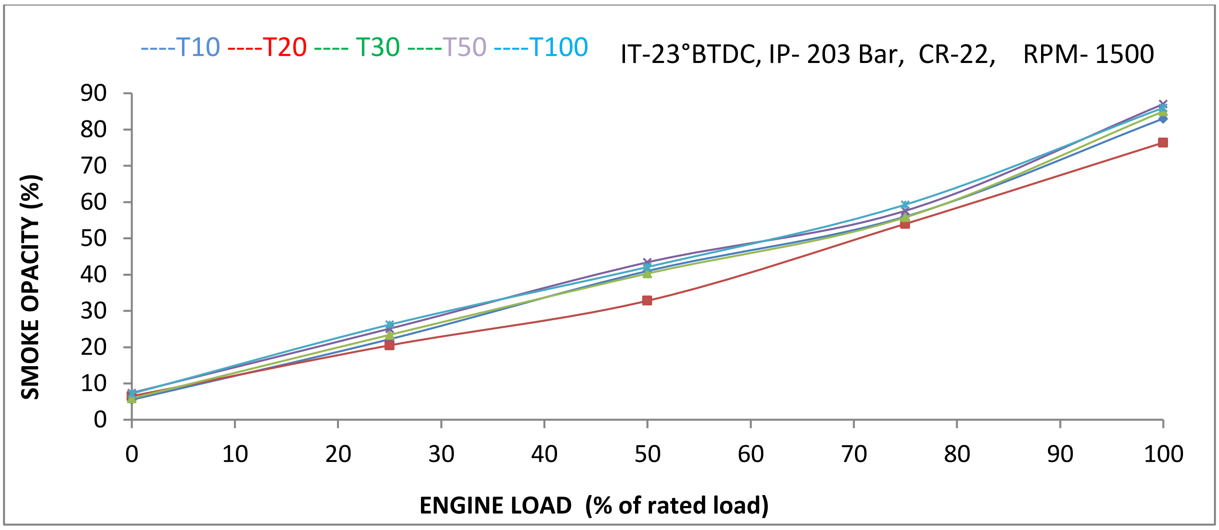


Figure 4.34 Variation of smoke opacity emissions with percentage increase in load for various preheated Thumba oil-diesel blends

CO_2 emission further depends on the combustion quality. Through complete combustion CO formed would be converted into CO_2, thereby reducing the emission of CO, which is hazardous for our environment. It is observed from the Figure 4.36 that high levels of $CO_2$was found upon using the low concentration of Thumba blends as fuel; however, the level of CO_2 reduced with the use of upper blends of Thumba oil with diesel owing to better combustion. CO_2 emission increases with increase in load because of more fuel entry at higher percentage of load.

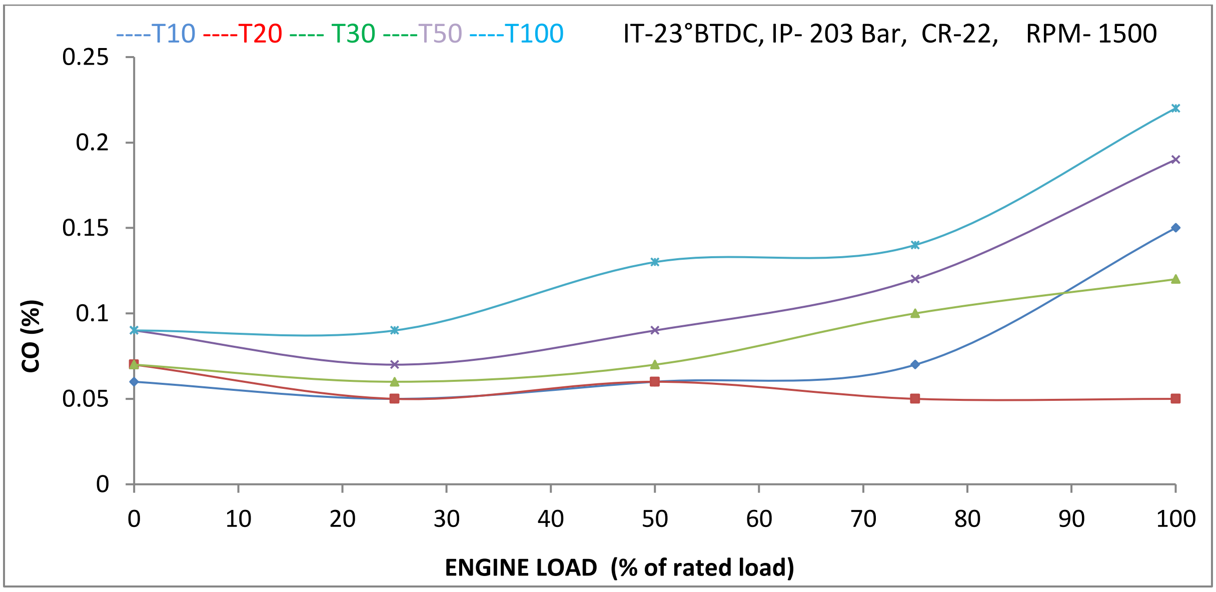


Figure 4.35 Variation of CO emissions with percentage increase in load for various preheated Thumba oil-diesel blends

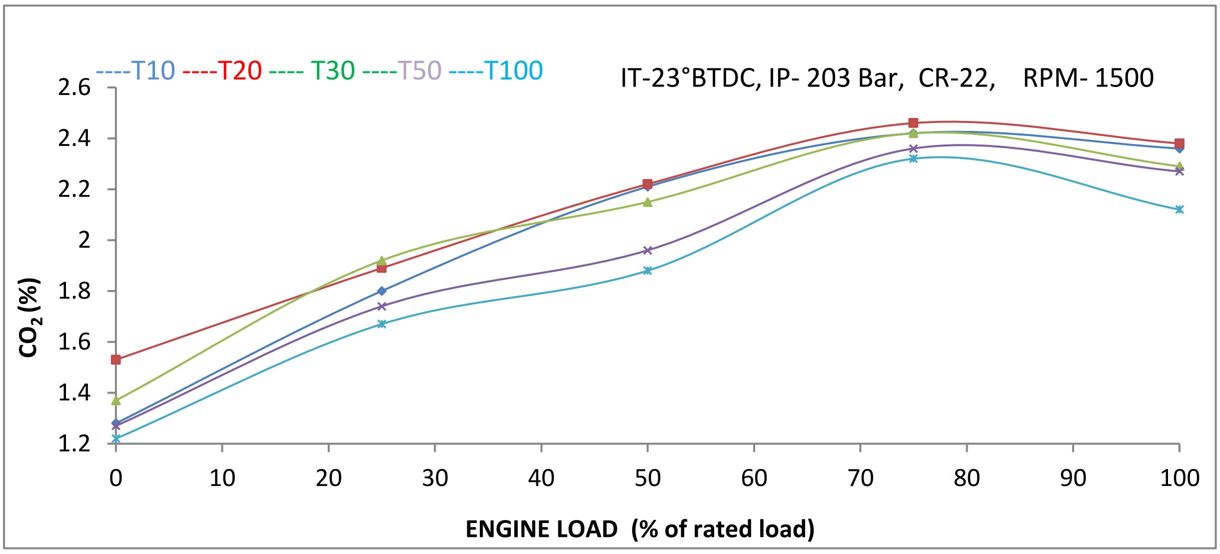


Figure 4.36 Variation of CO_2 emissions with percentage increase in load for various preheated Thumba oil-diesel blends

The variation of hydrocarbon emissions with the load for different preheated Thumba oil-diesel blends is shown in Figure 4.37. There was lowest emission of unburned HC upon operating the engine with T20 preheated Thumba oil-diesel blend. The figure shows that there was sharp decrease in HC emissions with increase in the percentage of load; and then there was gradual increase.

The variation of NOx emissions with the load for different preheated Thumba oil-diesel blends is shown in Figure 4.38. NOx emissions increased with the increase in the percentage of load and after reaching a limit, the emissions decreased further for all the blends of Thumba oil with diesel. NOx emissions were found to be lowest for the preheated T20 Thumba oil-diesel blend.

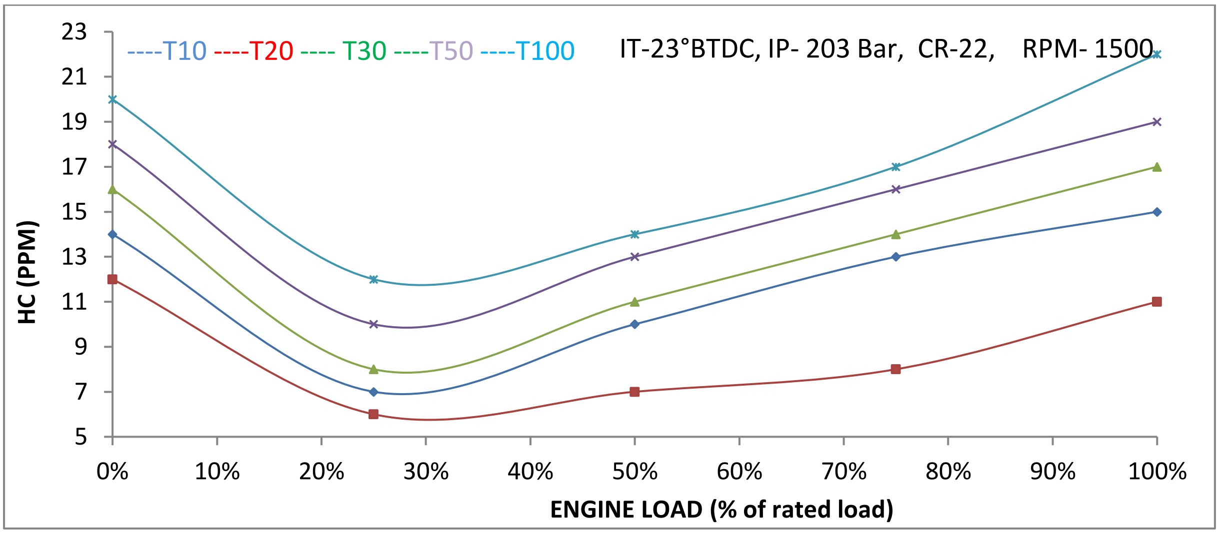


Figure 4.37 Variation of HC emissions with percentage increase in load for various preheated Thumba oil-diesel blends

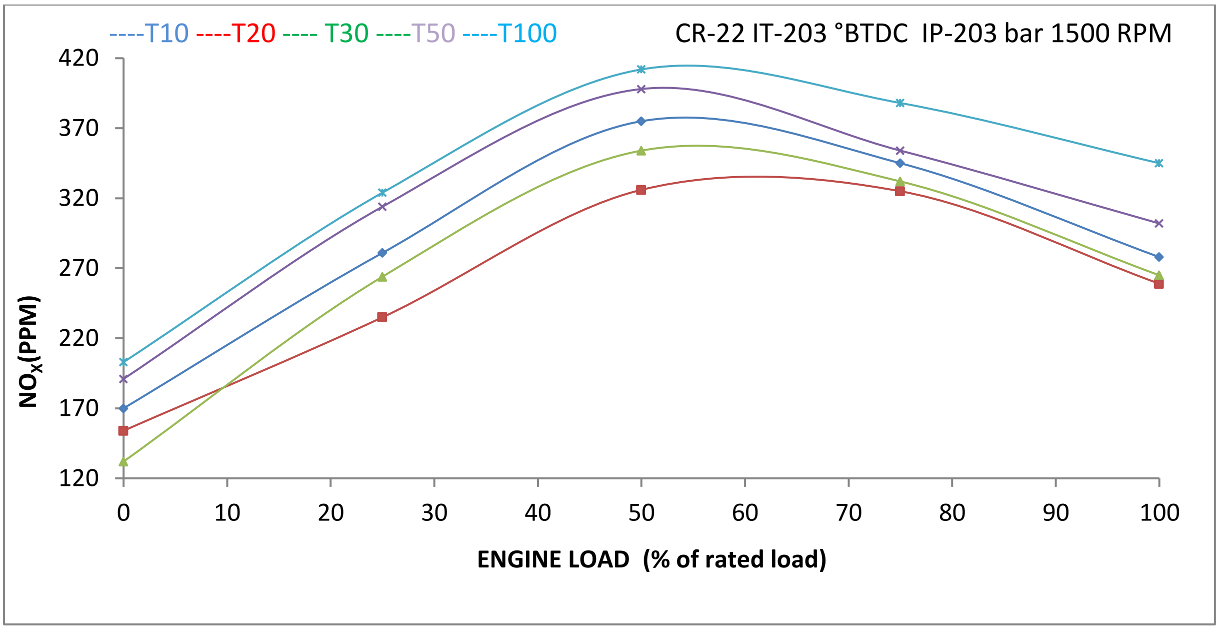


Figure 4.38 Variation of NOx emissions with percentage increase in load for various preheated Thumba oil-diesel blends

4.3.3 Combustion Studies

A graph plotted between cylinder pressure and crank angle for different Thumba oil blends is shown in Figure 4.39. The maximum cylinder pressure was found for the preheated pure Thumba oil because of the initiation of rapid combustion process owing to the physical properties of Thumba oil and the presence of oxygen molecules. The angle of maximum pressure increased with increase in the concentration of Thumba oil with diesel fuel. The maximum rate of pressure rise was achieved with T20 preheated Thumba oil blend. The maximum rate of pressure rise with various preheated Thumba oil blends is shown in Figure 4.40. Other combustion characteristics of various preheated Thumba oil-diesel blends are summarized in Table 4.6. The net heat release rate was the highest for T20 Thumba oil blend. When the concentration of Thumba oil increases in the Thumba oil diesel blend than viscosity, surface tension, cetane number, increases, spray cone angle decreases which reduces amount of air entertained in the spray. Lack of enough air in the fuel spray retard completion of combustion, therefore, pressure rise will occur later after TDC hence angle of maximum pressure increases as the blend percentage increases. Maximum rate of pressure rise is found for T20 Thumba oil blend owing to better combustion characteristics, better volatility, and low viscosity.

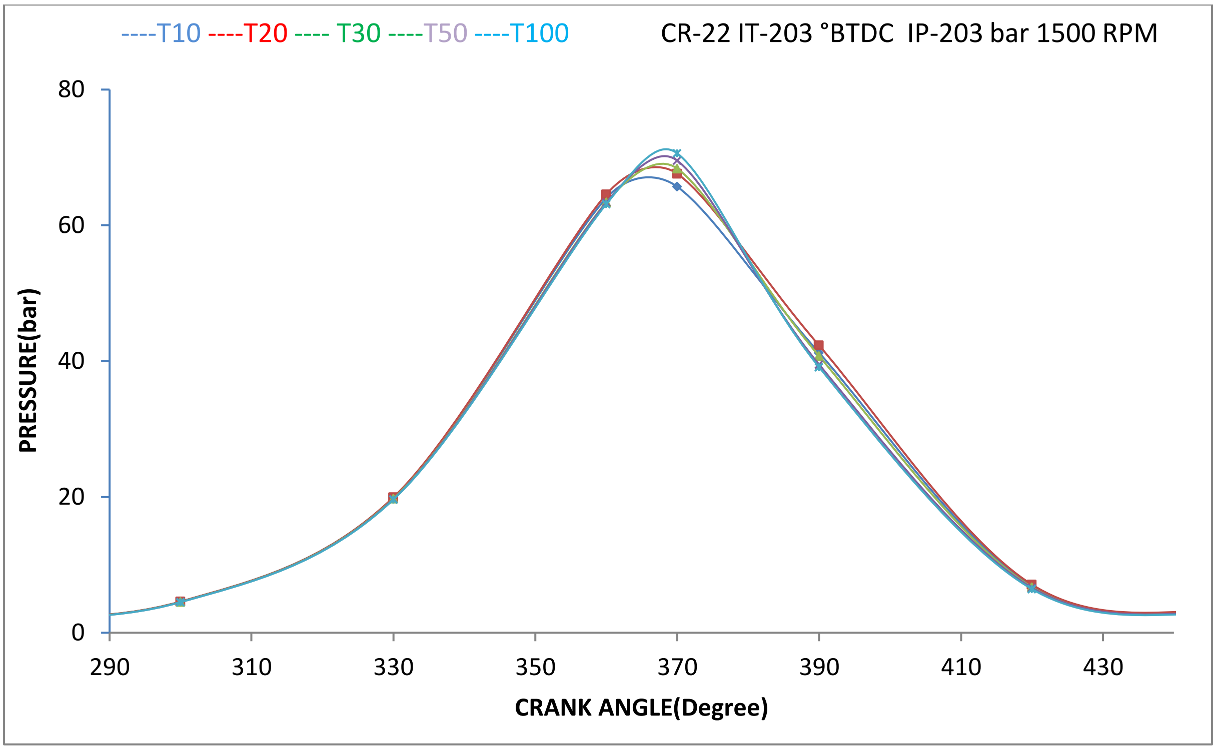


Figure 4.39 Variation of cylinder pressure with the crank angle for various Thumba oil-diesel blends

Table 4.6 Combustion parameters when the engine fuelled with various Thumba oil-diesel blend

Combustion Parameters	T10	T20	T30	T50	T100
Maximum Rate of Pressure Rise (bar/°CA)	4	4.2	4.1	3.9	3.8
Maximum Net Heat Release (MJ)	32.2	35.8	33.01	31.1	30
Ignition Delay (°CA BTDC)	13	13.5	13.2	12.7	12.3
Start of Combustion (°CA)	9.2	9.5	9.3	9	8.7
Maximum Pressure (bar)	71.23	72.43	70.89	70.94	71.2
Angle for Maximum Pressure (°CA ATDC)	8.5	9	9.5	10	10.5

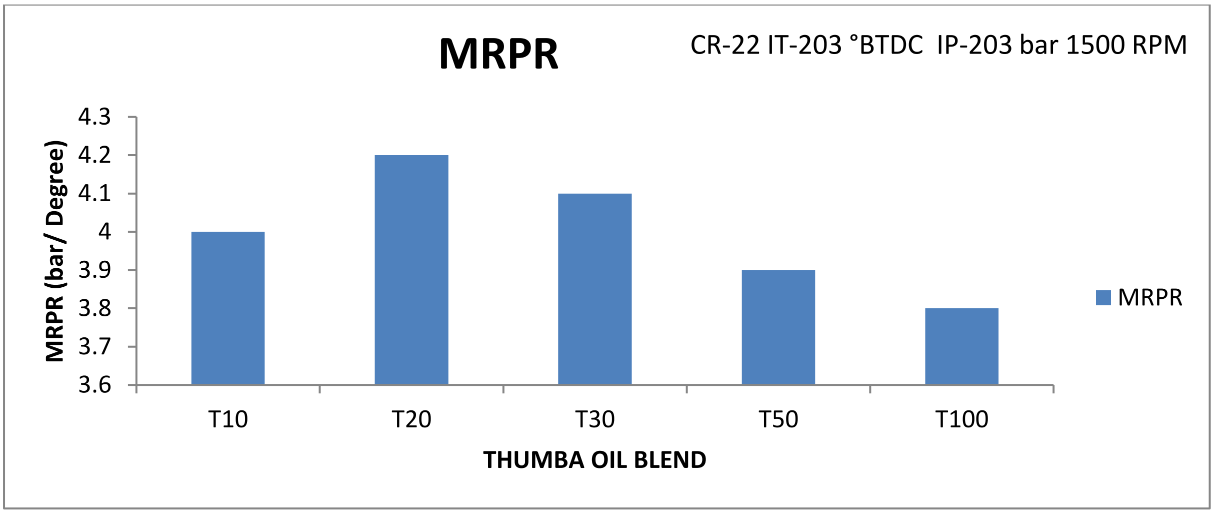


Figure 4.40 Variation of maximum rate of pressure rise for various Thumba oil-diesel blends

4.3.4 Summary of Results

In the experimentation with different preheated Thumba oil-diesel blends, the highest BTE and lowest BSFC were observed for the preheated T20 Thumba oil-diesel blend. The smoke opacity, CO emissions, and HC emissions were found to be lowest for the preheated T20 Thumba oil diesel blend. Therefore, T20 was considered as the optimized Thumba oil diesel blend.

4.4 OPTIMIZATION OF ENGINE PARAMETERS FUELED WITH PREHEATED OPTIMIZED THUMBA OIL BLEND (T20)

From the previous analysis, the T20 preheated Thumba oil-diesel blend was found to be the most optimized blend. Therefore, to evaluate the performance, combustion, and emission characteristics of the engine fuelled with optimized preheated Thumba oil-diesel blend, an experimental study was carried out and engine parameters like compression ratios, injection pressures, and injection timings were further optimized. The various engine input parameters and their ranges are given in Table 4.7.

Table 4.7 Various engine input parameters and their range when engine operated with preheated T20 Thumba oil blend

Sr. No.	Parameters	Unit	Range Values
1	Thumba oil blend	-	Optimized preheated Thumba oil blend T20
2	Load Range	% of rated load	0, 25, 50, 75 and 100
3	Engine Speed	RPM	1500
4	Compression Ratio	-	18, 19, 20, 21 and 22

5	Injection Pressure	Bar	190,203,210
6	Injection Timing	˚CA BTDC	20, 23, 27

4.4.1 Optimization for Compression Ratio

The experimental study was conducted at different compression ratios with an optimized Thumba oil-diesel blend at 1500 rev/min for the entire load range to optimize the compression ratio.

4.4.1.1 Performance Studies

The variation of the BTE with the load at different compression ratios is shown in Figure 4.41. The figure shows that BTE increased with increase in the percentage of load for all the compression ratios. Maximum BTE was observed at compression ratio of 22. High compression ratio enhanced the air density present in the cylinder, which in turn led to efficient combustion. However, poor BTE was observed at lower compression ratios. The reduced brake thermal efficiency was probably because of the dilution of charge at elevated compression ratios.

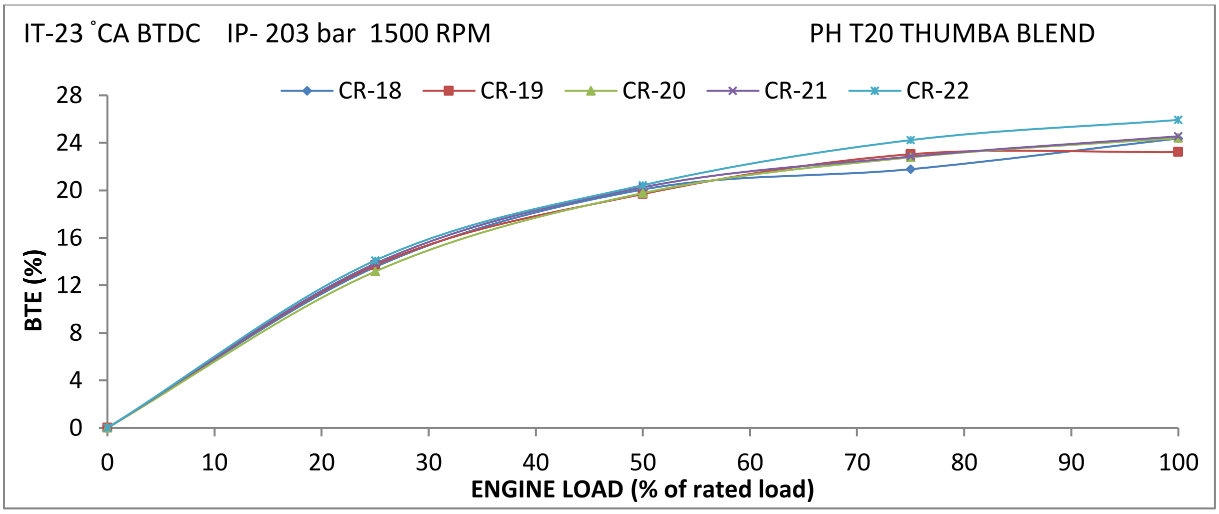


Figure 4.41 Variation of brake thermal efficiency with percentage increase in load at different compression ratios for preheated T20 Thumba oil blend

A graph was plotted to observe the variation of BSFC with the load at different compression ratios for the preheated optimized Thumba oil-diesel blend as shown in Figure 4.42. BSFC was observed to be minimum for the compression ratio of 22 owing to better combustion. Moreover, BSFC decreased with the increase in the percentage of load for all the compression ratios.

Exhaust gas temperature is a measure of the amount of the unutilized fuel in the cylinder. A graph was plotted between exhaust gas temperature and load with T20 preheated Thumba oil blend for all the compression ratios as shown in Figure 4.43. The exhaust gas temperature increased with increase in the percentage of load and lowest exhaust gas temperature was observed at the compression ratio of 22.

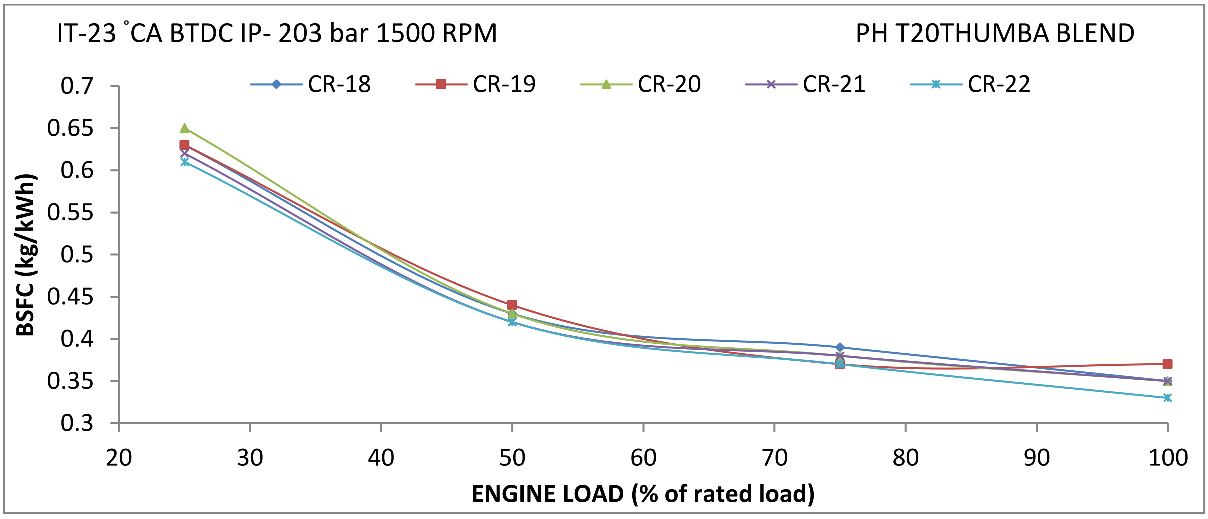


Figure 4.42 Variation of brake specific fuel consumption with percentage increase in load at different compression ratios for preheated T20 Thumba oil blend with diesel

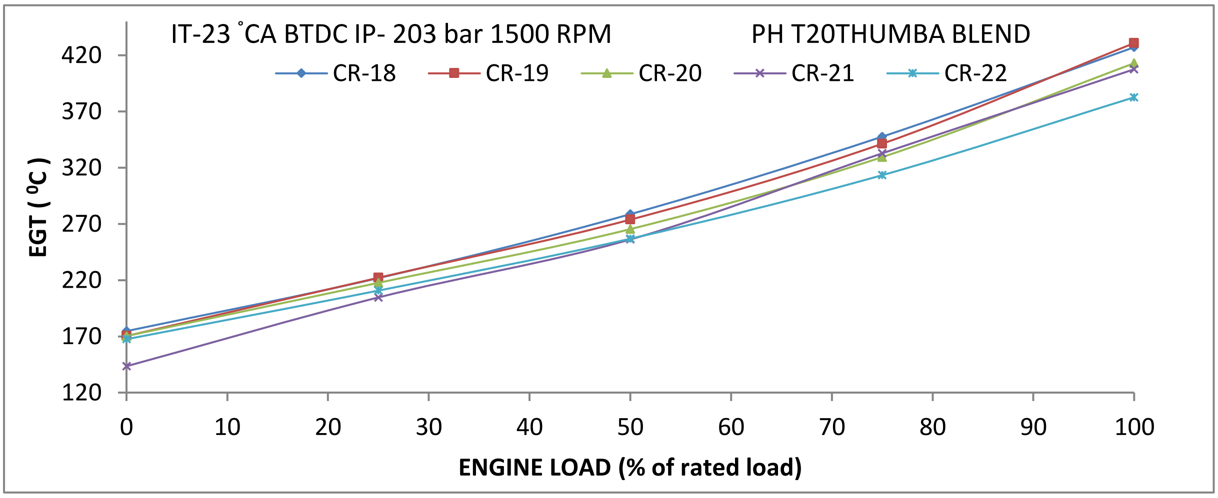


Figure 4.43 Variation of exhaust gas temperature with percentage increase in load at different Compression Ratios for preheated T20 Thumba oil blend

4.4.1.2 Emission Studies

The variation of smoke opacity with the load at different compression ratio is shown in Figure 4.44. The figure shows that the smoke opacity increases with increase in load. Smoke opacity decreased with increase in compression ratio because of poor atomization and combustion at lower compression ratios. Smoke opacity was found to be minimum at the compression ratio of 22 owing to better atomization and combustion.

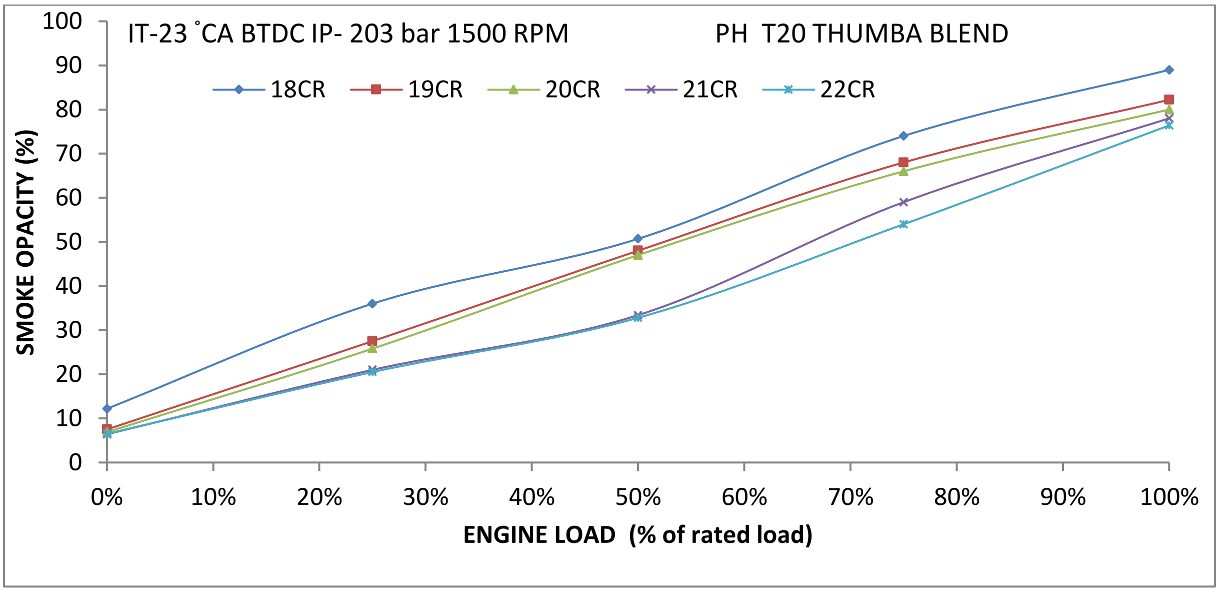


Figure 4.44 Variation of smoke opacity with percentage increase in load at different compression ratios for preheated T20 Thumba oil blend

A graph was plotted to observe the variation of CO emissions with the load at different compression ratios as shown in Figure 4.45. CO emissions increased with increase in the percentage of load as observed in the figure. CO emissions were observed to be minimum at the compression ratio of 22 owing to better combustion, but these emissions were found to be slightly high at lower compression ratios.

The variation of CO_2emissions with the load at different compression ratios is shown in Figure 4.46. Maximum CO_2 emissions was observed at the compression ratio of 22 for T20 preheated Thumba oil-diesel blend.

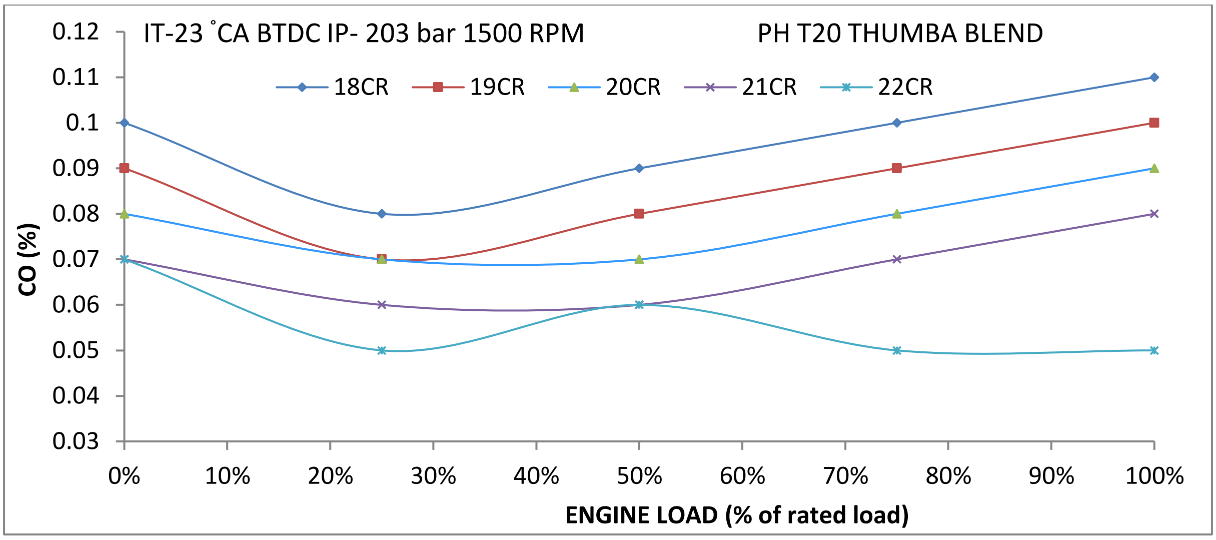


Figure 4.45 Variation of CO emissions with percentage increase in load at different compression ratios for preheated T20 Thumba oil blend

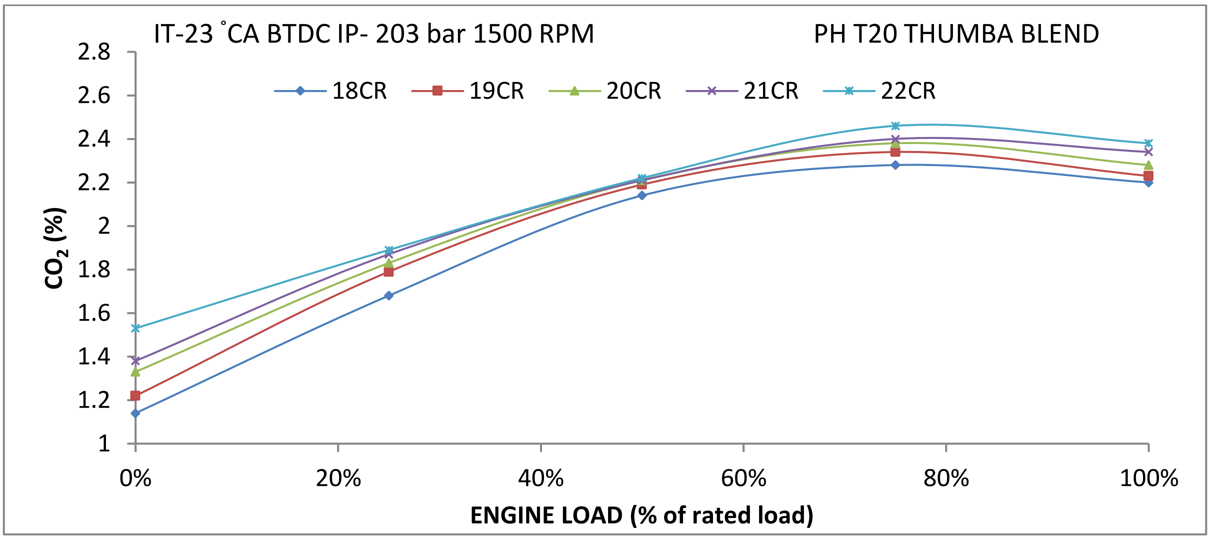


Figure 4.46 Variation of CO_2 emissions with percentage increase in load at different compression ratios for preheated T20 Thumba oil blend

A graph was plotted to observe the variation of HC emissions with the load at different compression ratios as shown in Figure 4.47. Hydrocarbon emissions increased with decrease in compression ratio as shown in the figure. HC emissions were lowest for the compression ratio of 22.

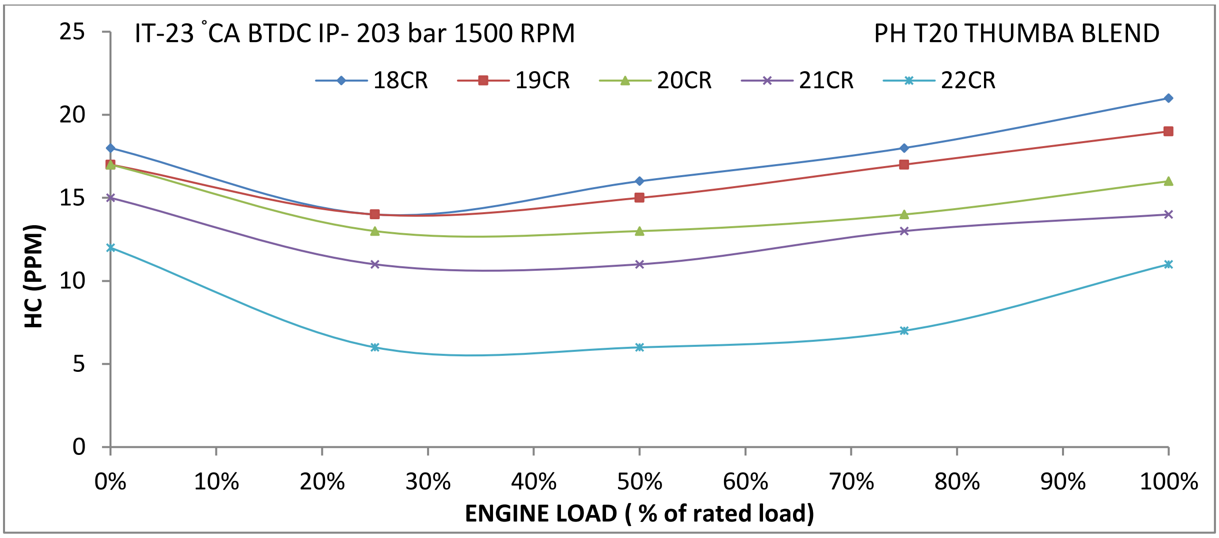


Figure 4.47 Variation of HC emissions with percentage increase in load at different Compression Ratios for preheated T20 Thumba oil blend

The variation of NOx emissions with the load at different compression ratios for preheated optimized Thumba oil blend is shown in Figure 4.48. NOx emissions were found to be lowest for the compression ratio of 22. Initially, there was increase in these emissions, but after reaching a point the emissions gradually decreased with increase in load.

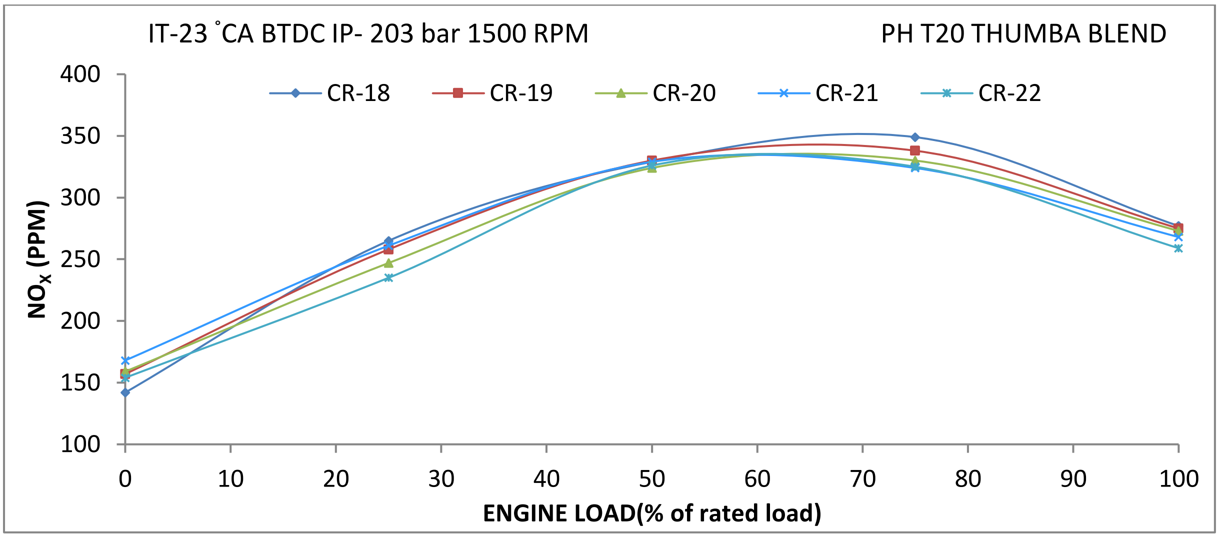


Figure 4.48 Variation of NOx emissions with percentage increase in load at different compression ratios for the preheated T20 Thumba oil blend

4.4.1.3 Combustion Studies

The combustion characteristics of engine fuelled with preheated T20 Thumba oil diesel blend has been analyzed in this section. The behavior of engine was observed for different compression ratios. The variation of cylinder pressure with the crank angle for different compression ratios is shown in Figure 4.49. Cylinder pressure increased with increase in compression ratio and maximum pressure was reached at the compression ratio of 22; subsequent to which the angle of maximum pressure decreased with increase in compression ratio.

The maximum rate of pressure rise at different compression ratios is shown in Figure 4.50 and the pressure rise was found to be maximum for the compression ratio of 22 possibly owing to complete and better combustion. The net heat release rate was also found to be maximum for the compression ratio of 22 with minimum delay period.

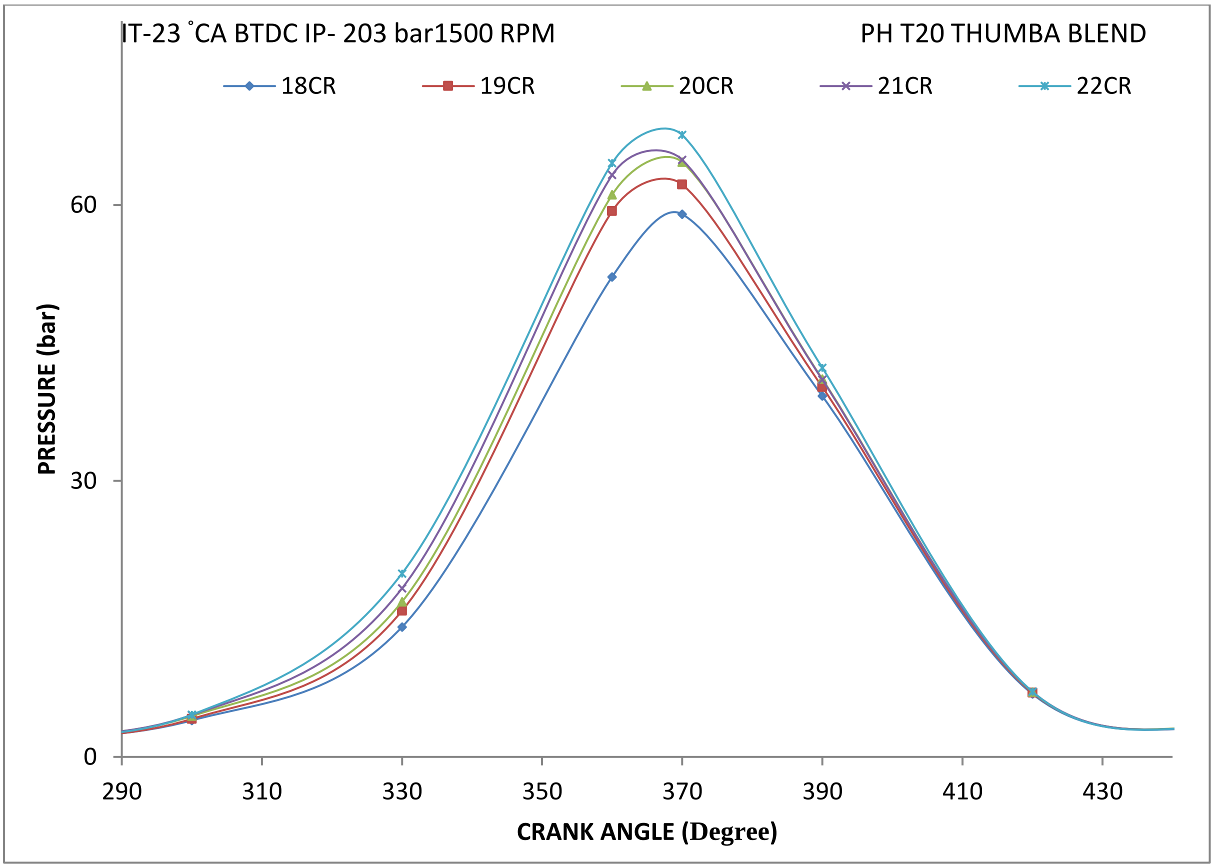


Figure 4.49 Variation of cylinder pressure with crank angle at different compression ratios for the preheated T20 Thumba oil blend

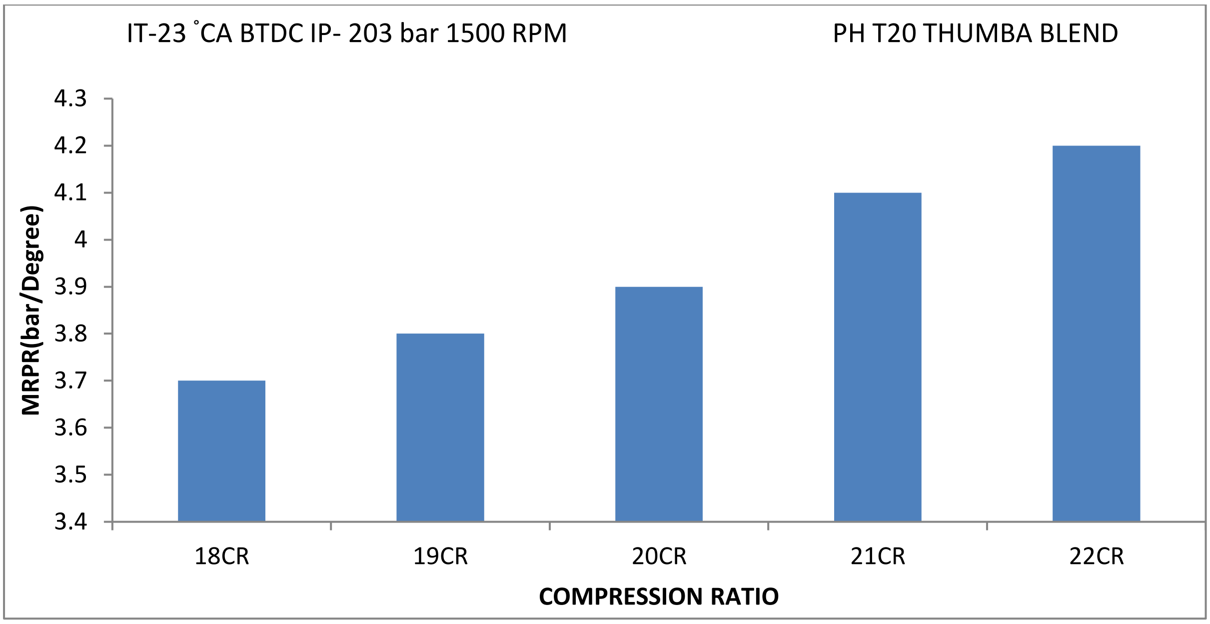


Figure 4.50 Variation of the maximum rate of pressure rise at different compression ratios for the preheated T20 Thumba oil blend

Table 4.8 Combustion parameters at different Compression Ratios for preheated T20 Thumba oil blend

Combustion Parameters	18CR	19CR	20CR	21CR	22CR
Maximum Rate of Pressure Rise (bar/˚CA)	3.7	3.8	3.9	4.1	4.2
Maximum Net Heat Release (MJ)	33.9	34.1	34.5	34.8	35.8
Ignition Delay (˚CA BTDC)	14.5	14.3	14.25	14	13.5
Start of Combustion (˚CA)	8.5	8.7	8.75	9	9.5
Maximum Pressure (bar)	61.87	65.75	69	70.93	72.43
Angle for Maximum Pressure (˚CA ATDC)	7	7.5	8	8.5	9

4.4.2 Optimization for Injection Pressure

Injection pressure is an important parameter that not only influences the engine characteristics but also affects the spray characteristics, atomization of the fuel, and mixing quality of the air-fuel mixture. Experimentation was conducted to find the injection pressure at which the engine performs excellently with the use of preheated T20 Thumba oil and diesel blend. The injection pressures employed were 190 bar, 203 bar, and 210 bar.

4.4.2.1 Performance Studies

The variation of BTE with the load at different injection pressures is shown in Figure 4.51. BTE increased with increase in the percentage of load at different injection pressures, and it was found to be maximum at the injection pressure of 203 bar due to better spray, atomization, and mixing of air and fuel. Vegetable oils are highly viscous, which leads to poor fuel atomization; hence, there is a need for high injection pressure for proper atomization.

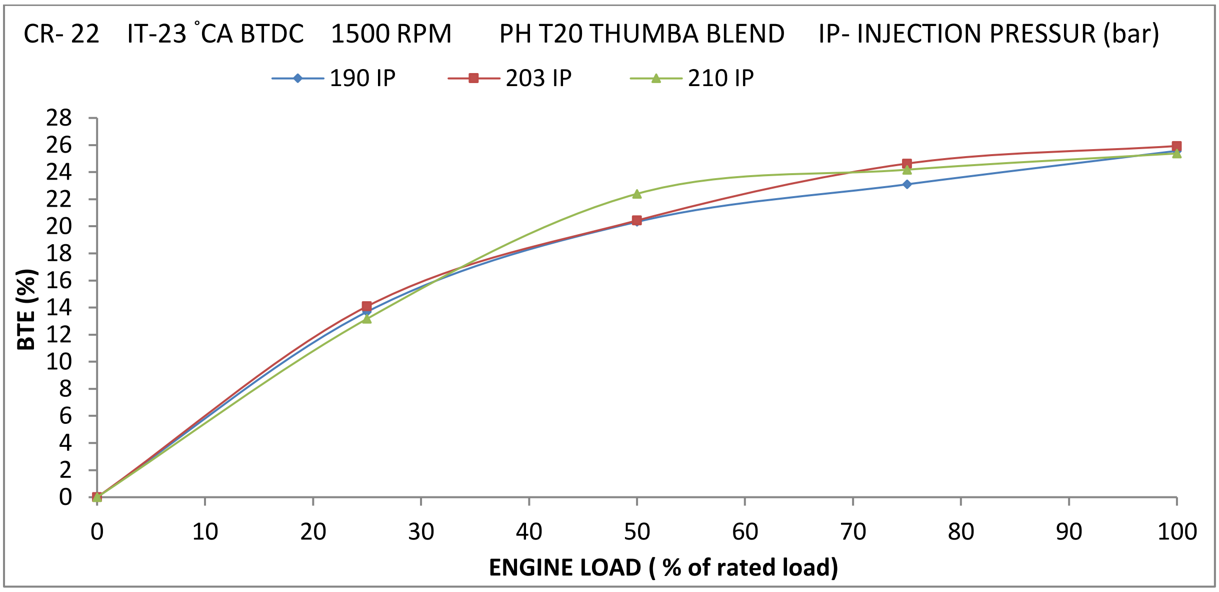


Figure 4.51 Variation of brake thermal efficiency with % load at different injection pressures for preheated T20 Thumba oil blend.

BSFC as a performance parameter was plotted with the load at different injection pressures, which is shown in the Figure 4.52. The figure shows that BSFC is minimum for injection pressure of 203 bar owing to better atomization and intermixing of fuel. BSFC for the injection pressure of 190 and 210 bars were very close to the injection pressure of 203 bar.

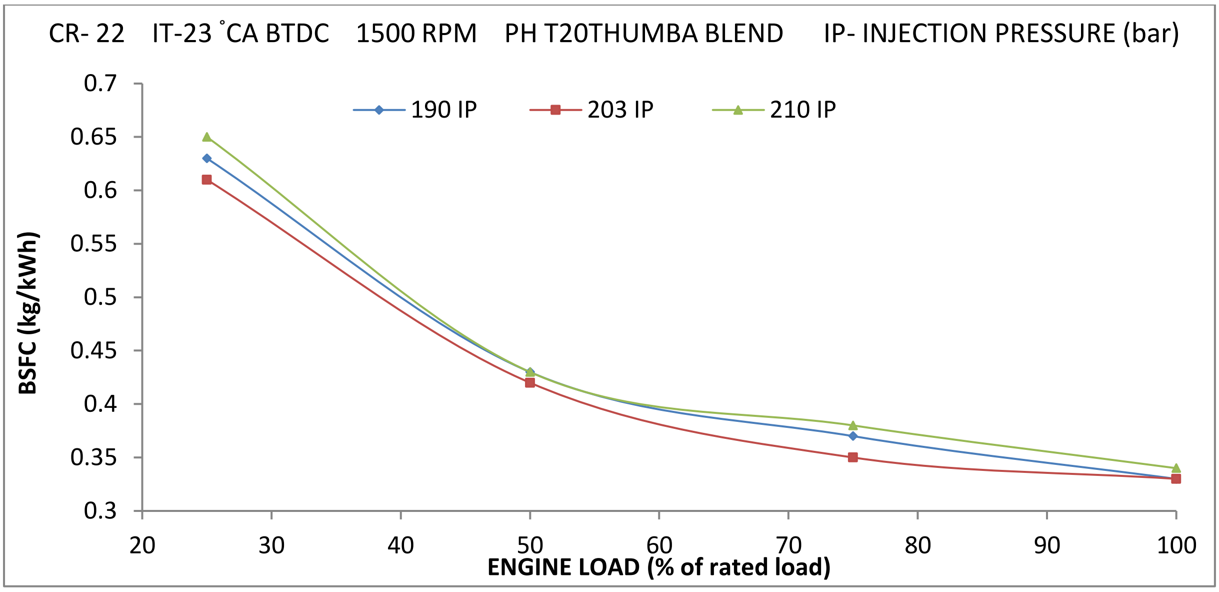


Figure 4.52 Variation of brake specific fuel consumption with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

The variation of exhaust gas temperature with the load at different injection pressures is plotted in Figure 4.53. Exhaust gas temperature increased with increase in the percentage of load for all injection pressure, but it was lowest for the injection pressure of 203 bar.

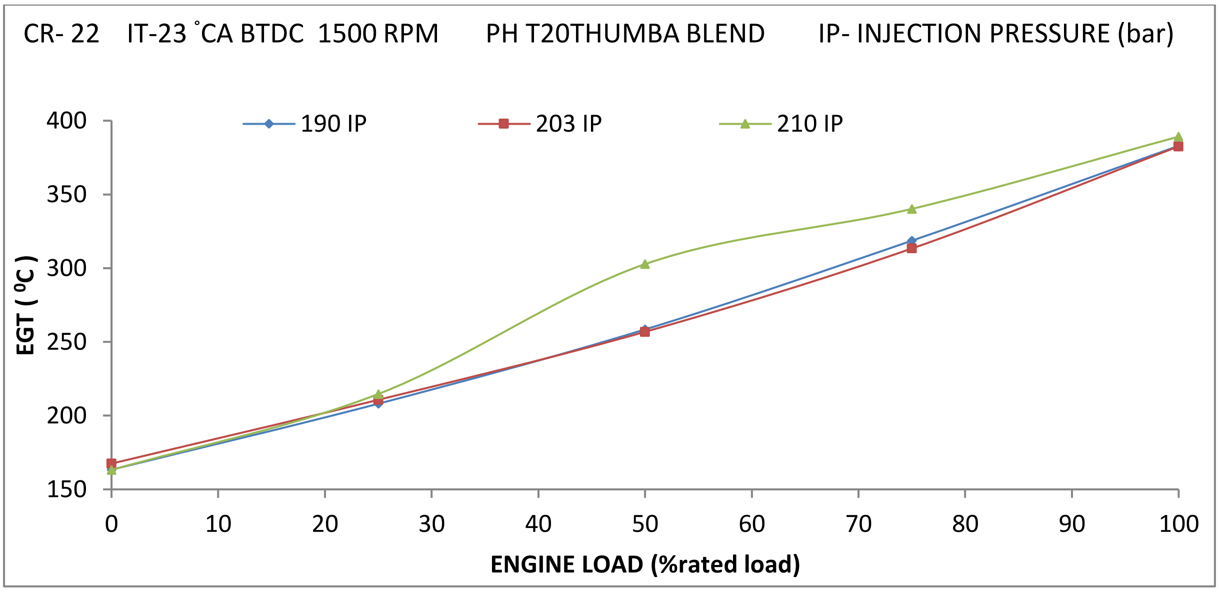


Figure 4.53 Variation of exhaust gas temperature with percentage increase load at different injection pressures for the preheated T20 Thumba oil blend

4.4.2.2 Emission Studies

The effect of injection pressure on engine emission characteristics operated with the preheated optimized T20 Thumba oil diesel blend has been analyzed in this section. Smoke opacity, CO, CO_2, HC, and NOx emissions were considered.

Smoke was formed in the fuel rich zone at high temperature and pressure. The variation of smoke opacity with load at different injection pressures is shown in Figure 4.54. Smoke opacity increased with the increase in the percentage of load, and it was found to be minimum at the injection pressure of 203 bar owing to proper atomization and combustion.

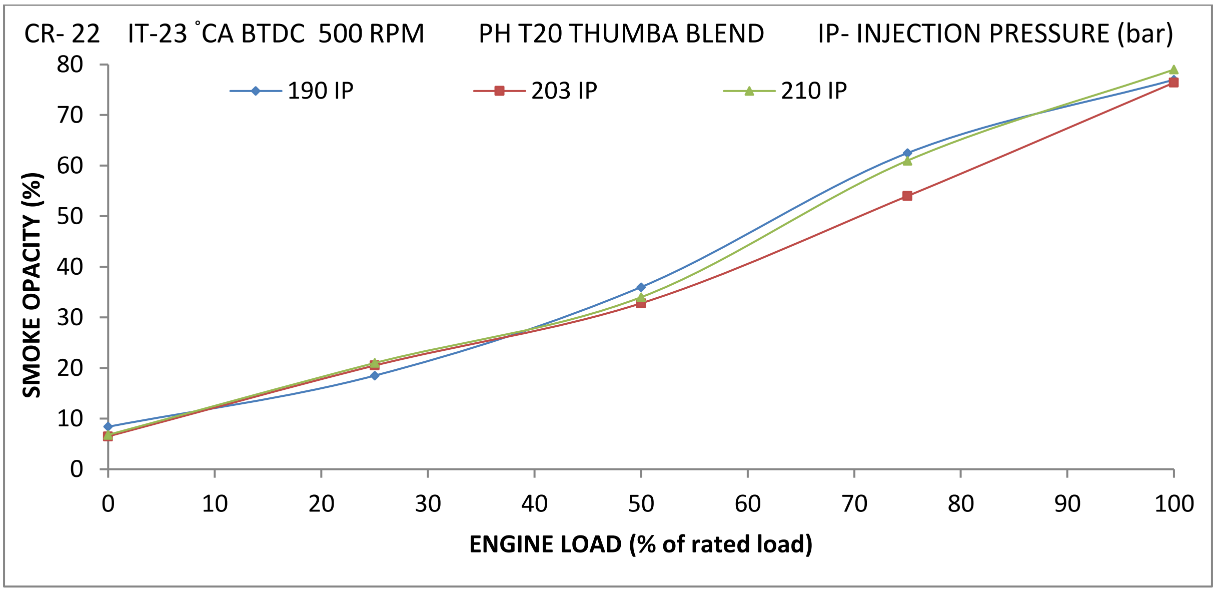


Figure 4.54 Variation of smoke opacity with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

The variation of CO with the load at different injection pressures is plotted in Figure 4.55. CO emissions was found to be minimum for the injection pressure of 203 bar possibly owing to almost complete combustion and better intermixing of fuel at this injection pressure. CO emissions for injection pressure of 190 and 210 bars were marginally higher than the injection pressure of 203 bar.

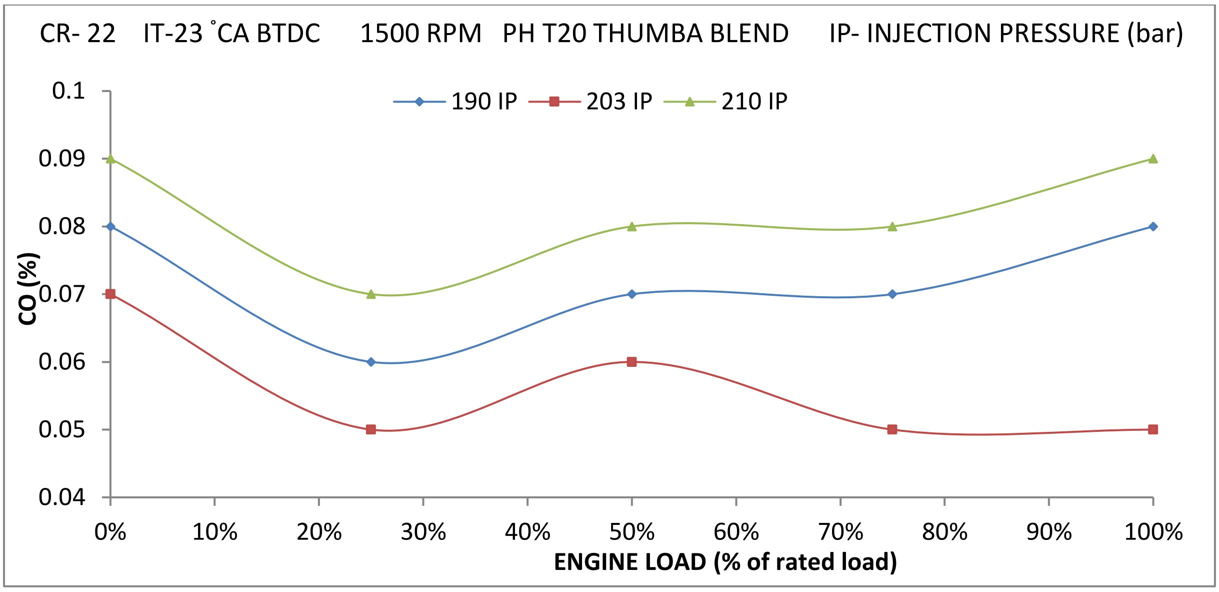


Figure 4.55 Variation of CO emissions with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

CO_2 emissions were maximum for the injection pressure at which combustion is complete. The variation of CO_2 emission with the load at different injection pressures for the preheated optimized Thumba oil diesel blend is shown in Figure 4.56. The highest CO_2 emissions were found for injection pressure of 203 bar owing to better combustion of the preheated optimized Thumba oil diesel blend.

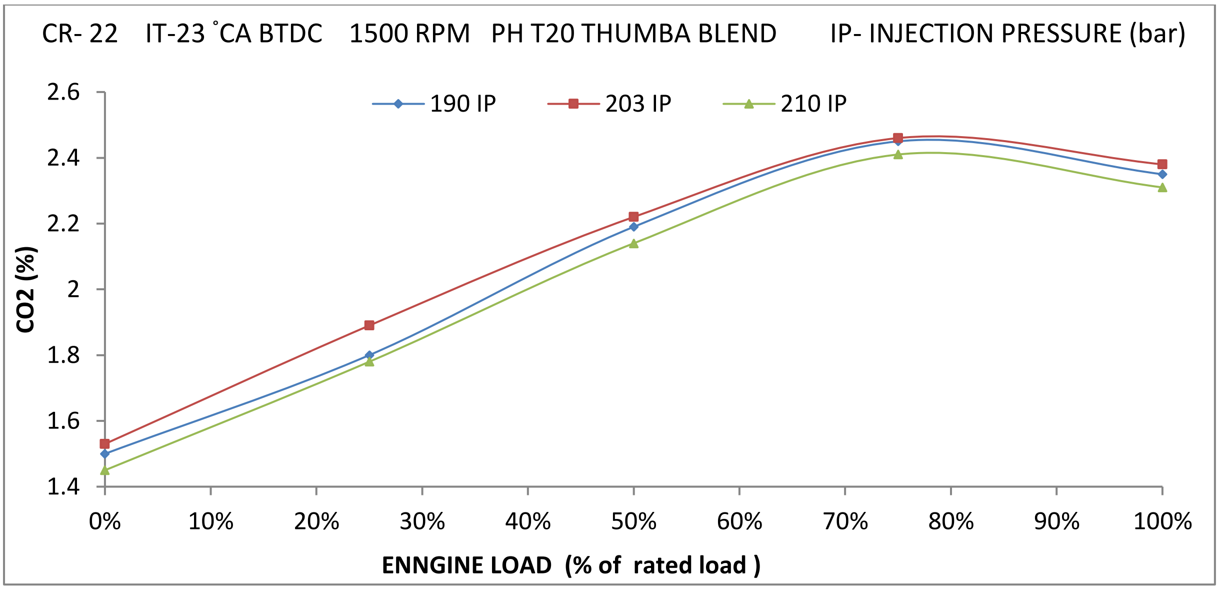


Figure 4.56 Variation of CO_2 emissions with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

The variation of HC emission with percentage increase in load for the optimized T20 Thumba oil blend with diesel is shown in Figure 4.57. HC emissions were observed to be the lowest at injection pressure of 203 bar possibly due to improved combustion at the injection pressure of 203 bars.

The variation of NOx emissions with load at different injection pressures for the preheated optimized Thumba oil diesel blend is indicated in Figure 4.58. NOx emissions were lowest for the injection pressure of 203 bar and minimal variation in the NOx emissions were observed for other injection pressures.

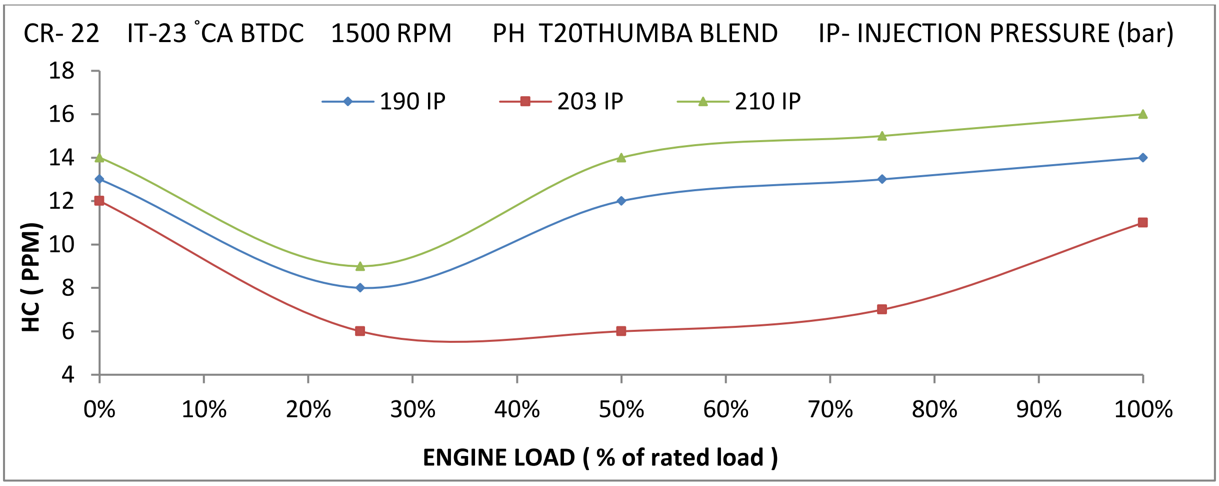


Figure 4.57 Variation of HC emissions with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

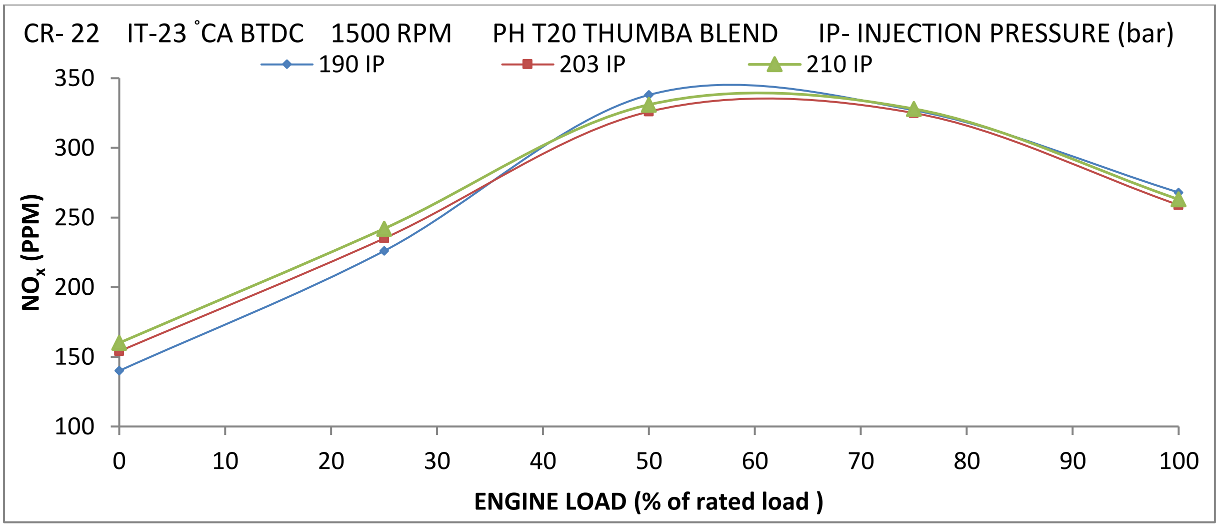


Figure 4.58 Variation of NOx emissions with percentage increase in load at different injection pressures for the preheated T20 Thumba oil blend

4.4.2.3 Combustion Studies

The combustion characteristics of engine operated with the preheated T20 Thumba vegetable oil at different injection pressures has been analyzed here. The variation of cylinder pressure with crank angle at different injection pressures is shown in Figure 4.59. Cylinder pressure increased with increase in injection pressure, but the angle of maximum pressure decreased with increase in injection pressure.

The maximum increase in the rate of pressure rise for different injection pressures are plotted in Figure 4.60. The rate of pressure rise was maximum for the injection pressure of 203 bar owing to better combustion and intermixing of fuel and air. The net heat release rate was also found to be the maximum for the injection pressure of 203 bar. Moreover, for the injection pressure of 203 bar, ignition delay was the least probably due to the early start of combustion at this injection pressure. Other combustion parameters at different injection pressures are given in Table 4.9.

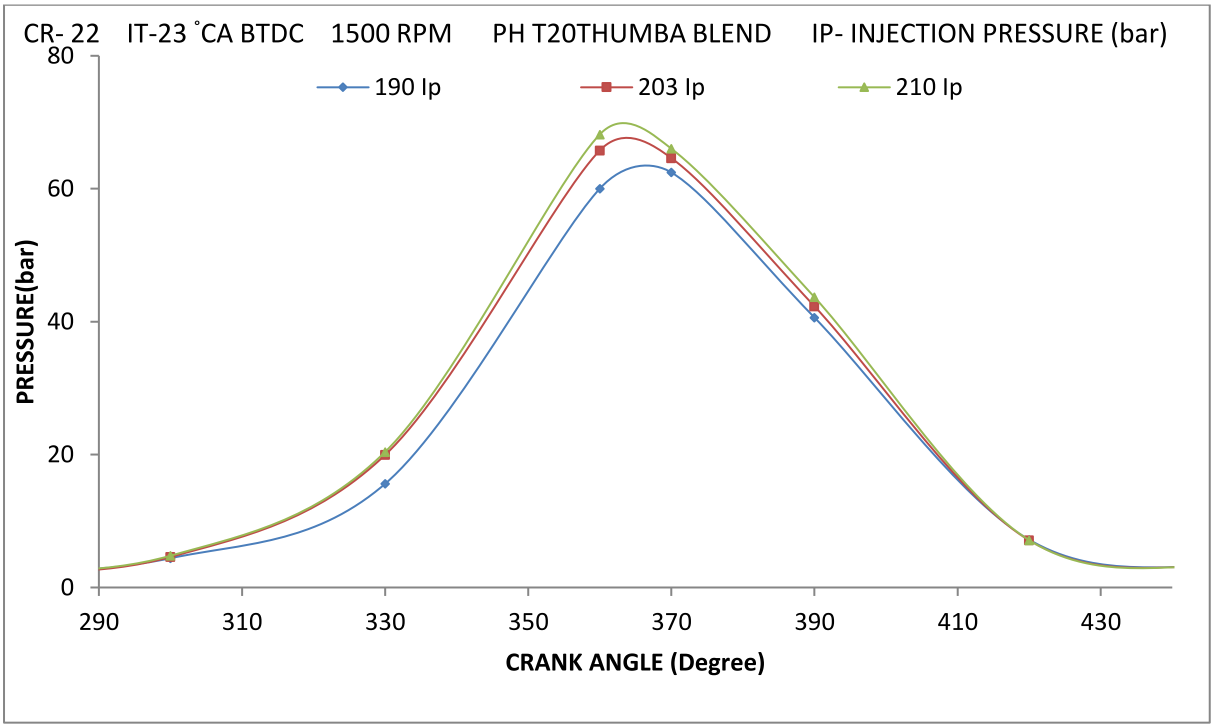


Figure 4.59 Variation of cylinder pressure with crank angle at different injection pressures for the preheated T20 Thumba oil blend

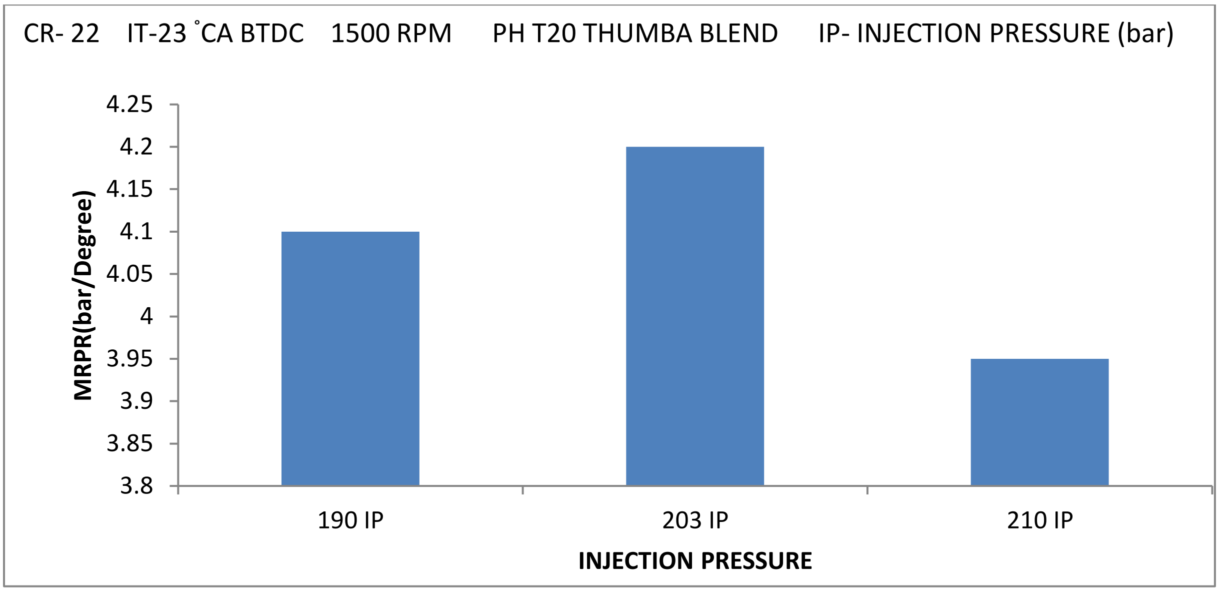


Figure 4.60 Variation of the maximum rate of pressure rises at different injection pressures for the preheated T20 Thumba oil blend

Table: 4.9 Combustion parameters for different injection pressures fuelled with Preheated T20 Thumba oil blend

Combustion Parameters	IP-190 bar	IP -203 bar	IP-210 bar
Maximum Rate of Pressure Rise (bar/˚CA)	4.1	4.2	3.95
Maximum Net Heat Release (MJ)	35.1	35.8	34.9
Ignition Delay (˚CA BTDC)	13.9	13.5	13.6
Start of Combustion (˚CA)	9.1	9.5	9.4
Maximum Pressure (bar)	68.8	72.43	72.1
Angle for Maximum Pressure (˚CA ATDC)	9.5	9	8.5

4.4.3 Optimization for Injection Timing

Injection timing is an important parameter that affects the engine performance, combustion, and emission characteristics. In the present study, the injection timings considered were 20°CA, 23°CA, and 27°CA BTDC. The experiments were conducted at different injection timings while keeping other parameters constant, i.e., optimized injection pressure of 203 bar, the compression ratio of 22, and engine RPM 1500.

4.4.3.1 Performance Studies

The effect of injection timing on engine performance characteristics operated with the preheated T20 Thumba oil diesel blend has been presented in this section. Moreover, an optimized injection timing that provided maximum BTE and minimum BSFC were evaluated.

A graph plotted between BTE and load at different injection timings for the T20 Thumba oil blend is shown in Figure 4.61. BTE increased with increase in the percentage of load, and BTE was found to be maximum for 23 °CA BTDC injection timing;; hence, the optimum injection timing was 23 °CA BTDC.

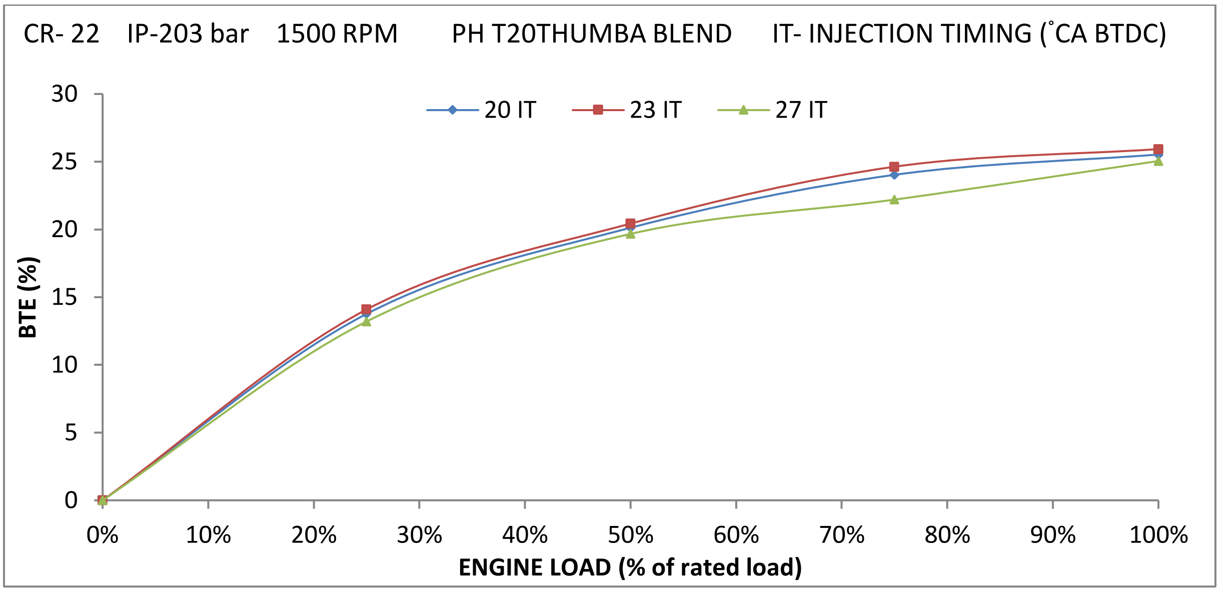


Figure 4.61 Variation of brake thermal efficiency with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

The variation of BSFC with the load at different injection timings is plotted in Figure 4.62. BSFC decreased with increase in the percentage of load, and it is the lowest at injection timing 23°CA BTDC for preheated T20 Thumba oil diesel blend.

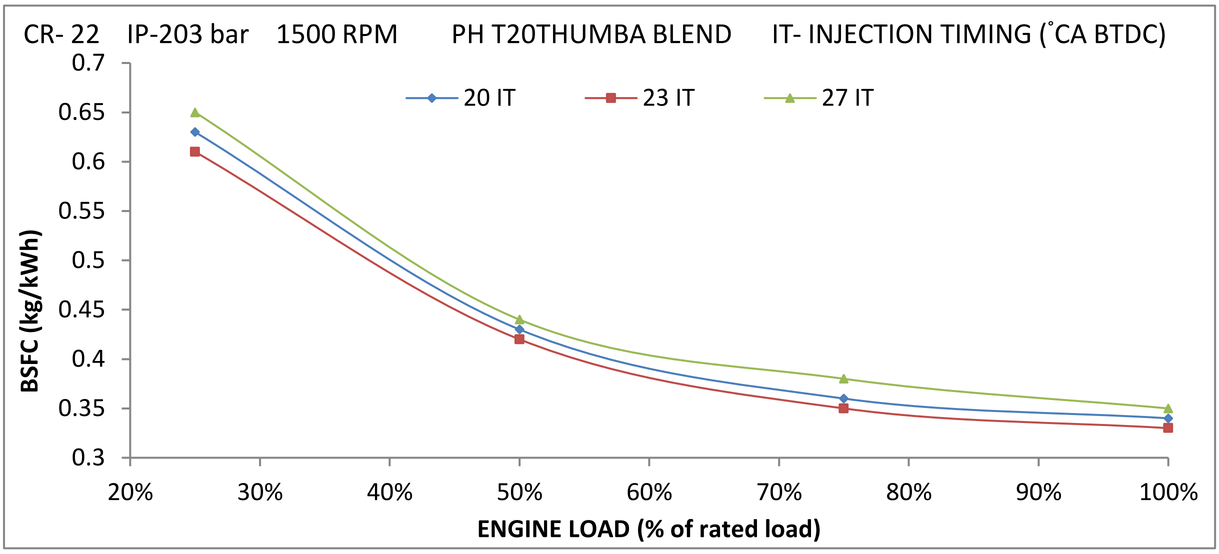


Figure 4.62 Variation of brake specific with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

The variation of exhaust gas temperature with the load at different injection timings is indicated in Figure 4.63. Exhaust gas temperatures at different injection timings were found to increase with increase in the percentage of load. Exhaust gas temperatures were very close at the injection timings considered in this study, and it was the lowest at 23 °CA BTDC.

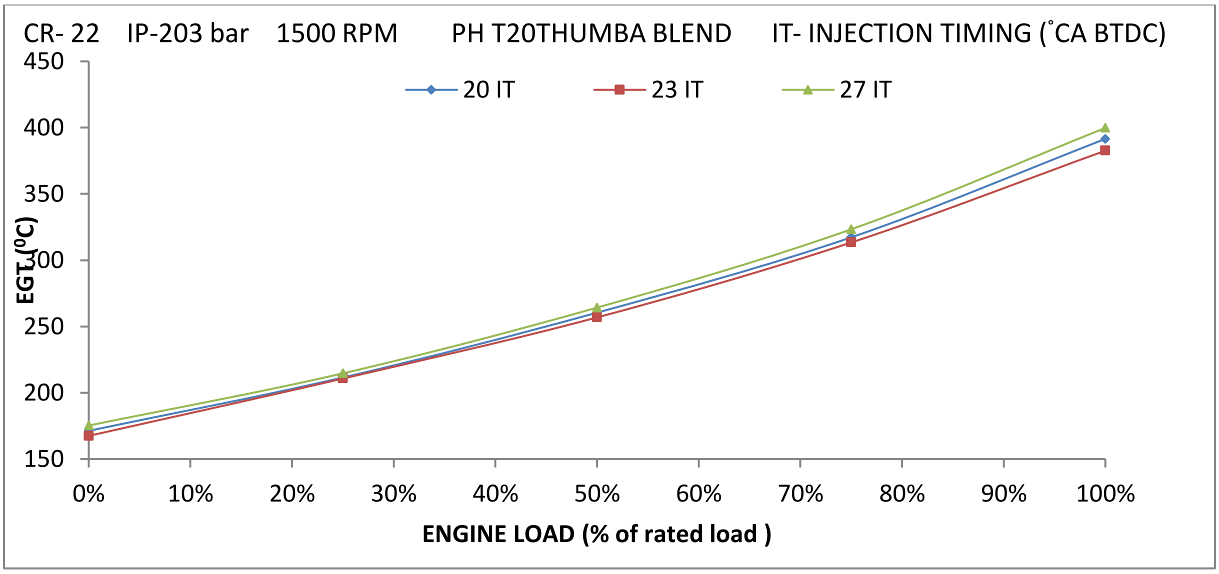


Figure 4.63 Variation of exhaust gas temperature with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

4.4.3.2 Emission Studies

The effect of injection timing on engine emission characteristics fuelled with the preheated T20 Thumba oil diesel blend has been summarized in this section. The variation of smoke opacity with the load at different injection timings is shown in Figure 4.64. It was observed from the plot that smoke opacity increased with increase in the percentage of load. Owing to superior combustion, smoke density was found to be lowest for injection timing of 23°CA BTDC than injection timing of 20 and 27 °CA BTDC.

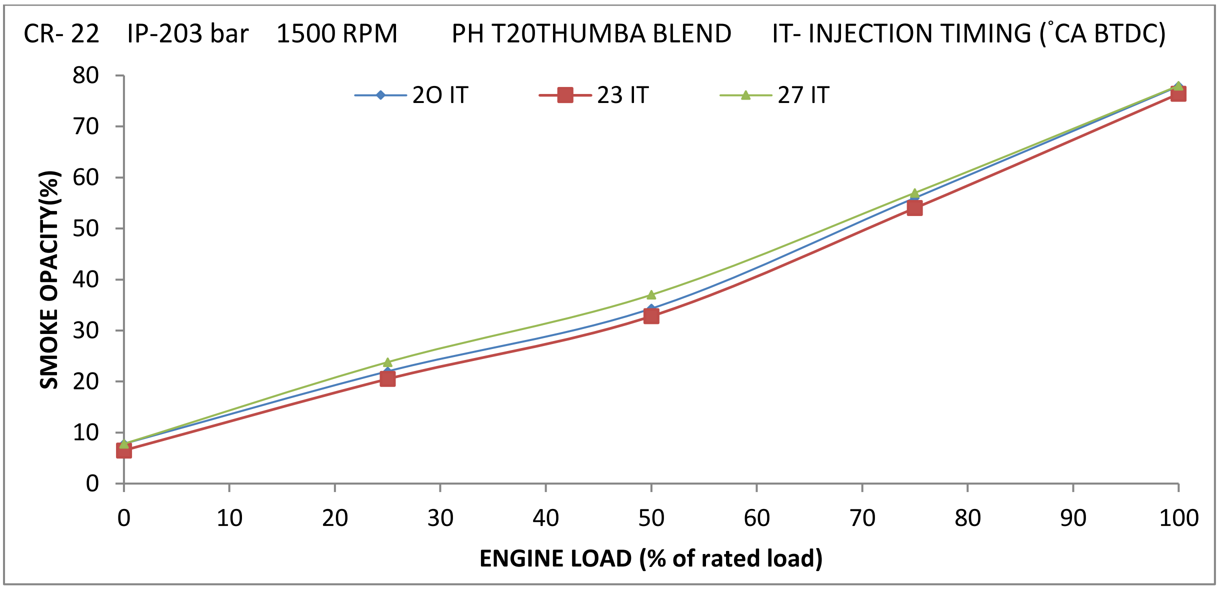


Figure: 4.64 Variation of smoke opacity with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

The variation of CO emissions with the load for the preheated T20 Thumba oil diesel blend at different injection timings is shown in Figure 4.65. CO emissions decreased up to a limit and then gradually increased with increase in load. CO emissions were found to be minimum for injection timing of 23°CA BTDC owing to almost complete combustion.

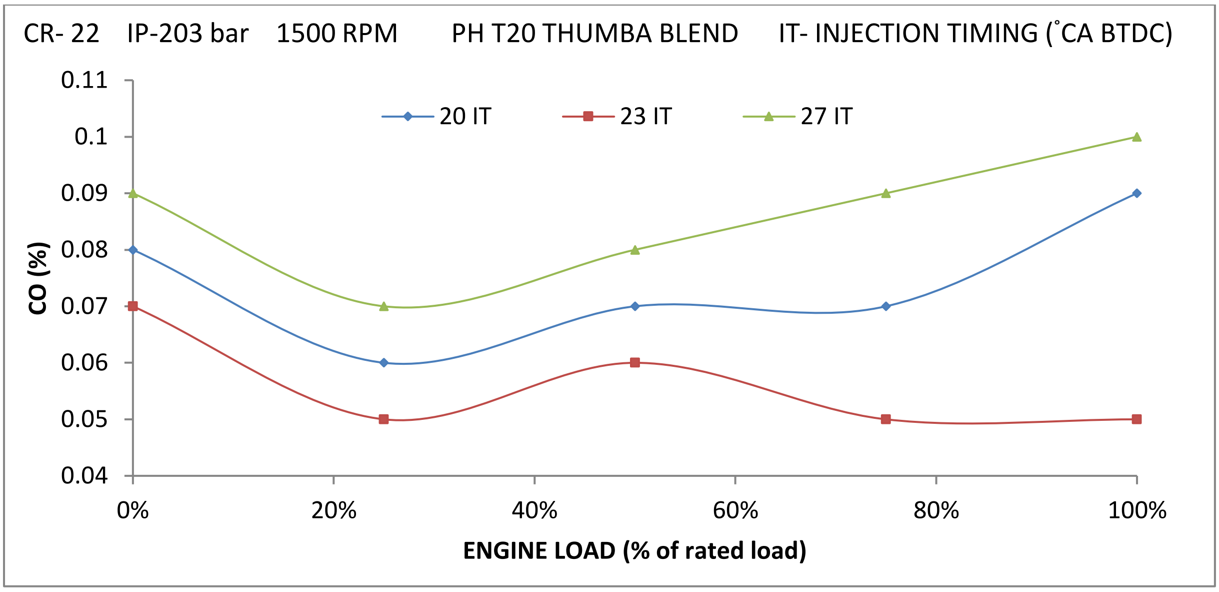


Figure: 4.65 Variation of CO emissions with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

Higher CO_2 emissions in exhaust gases indicated better combustion quality. CO_2 emissions with the load at different injection timings are shown in Figure 4.66. CO_2 emissions increased with the increase in the percentage of load, and the emissions were maximum for injection timing of 23°CA BTDC owing to better combustion than others.

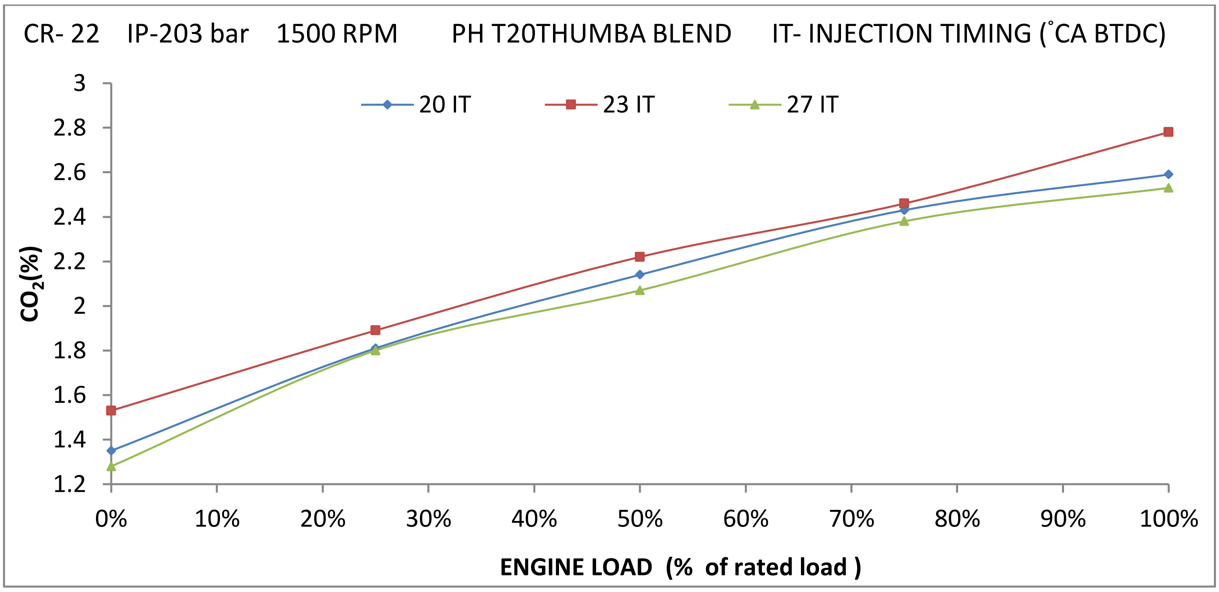


Figure: 4.66 Variation of CO_2 emissions with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

The variation of hydrocarbon emissions with the load at different injection timings for the preheated T20 Thumba oil-diesel blend is plotted in Figure 4.67. HC emissions were lowest at an injection timing of 23°CA BTDC because of superior combustion than other injection timing.

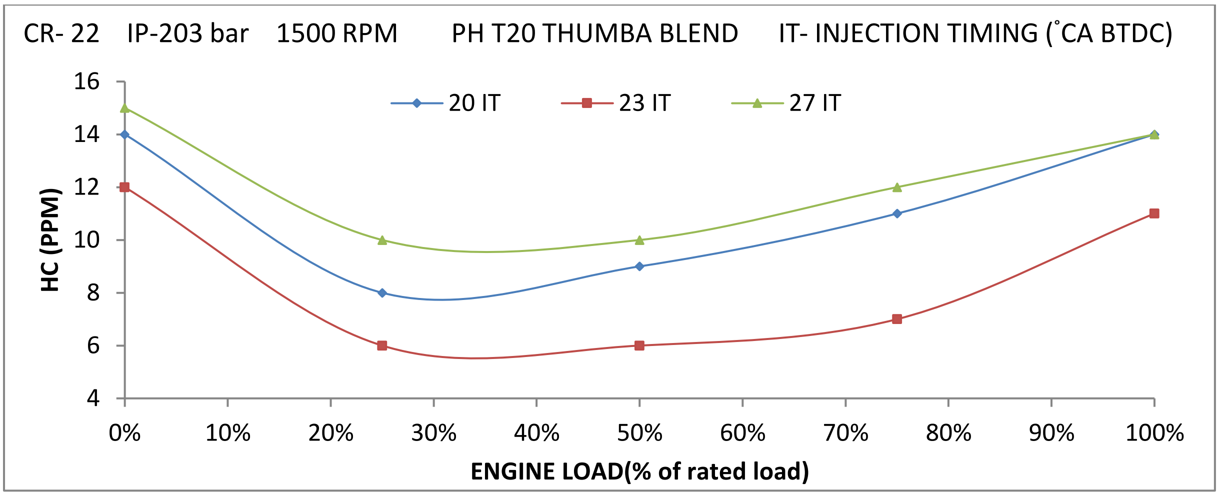


Figure: 4.67 Variation of HC emissions with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

NOx emissions in the exhaust gases depended on combustion temperature, combustion duration, and presence of oxygen in the fuel. The variation of NOx emissions with the load at different injection timings is presented in Figure 4.68. NOx emissions were lowest at injection timing of 23°CA BTDC.

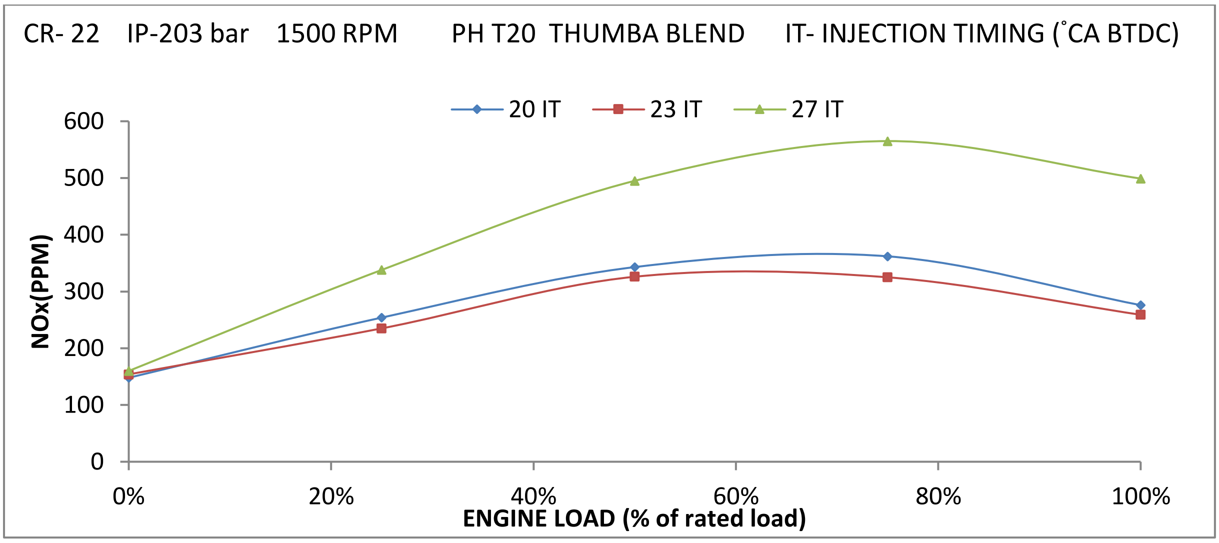


Figure: 4.68 Variation of NOx emissions with percentage increase in load at different injection timings for the preheated T20 Thumba oil blend

4.4.3.3 Combustion Studies

The combustion characteristics of engine operated with the preheated T20 Thumba oil diesel blend at different injection timings are summarized in this part of the analysis. The effect of injection timing on different engine combustion characteristics like cylinder pressure, the maximum rate of pressure rise, heat release rate, ignition delay, and other combustion parameters was observed. The variation of cylinder pressure with crank angle at different injection timings is presented in Figure 4.69. Cylinder pressure increased with increase in injection timing, and the angle of maximum cylinder pressure reduced with the rise of injection timings.

The maximum rate of pressure rise for different injection timings is presented in Figure 4.70. The maximum rate of pressure rise was highest for injection timing of 23°CA BTDC owing to better combustion and intermixing of fuel and air. The net heat release rate was found to be maximum and ignition delay was minimum for the injection timing of 23°CA BTDC. Other combustion parameters for different injection timings are presented in Table 4.9.

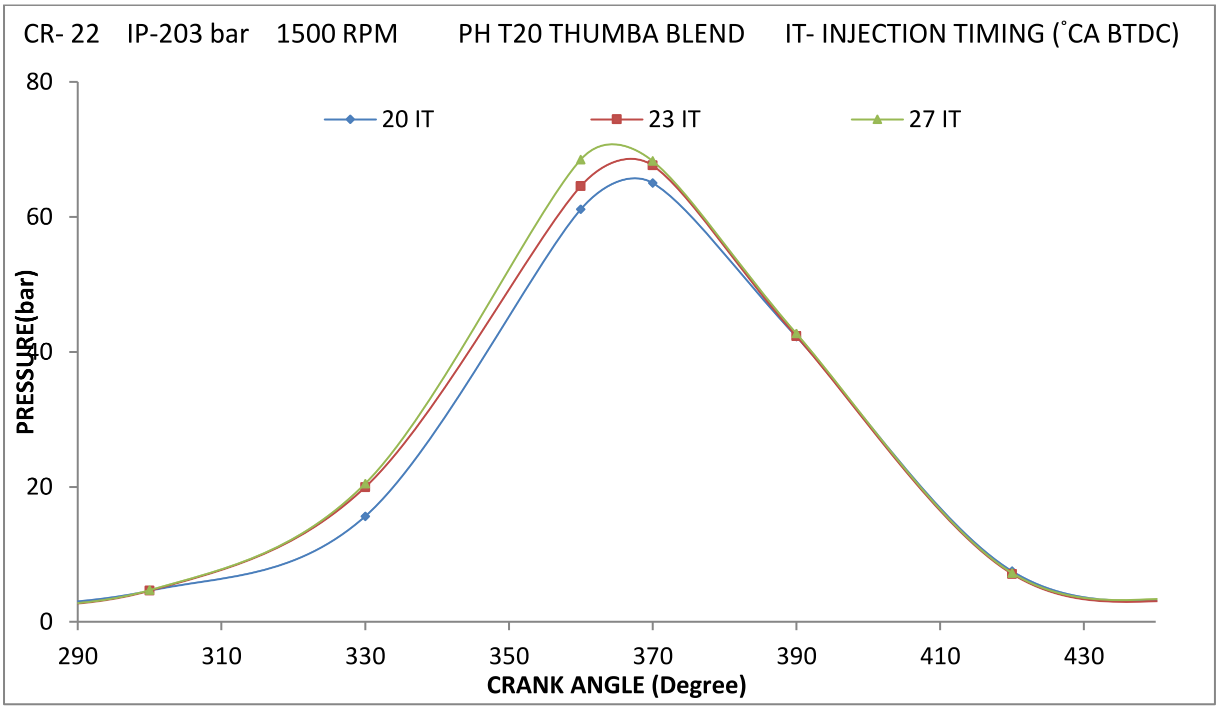


Figure 4.69 Variation of cylinder pressure with crank angle at different injection timings for the preheated T20 Thumba oil blend

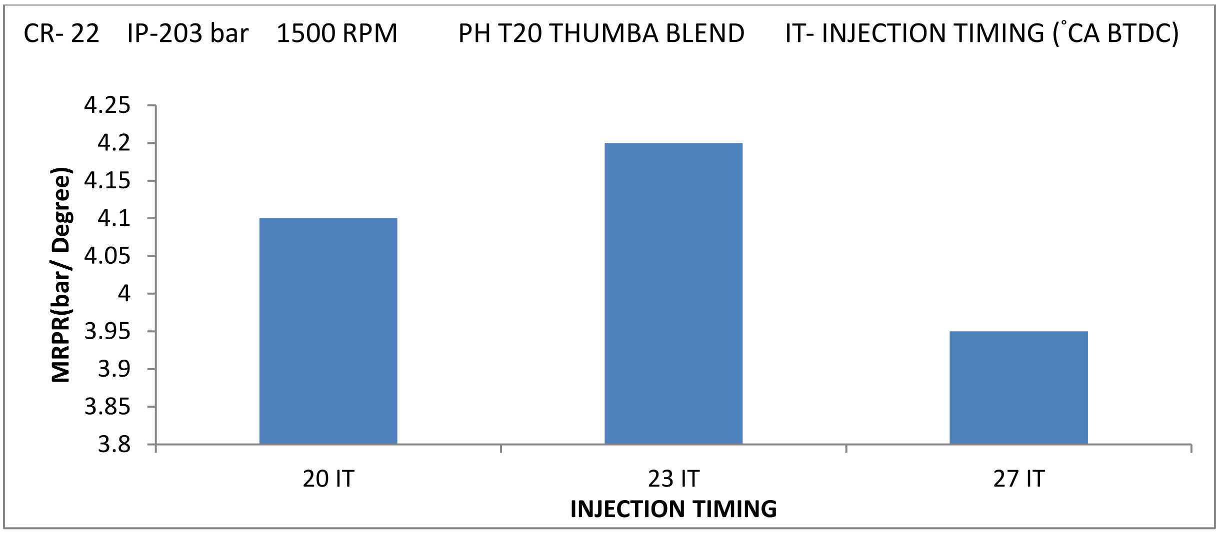


Figure 4.70 Variation of the maximum rate of pressure rise at different injection timings for the preheated T20 Thumba oil blend

Table 4.10 Combustion parameters for different injection timings fuelled with Preheated T20 Thumba oil blend

Combustion Parameters	IT-20 (˚CA BTDC)	IT-23 (˚CA BTDC)	IT-27 (˚CA BTDC)
Maximum Rate of Pressure Rise (bar/˚CA)	4.1	4.2	3.95
Maximum Net Heat Release (MJ)	34.7	35.8	35.12
Ignition Delay (˚CA BTDC)	13.6	13.5	13.8
Start of Combustion (˚CA)	6.4	9.5	13.2
Maximum Pressure (bar)	68.2	72.43	73.4
Angle for Maximum Pressure (˚CA ATDC)	9.5	9	8.5

4.4.3.4 Summary of Results

The engine parameters like compression ratio of 22, an injection pressure of 203 bar, and injection timings of 23 °CA BTDC were found to be the optimized parameters according to the results of experimentation conducted using the preheated T20 Thumba oil diesel blend as fuel for engine. Upon using the preheated T20 Thumba oil diesel blend, it was observed that there was least smoke opacity, CO emissions, and HC emissions along with the highest CO_2 emissions in the exhaust emissions at optimum engine parameters, i.e., injector pressure of 203 bars, injections timing of 23°CA BTDC, and compression ratio of 22. The optimized parameters are presented in Table 4.11.

Table: 4.11 Optimized parameters engine operated with preheated T20 Thumba oil blend

S. No.	Engine parameter	Optimum value
1	Compression Ratio	22
2	Injector Pressure (bars)	203
3	Injection Timing (°CA BTDC)	23
4	Load	100%

4.5 ENGINE CHARACTERISTICS FUELED WITH UNHEATED T20 THUMBA OIL BLEND

The present experimental study was extended to unheated optimize Thumba oil diesel blend to observe the effect of preheating on performance, combustion, and emission characteristics. The operating conditions of engine were same for the unheated T20 Thumba oil-diesel blend as for the preheated T20 Thumba oil diesel blend with diesel.

4.5.1 Performance Studies

A graph plotted between BTE and load for the unheated T20 Thumba oil diesel blend is shown in Figure 4.71. BTE increased with an increase in the percentage of load.

The variation of BSFC with the load for unheated T20 Thumba oil diesel blend is shown in Figure 4.72. BSFC decreased with increase in the percentage increase in load.

A graph plotted to observe the variation of exhaust gas temperature with the load is shown in Figure 4.73. Exhaust gas temperature increased with increase in the percentage of load.

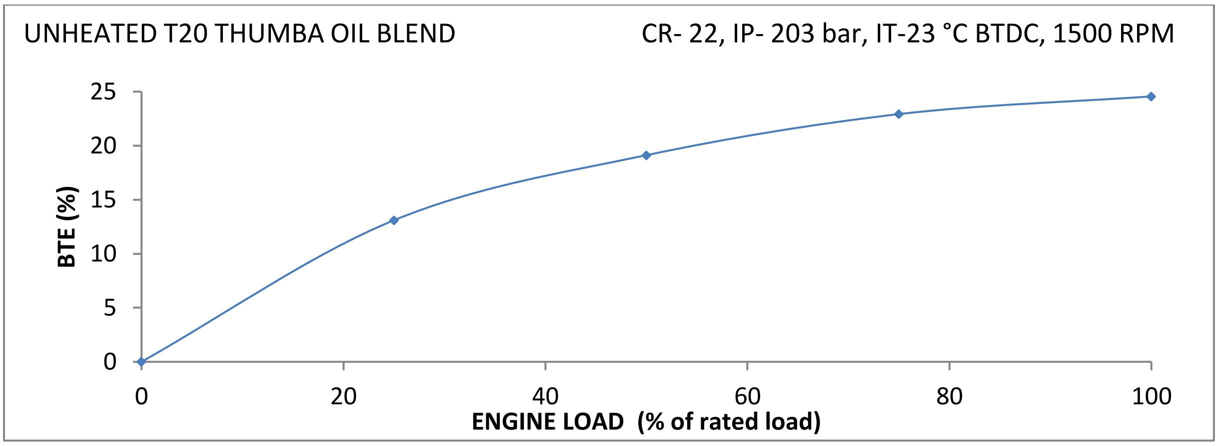


Figure 4.71Variation of brake thermal efficiency with percentage increase in load for unheated T20 Thumba oil blend

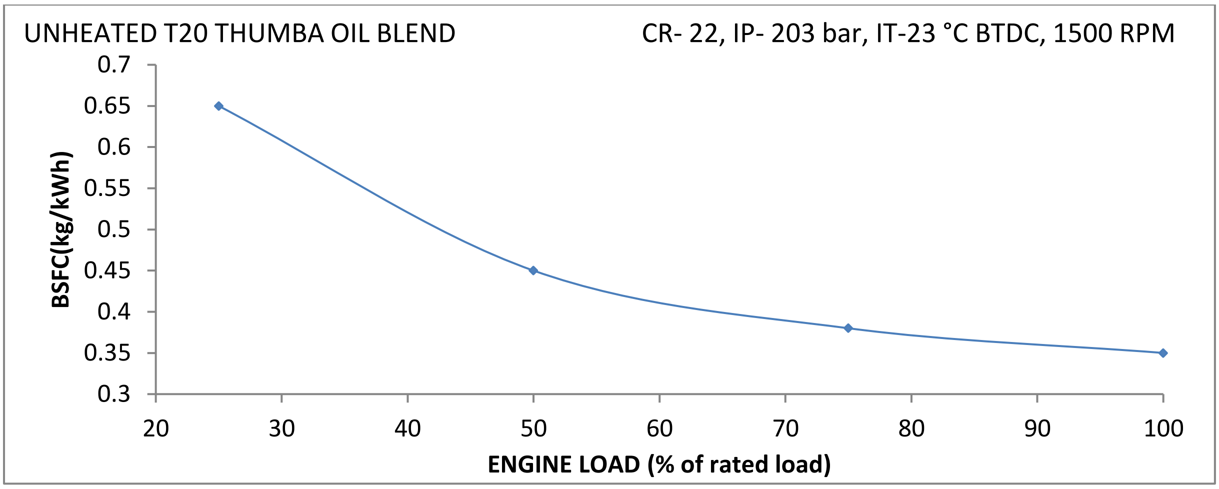
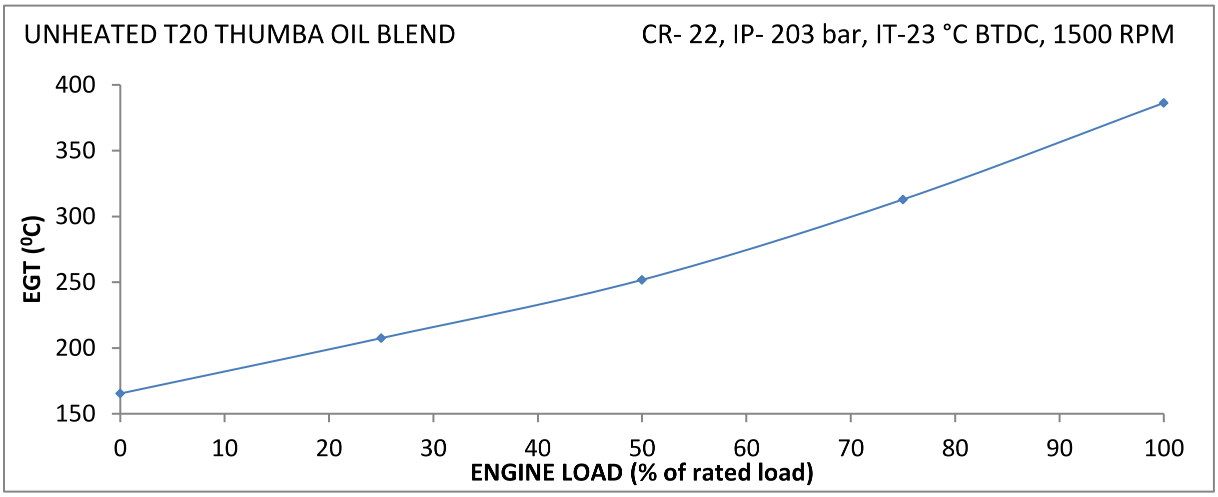


Figure 4.72Variation of brake specific fuel consumption with percentage increase in load for



unheated T20 Thumba oil blend

Figure 4.73 Variation of exhaust gas temperature with percentage increase in load for unheated T20 Thumba oil blend

4.5.2 Emission Studies

The variation of smoke opacity with the load for unheated T20 Thumba oil diesel blend is plotted in Figure 4.74. Smoke emissions increased with increase in the percentage of load. The variation of CO emissions with the load for unheated T20 Thumba oil blend with diesel is shown in Figure 4.75. CO emissions initially decreased with an increase in the percentage of load, but gradually increased later. The variation of CO_2 emissions with the load for unheated T20 Thumba oil blend is shown in Figure4.76. CO_2 emissions increased with increase in the percentage of load, but these emissions tended to decrease at higher percentage of load.

HC emissions were present in the exhaust gases owing to incomplete combustion, and the variation of HC emission with the load is shown in Figure 4.77. NOx emissions increased with increase in the load, but at the higher percentage of load, the emissions decreased with increase in the percentage of load, as given in Figure 4.78.

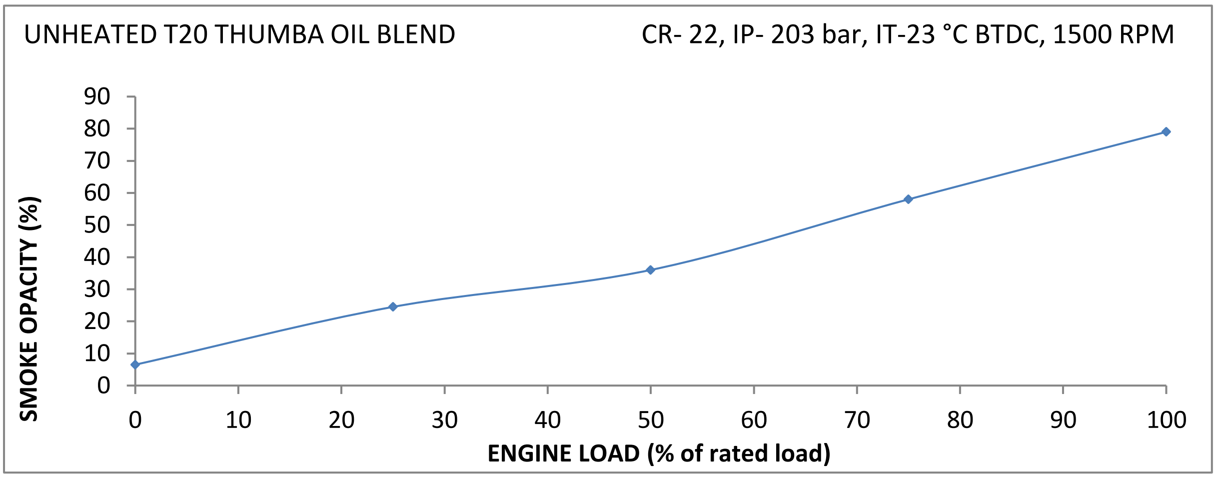


Figure 4.74Variation of smoke opacity with percentage increase in load for unheated T20 Thumba oil blend

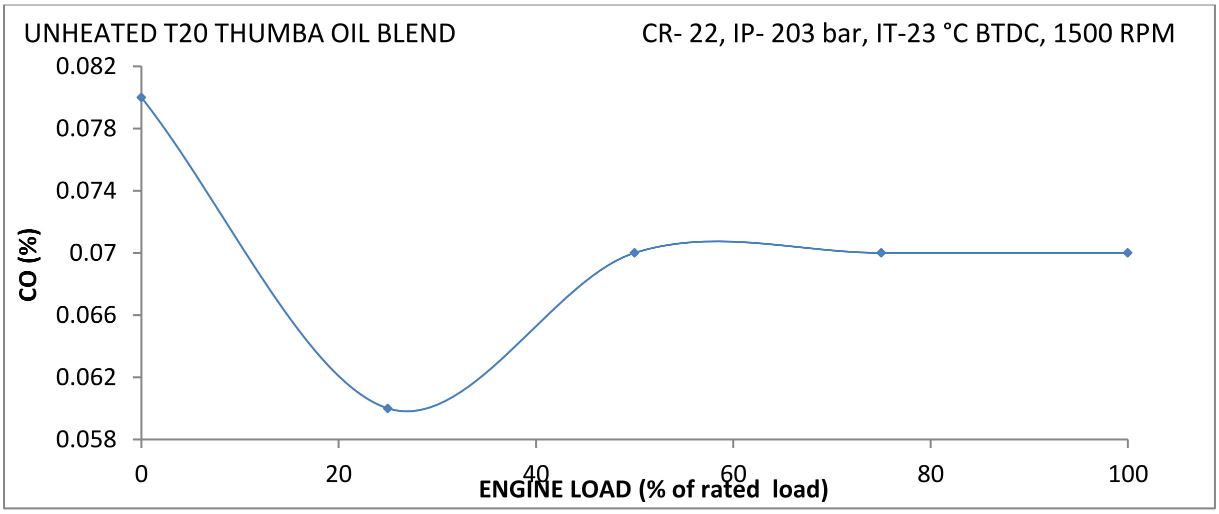


Figure 4.75 Variation of CO emissions with percentage increase in load for unheated T20 Thumba oil blend

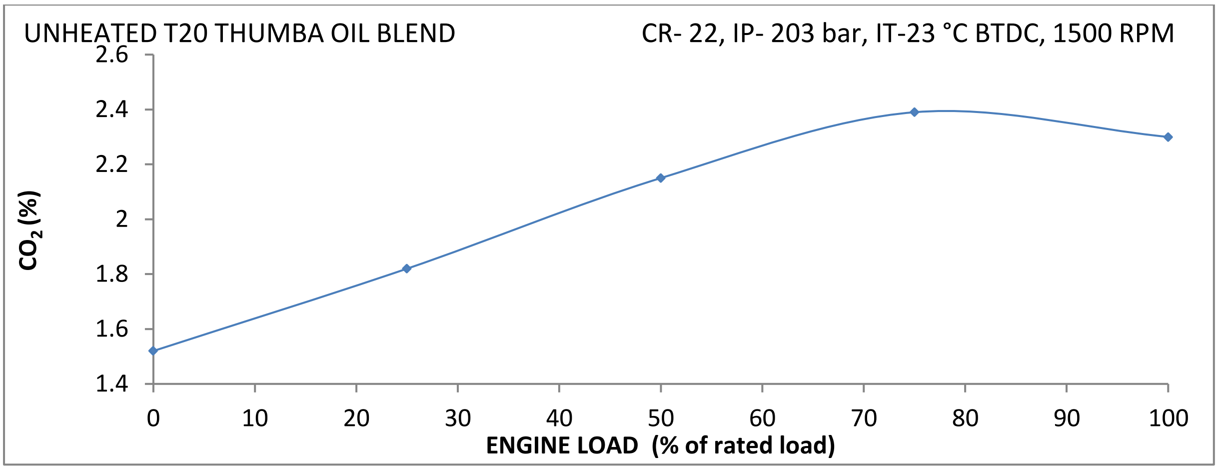


Figure 4.76 Variation of CO_2 emissions with percentage increase in load for unheated T20 Thumba oil blend

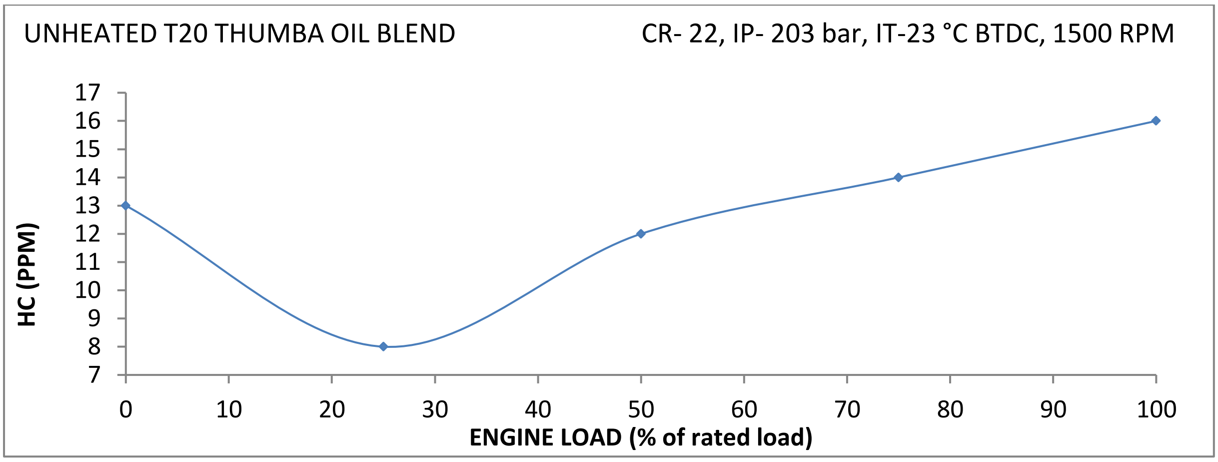


Figure 4.77 Variation of HC emissions with percentage increase in load for unheated T20 Thumba oil blend

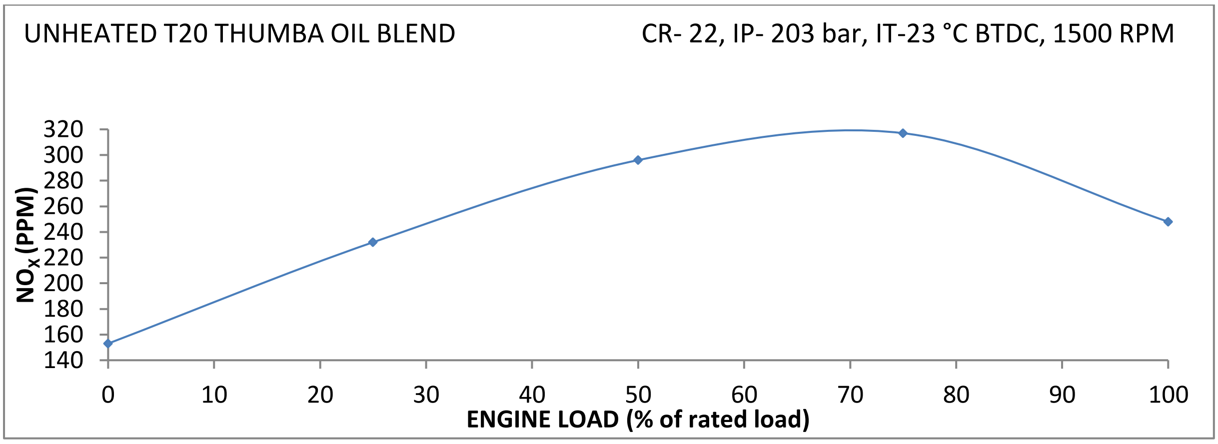


Figure 4.78 Variation of NOx emissions with percentage increase in load for unheated T20 Thumba oil blend

4.5.3 Combustion Studies

The combustion parameters of engine were observed under the same operating conditions. The engine parameters were the compression ratio of 22, an injection pressure of 203 bar, and the injection timing of 23°CA BTDC. The variation of cylinder pressure with the crank angle for the unheated Thumba oil diesel blend is shown in Figure 4.79. Other obtained combustion parameters are also given in Table 4.12.

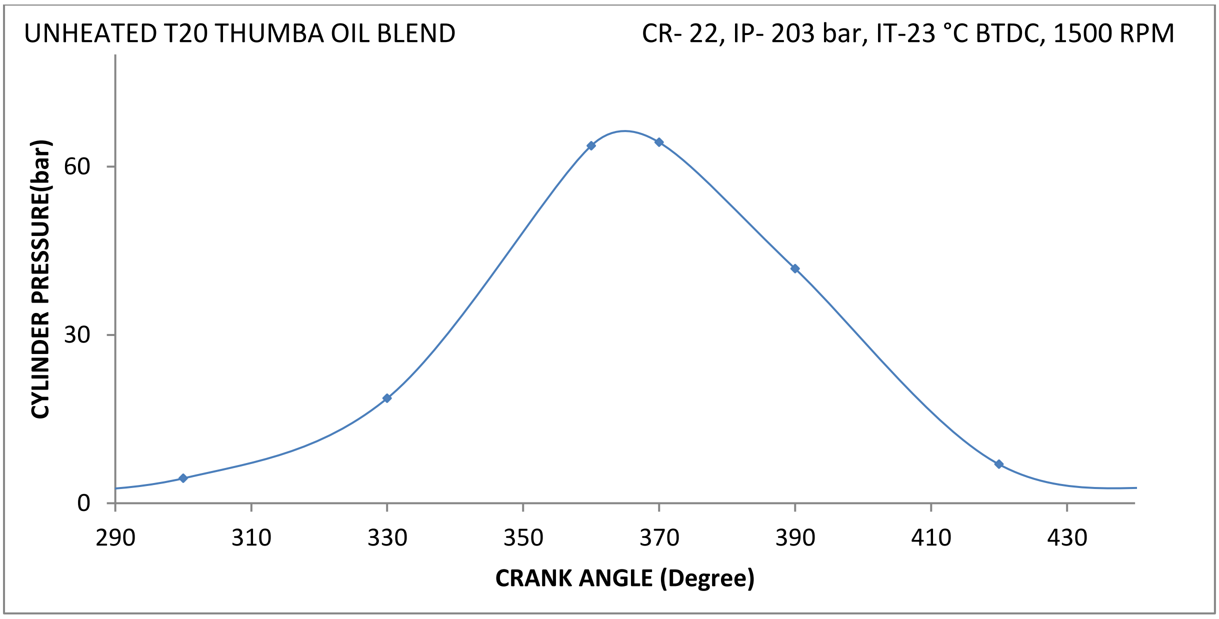


Figure 4.79 Variation of cylinder pressure with the crank angle for unheated T20 Thumba oil blend

Table 4.12 Combustion parameters for unheated T20 Thumba oil blend

Combustion Parameters	**UTODB**
Maximum Rate of Pressure Rise (bar/˚CA)	3.8
Maximum Net Heat Release (MJ)	31.2
Ignition Delay (˚CA BTDC)	13.7
Start of Combustion (˚CA)	8.5
Maximum Pressure (bar)	69.78
Angle for Maximum Pressure (˚CA ATDC)	8.5

4.6 COMPARATIVE ANALYSIS OF ENGINE CHARACTERISTICS

A comparative analysis of engine performance, combustion, and emission characteristics operated with diesel, preheated and unheated Thumba oil-diesel blend is presented in this section.

4.6.1 Comparison of Performance Studies

A comparative analysis of the engine performance fuelled with diesel, preheated and unheated T20 Thumba oil-diesel blend was carried out to observe the variation of engine performance upon using the preheated and unheated T20 Thumba oil-diesel blend in comparison to that of diesel.

The comparison of BTE for these fuels at different load is shown in Figure 4.80. BTE for diesel was higher than both the unheated and preheated optimize Thumba oil-diesel blends for all the load conditions, which was probably owing to the higher viscosity of both the preheated and unheated T20 Thumba oil diesel blend. BTE for diesel, preheated and unheated T20 Thumba oil-diesel blends were 26.5%, 25.93%, and 24.66%, respectively. Preheating improved the thermal efficiency of T20 Thumba oil blend by 1.27% because preheating reduced the viscosity of the Thumba oil blend.

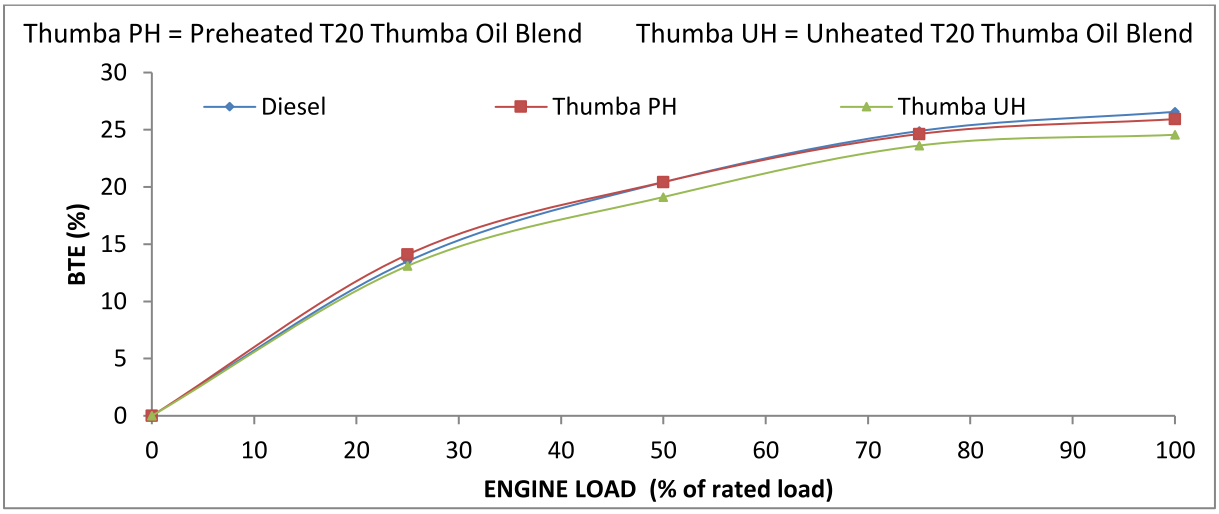


Figure 4.80 Comparison of brake thermal efficiency with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

The comparison of BSFC for diesel, preheated and unheated T20 Thumba oil blend with diesel is shown in Figure 4.81. BSFC decreased with percentage increase in load, and it was found to be lowest at full load. BSFC for diesel and preheated T20Thumba oil diesel blend at full load condition was 0.33 kg/kWh and for unheated T20 Thumba oil diesel blend was 0.35 kg/kWh. The preheating of the blended oil saved 0.02 kg/kWh of fuel because preheating enhanced the atomization and mixing of fuel.

The variation of EGT with the load for diesel, preheated and unheated T20 Thumba oil diesel blend is shown in Figure 4.82. Exhaust gas temperature increased with percentage increase in load for all the fuels and it was found to be less for the preheated and unheated T20 Thumba oil-diesel blends in comparison to diesel, except for full load condition. The figure shows that the exhaust gas temperature for diesel, T20 preheated and unheated Thumba oil diesel blend is very close.

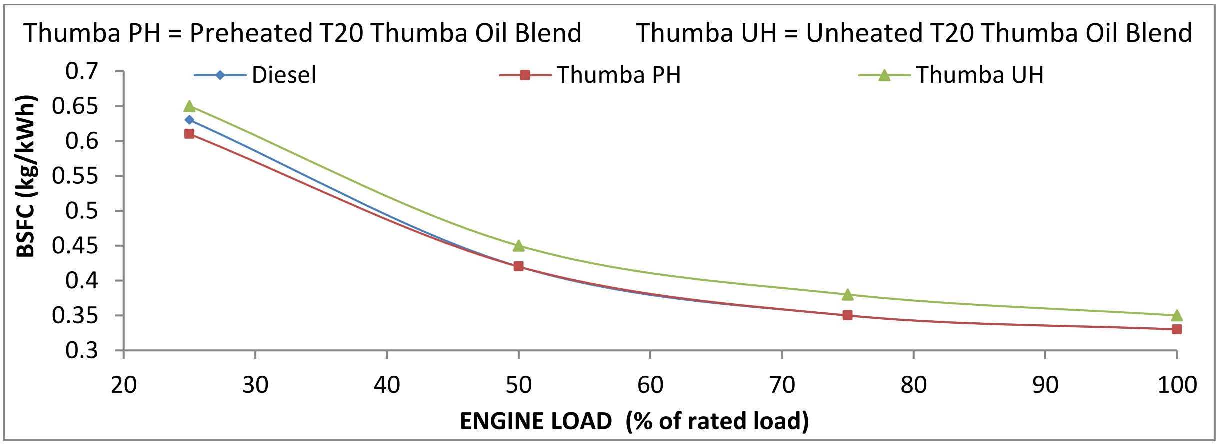


Figure 4.81 Comparison of brake specific fuel consumption with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

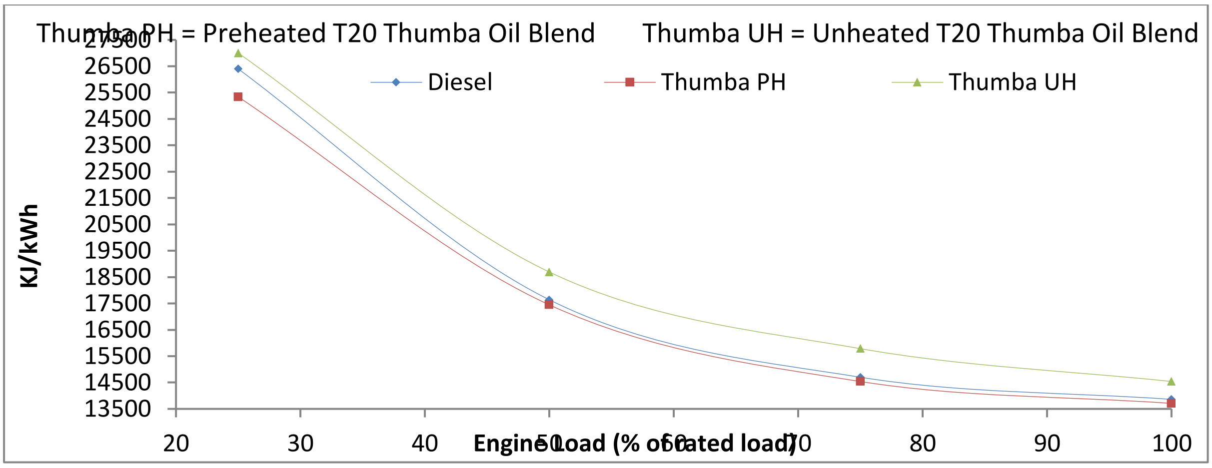


Figure 4.81(a) Comparison of brake specific energy consumption with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

The comparison of BSEC for diesel, preheated and unheated T20 Thumba oil blend with diesel is shown in Figure 4.81(a). BSEC decreases with percentage increase in load, and it is found to be lowest at full load. It has been observed from the plot that BSEC is found minimum at all the load for preheated T20Thumba oil diesel blend and BSEC is found maximum for unheated T20Thumba oil diesel blend.

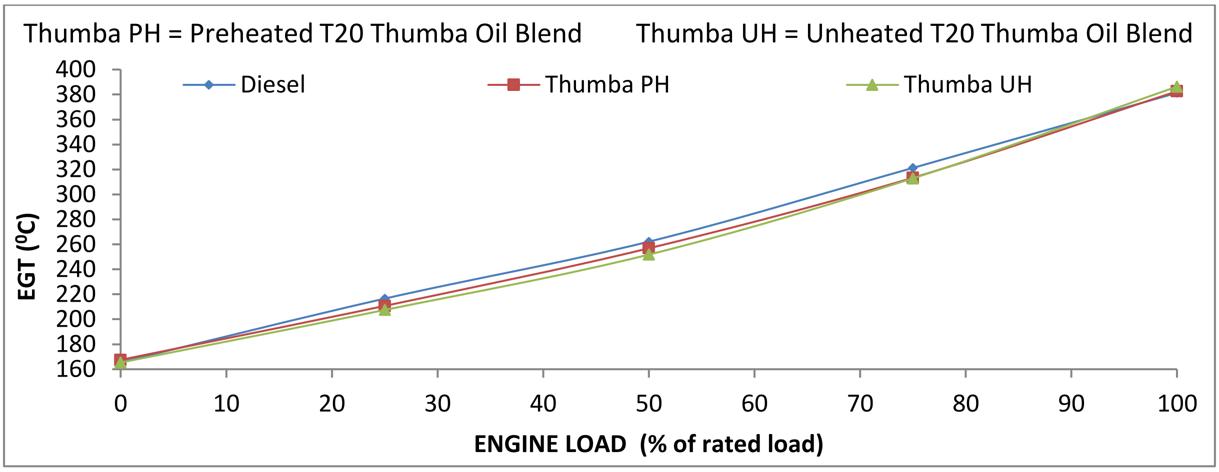


Figure 4.82 Comparison of exhaust gas temperature with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

4.6.2 Comparison of Emission Studies

In this section, the emission parameters such as smoke opacity, CO, CO_2, HC, and NOx are compared for diesel, unheated and preheated optimized Thumba oil diesel blend (T20).

The comparison of smoke density with diesel, preheated and unheated T20 Thumba oil blend is shown in Figure 4.83, which shows that smoke emissions increased with increases in load. However, T20 preheated Thumba oil diesel blend emitted less smoke in comparison to diesel and unheated Thumba oil T20 blend. The preheating of the blended oil resulted in the reduction of smoke emission by 3% due to better combustion. Both the preheated and unheated Thumba oil emitted less smoke in comparison to diesel possibly due to the presence of extra oxygen in Thumba vegetable oil and better combustion to diesel.

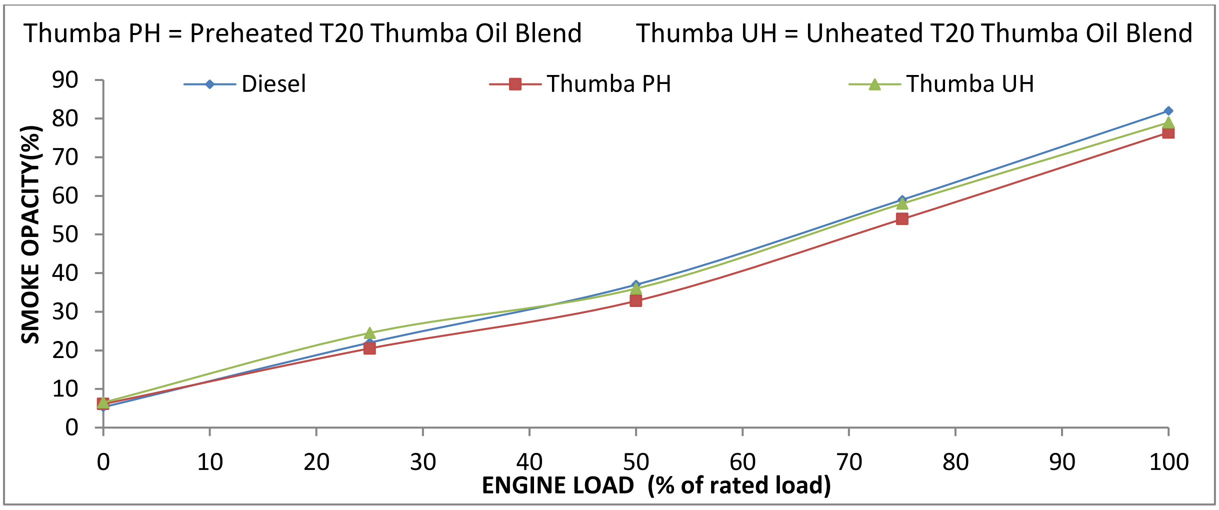


Figure 4.83 Comparison of smoke opacity with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

The comparison of CO emission for diesel and preheated and unheated T20 blend with diesel is shown in Figure 4.84. CO emission for the preheated T20 (0.05%) Thumba oil-diesel blend is lower than both diesel (0.06%) and unheated T20 Thumba oil-diesel blend (0.07%). It shows better combustion for the preheated T20 Thumba oil-diesel blend than diesel and unheated Thumba oil-diesel blend.

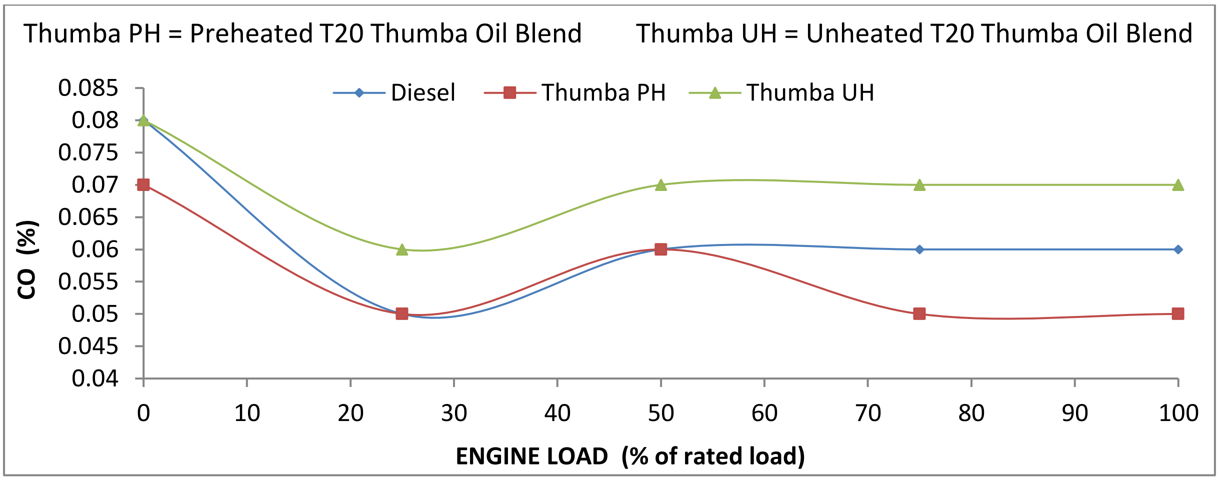


Figure 4.84 Comparison of CO emissions with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

The comparison of CO_2 emissions for diesel, preheated, and unheated Thumba oil-blend is shown in Figure 4.85. It was observed that CO_2 emissions were higher for the preheated T20 Thumba oil-diesel blend (2.38%) than the unheated blended oil (2.3%), and it was even lower than diesel (2.6%).

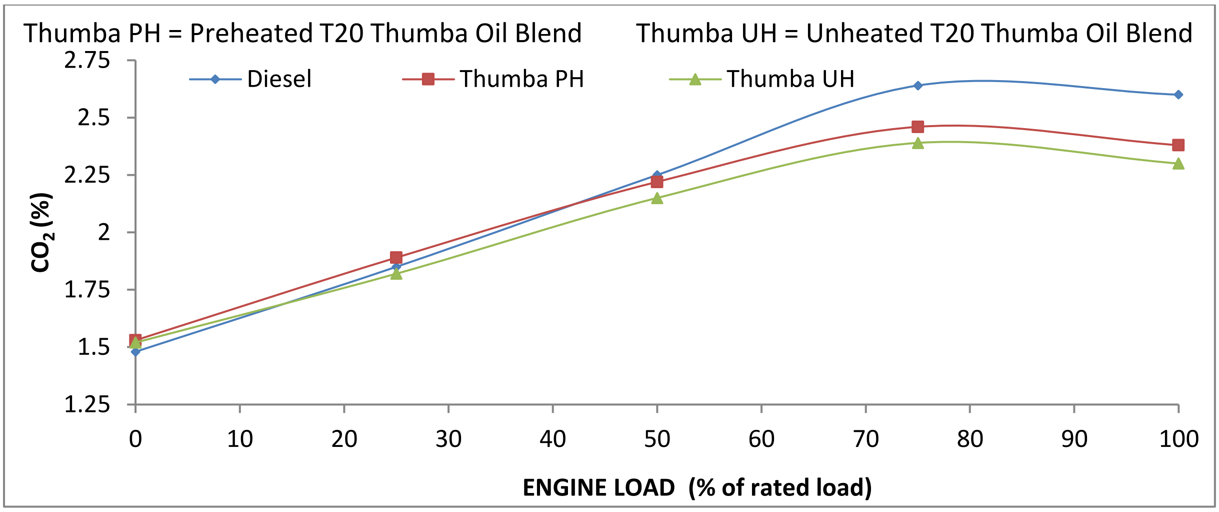


Figure 4.85Comparison of CO_2 emissions with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

HC emissions were measured in parts per million in this study. HC emissions decreased initially and then subsequently increased. The comparison of HC emissions for diesel, preheated and unheated T20 Thumba oil blend is shown in Figure 4.86. HC emissions from the preheated T20 Thumba oil blend (11 PPM) were lower than the unheated Thumba oil blend (16 PPM) and diesel (20 PPM).

Generally, NOx emissions increased with the use of vegetable oils because of the presence of additional oxygen, but upon using the Thumba oil-diesel blend, the emissions decreased possibly owing to less combustion duration compared to diesel. Unheated T20 Thumba oil-diesel blend emitted less NOx as compared to diesel and preheated T20 Thumba oil diesel blend for the entire load conditions as shown in Figure 4.87. However, NOx emissions were higher for the preheated Thumba T20 blend than both diesel and unheated same blend for 50-75% load. NOx emissions in preheated Thumba oil blend were higher than the unheated Thumba oil blend because of higher combustion temperature as more NOx were formed.

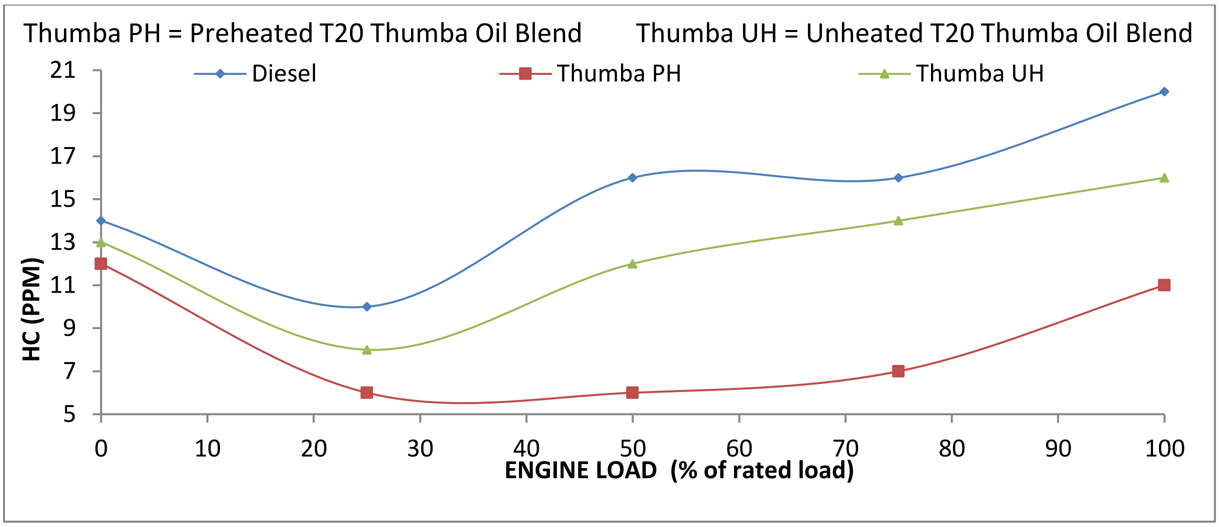


Figure 4.86 Comparison of HC emissions with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

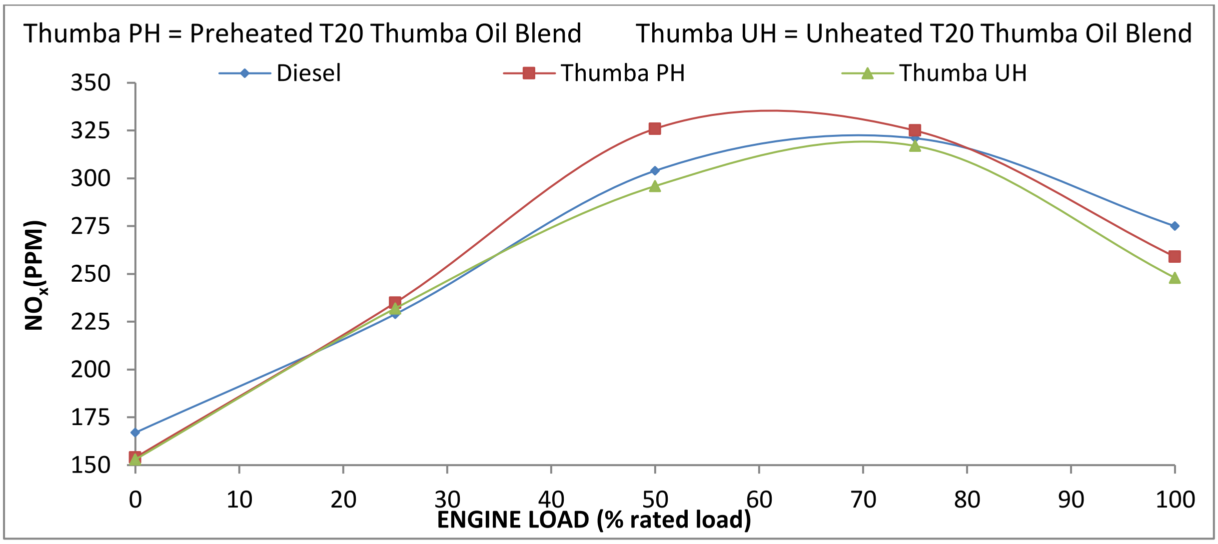


Figure 4.87 Comparison of NOx emissions with percentage increase in load for diesel, unheated and preheated T20 Thumba oil blend

4.6.3 Comparison of Combustion Studies

The combustion characteristics of diesel, preheated and unheated T20 Thumba oil blend with diesel have been compared. Higher cylinder pressure was observed for the diesel than preheated T20 Thumba oil diesel blend and unheated Thumba T20 blend with diesel The comparison of cylinder pressure with the crank angle for these fuels is shown in Figure 4.88.

The rate of pressure rise was maximum for diesel but minimum for unheated Thumba oil diesel blend. The rate of pressure rise for diesel, preheated and unheated Thumba oil-diesel blend is shown in Figure 4.89. Other combustion characteristics are also given in Table 4.13.

The heat release rate for diesel fuel was found to be higher than both unheated and preheated Thumba oil blends with diesel.

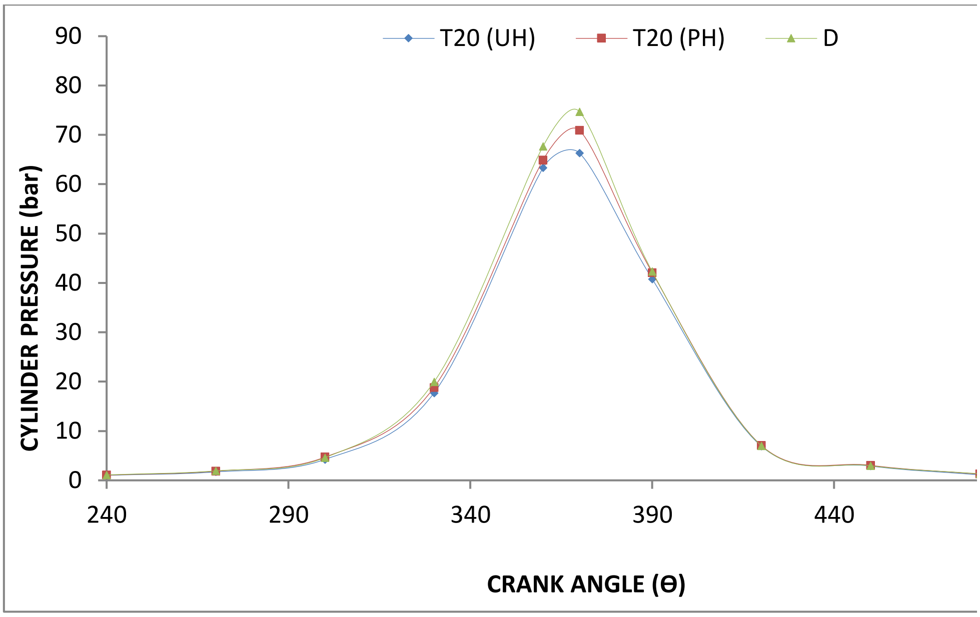


Figure 4.88 Comparison of cylinder pressure with the crank angle for diesel, unheated and preheated T20 Thumba oil blend

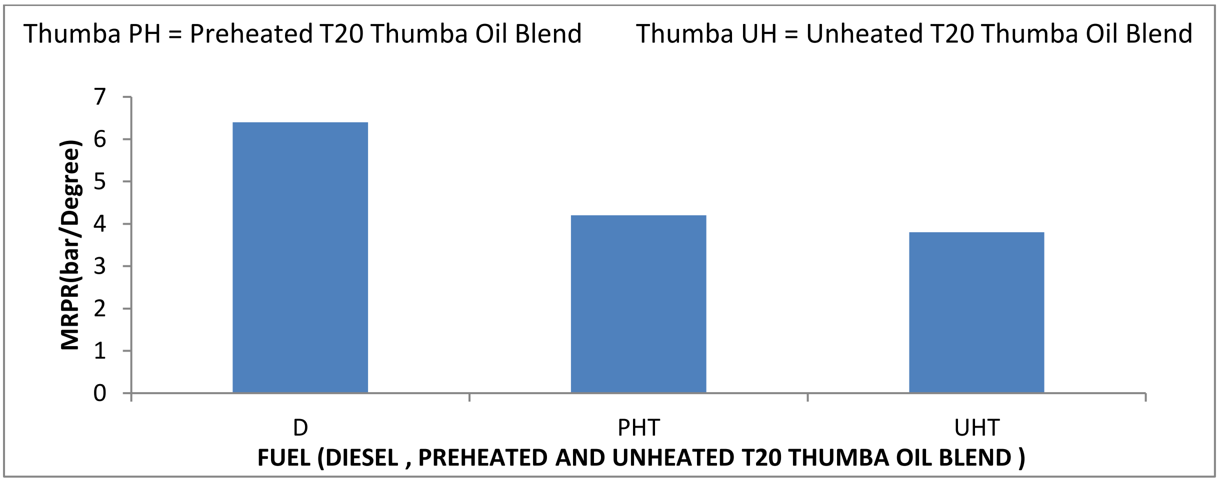


Figure 4.89 Comparison of maximum rate of pressure rise for diesel, unheated and preheated T20 Thumba oil blend

Table 4.13 Combustion parameters for diesel, unheated and preheated T20 Thumba oil blend.

Combustion Parameters	D	PHT	UHT
Maximum Rate of Pressure Rise (bar/°CA)	6.4	4.2	3.8

Maximum Net Heat Release (MJ)	46.4	35.8	31.2
Ignition Delay (°CA BTDC)	13.9	13.5	13.7
Start of Combustion (°CA)	9	9.5	8.5
Maximum Pressure (bar)	76.11	72.43	69.78
Angle for Maximum Pressure (°CA ATDC)	9	9	8.5

4.6.4 Summary of Comparative Results

In this part of the analysis, a summary of the comparative study made for diesel, and the preheated and unheated T20 Thumba oil diesel blends for their optimize condition has been presented. The comparative analysis of all the characteristics for diesel, preheated and unheated T20 Thumba oil-diesel blend is given in Table 4.14. BTE for diesel (26.57%) was found to be slightly higher than the preheated T20Thumba oil diesel blend (25.93%), and BTE for the preheated Thumba oil-diesel blends was marginally higher than unheated Thumba oil diesel blend (24.66%). The emissions were slightly lower for the preheated Thumba oil diesel blend than diesel and unheated Thumba oil diesel blend.

Table 4.14 Summary of comparative results for diesel, preheated and unheated T20 Thumba oil diesel blend at full load and their optimized conditions

Parameters	Diesel	PTODB (T20)	UTODB (T20)
BTE (%)	26.57	25.93	24.66
BSFC (kg/kWh)	0.33	0.33	0.35
BSEC (kJ/ kWh)	13860	13708	14539
EGT (°C)	381.34	382.6	386.88
SMOKE OPACITY (%)	82	76.4	79
CO (%)	0.07	0.05	0.06
CO_2 (%)	2.6	2.38	2.3
HC (PPM)	21	16	11
NO_x (PPM)	275	269	248
Maximum Pressure (bar /°CA)	76.11	72.43	69.78
Maximum Rate of Pressure Rise(bar)	6.4	4.2	3.8
Heat Release Rate (MJ/kg)	46.4	35.8	31.2
Engine optimized condition	CR:21	CR:22	Results are obtained at preheated Thumba oil optimized condition.
	IP: 203 Bar	IP: 203 Bar	
	IT: 23°CA BTDC	IT: 23°CA BTDC	

CHAPTER 5

RESULT AND DISCUSSION (PART 2)

5.1 INTRODUCTION

The short run studies of CI engines using vegetable oils as fuel have been found to be very successful, but the long term use of these oils starts generating problems such as severe engine deposits, piston ring sticking, injector coking, gum formation and lubricating oil thickening. Another issue of using vegetable oils is the contamination of lubricating oil that occurs due to escaping of gases through the clearance into the engine crankcase as the sticking rings fail to seal adequately. The contaminated lubricating oil forms a tough rubber-like coating on the engine parts such as walls of the crankcase, fuel pump, camshaft, and pushrods [87].

The carbon deposits are generally attributed to the large molecular size and the high viscosity of the medium-chain and long-chain triglycerides, which are the constituents of most commercial vegetable oils [88]. Therefore, it becomes imperative to have long-run testing of every vegetable oils before their regular use in diesel engines. The results of the present long run study of the research engine using the preheated T20 Thumba oil blend with diesel have been summarized, which also comprises wear, lube oil, and visual analysis.

5.2 WEAR ANALYSIS OF ENGINE INTERNAL PARTS

Sliding contact between metallic components of any mechanical system is always accompanied with wear, which results in generating minute particles of metal. In diesel engines, the components usually subjected to wear are piston, piston rings, cylinder liner, connecting rod, valves, tappet, and valve guides. For measurement of wear in a diesel engine, the engine has to be dismantled entirely before and after the test. The dimensions of the internal parts have also been measured. The difference in dimensions before and after the test has been counted as wear in the engine parts.

Wear measurement of following engine parts are given in Tables 5.1 to 5.7.

(a) Cylinder/ Cylinder liner bore

(b) Piston

(c) Piston rings

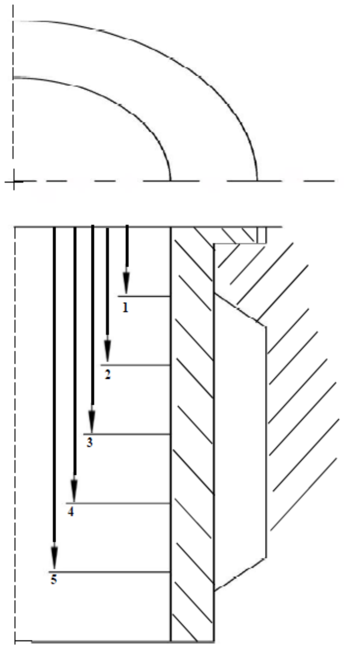
(d) Gudgeon pin

(e) Connecting rod

(f) Valves (inlet and exhaust)

(g) Valve guides Table

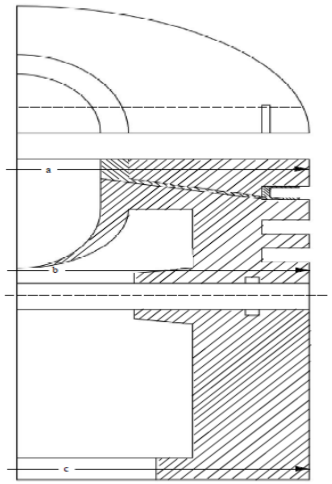
Table 5.1 Wear Measurements of Cylinder Bore / Cylinder Liner



Dimensions(mm) at Positions Indicated in Fig.															
	Before Endurance Test					After Endurance Test					Wear				
Position	1	2	3	4	5	1	2	3	4	5	1	2	3	4	5
Along crankshaft axis	87.5122	87.5214	87.517	87.521	87.515	87.43	87.441	87.46	87.43	87.445	0.0822	0.0804	0.057	0.091	0.07
Perpendicular to the Crank-shaft Axis	80.04	80.04	80.06	80.05	80.06	80.04	80.07	80.06	80.13	80.14	0.00	0.03	0.00	0.08	0.08

Surface Condition (Specify) (Add or Cross Out)	Erosion, Burning, Warping, etc.

Table 5.2 Wear Measurements of Piston

	Diameter of Piston Position		Before Endurance Test		After Endurance Test		Wear	
			ϕX	ϕY	ϕX	ϕY	ϕX	ϕY
		1	87.0739	87.2438	87.0601	87.2136	0.0138	0.0302
		2	87.3376	87.282	87.316	87.280	0.0216	0.02
Dimensions (mm)								

Surface Condition	Crown	Burn and spot

(Specify)	Top Land	Thick deposit, black and burn

Table 5.3 Wear Measurements for Piston Rings

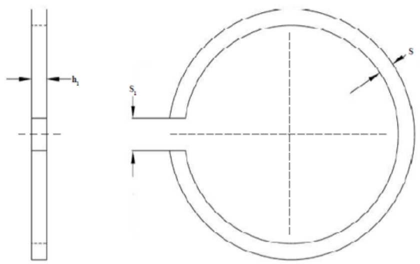
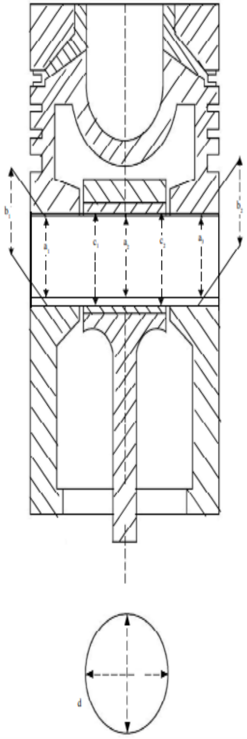
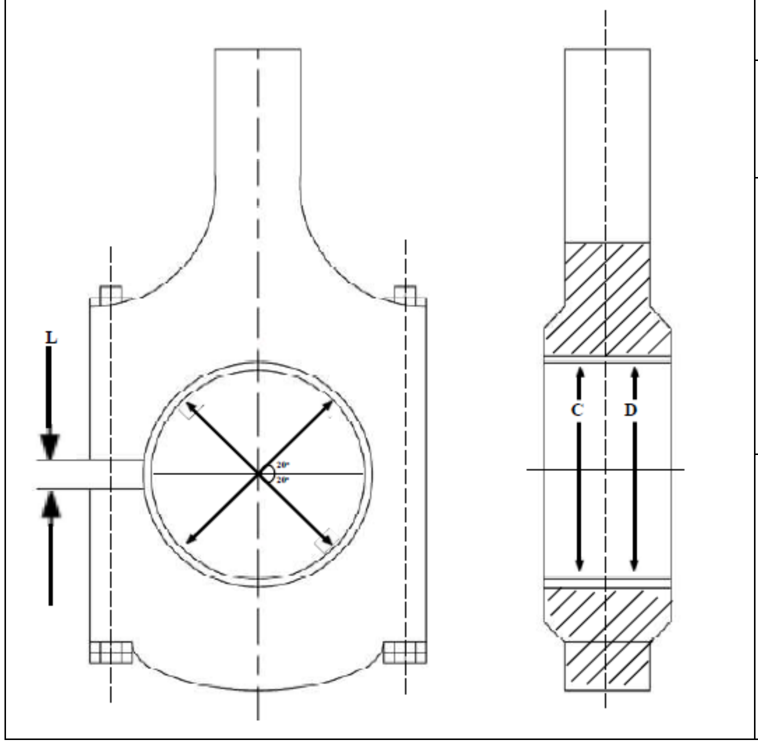
	Dimensions (mm) of Piston Rings									
	Before Endurance Test				After Endurance Test			Wear		
	Ring No.	Radial Wall Thickness a_1	Ring closed Gap, s_1 when in the Nominal Bore Specified in IS:3511	Axial width h_2	Radial Wall Thickness a_1	Ring closed Gap, s_1 when in the Nominal Bore Specified in IS:3511	Axial width h_2	Radial Wall Thickness a_1	Ring closed Gap, s_1 when in the Nominal Bore Specified in IS:3511	Axial width h_2
			Compression			Compression			Compression	
	1	3.37	0.35	2.385	3.362	0.5715	2.38	0.008	0.221	0.005
	2	3.37	0.35	2.385	3.3511	0.612	2.367	0.0189	0.262	0.018
	3	3.37	0.35	2.385	3.3486	0.623	2.332	0.0214	0.0237	0.053
	4	3.15	0.25	4.762	3.1218	0.408	4.73	0.0282	0.158	0.032

Table 5.4 Wear Measurements for Gudgeon pin, pin bore and small end bush of connecting rod

	Before Endurance Test							After Endurance Test							Wear						
	Gudgeon pin bore in piston		Gudgeon pin diameter			Small end bush		Gudgeon pin bore in piston		Gudgeon pin diameter			Small end bush		Gudgeon pin bore in piston		Gudgeon pin diameter			Small end bush	
		φb1	φb2	φa1	φa2	φa3	φc1	φc2	φb1	φb2	φa1	φa2	φa3	φc1	φc2	φb1	φb2	φa1	φa2	φa3	φc1	φc2
	X	29.9947	29.9955	29.9911	29.9909	29.9912	30.0606	30.0573	29.99	29.992	29.9901	29.99	29.9901	30.051	30.05	0.0046	0.0054	0.001	0.0009	0.0011	0.0096	0.0073
	Y	29.9939	29.9951	29.9909	29.9909	29.9911	30.0604	30.0578	29.9922	29.9927	29.99	29.99	29.99	30.0516	30.048	0.0037	0.0034	0.009	0.009	0.0011	0.0088	0.0098

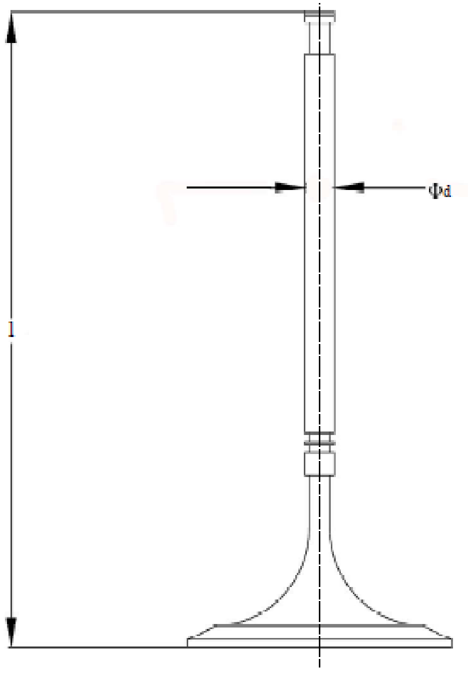
Dimensions (mm)

Table 5.5 Wear Measurements of Connecting Rod Bearing Bore



Dimensions (mm)				
		Before Endurance Test	After Endurance Test	Wear
ϕC	ϕl	60.4373	60.4130	0.0243
	ϕm	60.4207	60.3982	0.0225
	ϕn	60.4256	60.3573	0.0683
ϕD	ϕl	60.4367	60.41	0.0267
	ϕm	60.441	60.3371	0.1039
	ϕn	60.4349	60.3600	0.0749

Table 5.6 Wear Measurements for Inlet and Exhaust Valves

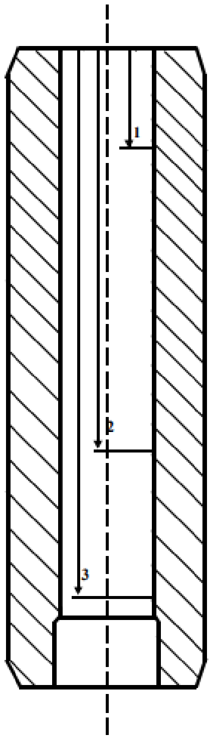


Dimensions(mm)				
		Before Endurance Test	After Endurance Test	Wear
Stem Diameter 'd'	Inlet valve	7.8813	7.8701	0.112
	Exhaust valve	7.8817	7.8708	0.109
Length 'l'	Inlet valve	113.67	113.70	-0.3
	Exhaust valve	113.69	113.71	-0.2

Surface Condition (Specify)		Valve Stem	Valve Face
	Inlet	Scratches	

	Exhaust	Scratches	Erosion, black

Table 5.7 Wear Measurements for Valve Guides

	Diameter (mm) at Positions Indicated in Fig.									
	Before Endurance Test				After Endurance Test			Wear		
	Position	1	2	3	1	2	3	1	2	3
	Inlet valve	7.951	7.9512	7.93	7.948	7.9496	7.912	0.003	0.0016	0.0018
	Exhaust valve	7.9611	7.9658	7.9618	7.9601	7.9643	7.959	0.001	0.0015	0.0028

5.2.1 **Summary of Wear Analysis Results**

Cylinder liner wear was found between 80 and 82 microns. Maximum piston wear occurred at the middle of the piston, i.e., 30 microns. Gudgeon pin wear took place from 1 micron to 11 microns throughout its length. The internal diameter of small and big end bearing increased from 7 to 9 microns and 24 to 26 microns, respectively. Wear in axial and radial thickness of the piston rings were minimum in chrome plated or first ring but were maximum in the third ring. A lot of variation was observed in the wear of suction and exhaust valve, which was 11 to 100 microns throughout their length. It is also noted that the wear of the different components during the test was found within the allowable limits. However, there was nominal pitting on the cylinder head, minor oval in cylinder liner, spots on the crown of the piston, and faint lines on gudgeon pin and valve stem.

5.3 ANALYSIS OF LUBRICATING OIL

The complete history of engine wear is depicted by used lubricating oil. Lubricating oil failure and engine wear are vital parameters in determining the operation life, profitability, and reliability of engines and machinery mechanisms. One way of determining the satisfactory performance of an engine and the efficiency of lubricating oil in a running engine is by monitoring the acceleration of wear metals in lubricating oils. In a lubricated diesel engine system, wear particles being washed away by lubricating oil, they remain suspended in the oil. By analyzing and examining the variation in the concentration of the metallic particles in the lubricating oil, it is possible to predict wear rate, wear source, and engine condition. The detail of the metals found in the used lubricating oil of the engine and their sources are given in Table 5.8.

Table 5.8 Metals found in used engine lubricating oils and their sources [119]

Metals	Possible Sources
aluminum	Piston, bearings and cylinders; dirt & dust contamination
Barium	Oil additive, Diesel fuels additive.
Boron	Cooling water conditioners
Calcium	Oil additive (major); dirt & dust contamination (minor)
Chromium	Rings, cylinder liners, plated rocker arms or crankshafts, cooling water conditioners
Copper	Bearings and bushings, air filter mesh
Iron	Engine parts
Lead	Gasoline antiknock additive (major gasoline engine), bearings
Magnesium	Oil additive (major); sea water contamination (minor)
Phosphorous	Oil additive
Silicon	Sand & dust contamination (major), wear of engine parts (minor)
Sodium	Sea water contaminants, cooling water conditioners, dust contaminants
Tin	Tin plated pistons, bearings
Zinc	Oil additives (major), bearings (minor); galvanized metal surfaces

The operation of an engine with compatible fuel shows that the amount of wear taken place in the sliding parts is in healthy condition. Wear has been evaluated from metals that contaminated the used lubricating oil. If the fuel is not compatible with the engine parts, then excess or abnormal wear is found in the engine parts. The maximum allowable limit of the wear and the quantity of dissolved material in the lubricating oil has been given in Table 5.9.

Table 5.9 Quantity of dissolve metals in the fresh and used lubricating oil during test

<table>
<tr><th>Name of item</th><th>ERH*</th><th>Fe (mg/kg)</th><th>Al (mg/kg)</th><th>Cr (mg/kg)</th><th>Cu (mg/kg)</th><th>Si (mg/kg)</th><th>Pb (mg/kg)</th><th>Sn (mg/kg)</th><th>Ni (mg/kg)</th></tr>
<tr><td>Critical value</td><td></td><td>100</td><td>25</td><td>20</td><td>50</td><td>25</td><td>50</td><td>20</td><td>20</td></tr>
<tr><td>Testing Method</td><td></td><td>AOAC 985.35</td><td colspan="7">EPA 051 A</td></tr>
<tr><td rowspan="3">First oil</td><td>0 (H1)</td><td>NIL</td><td>NIL</td><td>NIL</td><td>NIL</td><td>NIL</td><td>NIL</td><td>NIL</td><td>NIL</td></tr>
<tr><td>50 (H2)</td><td>56.67</td><td>4.78</td><td>2.23</td><td>13.9</td><td>13.5</td><td colspan="3" rowspan="7">Not Detected</td></tr>
<tr><td>150 (H4)</td><td>78.21</td><td>11.47</td><td>2.92</td><td>15.59</td><td>14.43</td></tr>
<tr><td rowspan="2">Second oil</td><td>50 (H2)</td><td>53.78</td><td>4.32</td><td>1.65</td><td>4.59</td><td>4.82</td></tr>
<tr><td>150 (H4)</td><td>84.82</td><td>11.37</td><td>1.69</td><td>8.67</td><td>34.4</td></tr>
<tr><td rowspan="2">Third oil</td><td>50 (H2)</td><td>46.27</td><td>4.22</td><td>0.56</td><td>8.92</td><td>4.89</td></tr>
<tr><td>150 (H4)</td><td>89.42</td><td>13.67</td><td>1.35</td><td>66</td><td>112</td></tr>
<tr><td>Final oil</td><td>62</td><td>60.23</td><td>11.13</td><td>1.62</td><td>18.4</td><td>8.37</td></tr>
</table>

* Engine Running Hours

It has been observed that Iron, Aluminum, and Chromium dissolved in all the evaluated samples were less than the referred critical values. Copper and Silicon dissolved in the lubricating oil after completion of 450 hours of engine run were found to be 66 mg/kg and 112 mg/kg, respectively, which is higher than the allowable limit of these materials, i.e., 50 mg/kg and 25 mg/kg, respectively [122]. Lead, Tin, and Nickel dissolved were less than the detection limit of the measuring machine. The detected limit of the machine for Lead, Tin, and Nickel is 5 mg/kg for each. The comparison of dissolved materials presented in the fresh oil and the used oil has been shown in Figure 5.1.

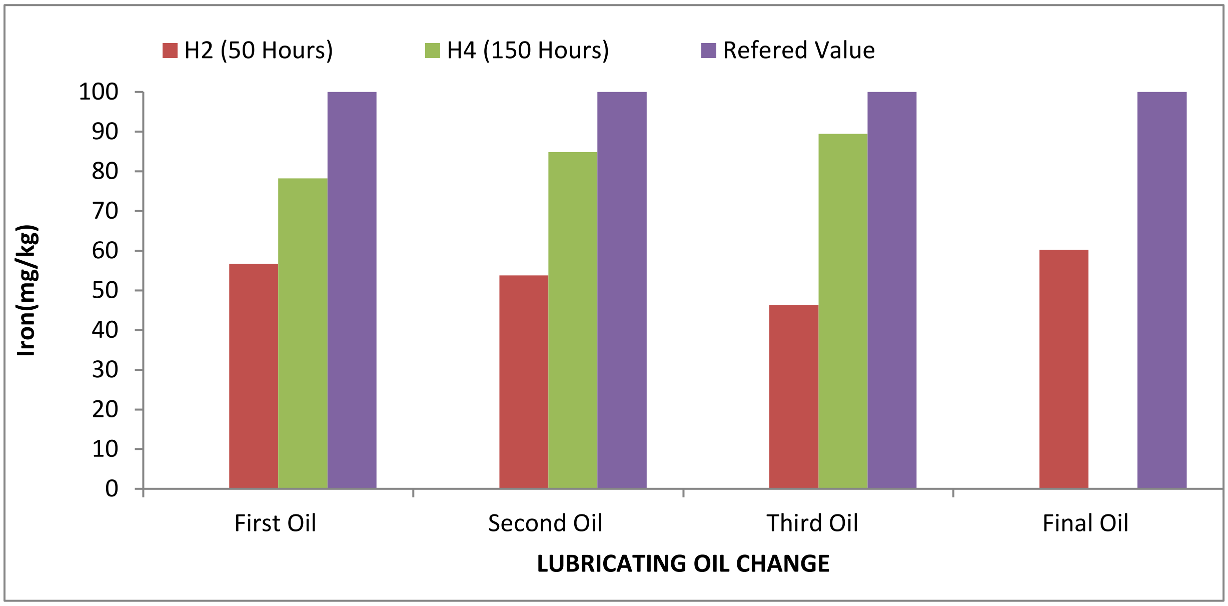


Figure 5.1 (a) Comparison of dissolved iron in lubricating oil for different oil change

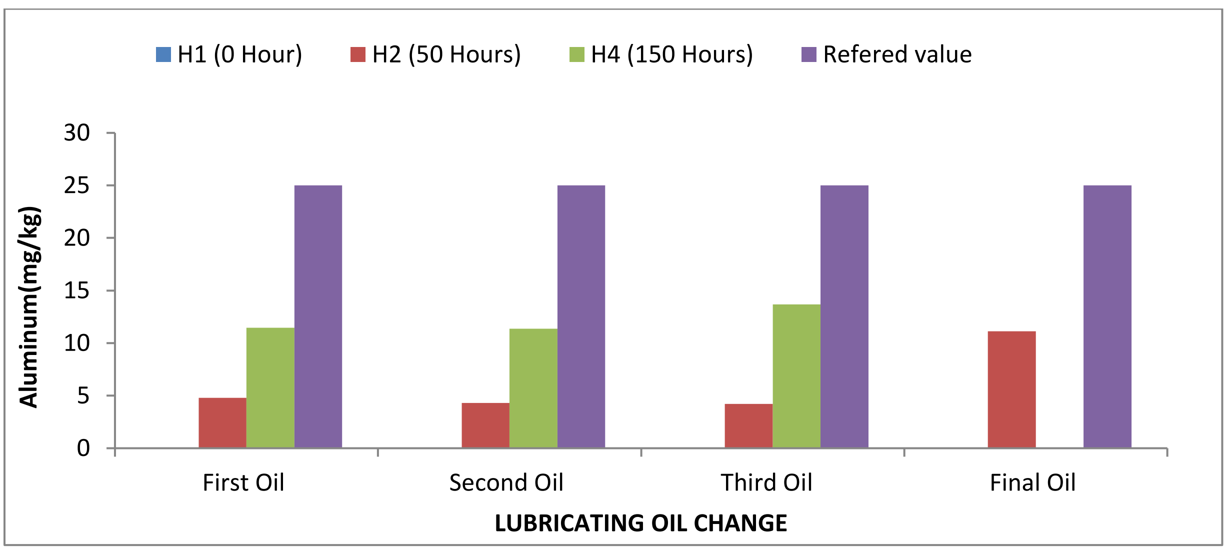


Figure 5.1 (b) Comparison of dissolved Aluminum in lubricating oil for different oil changes

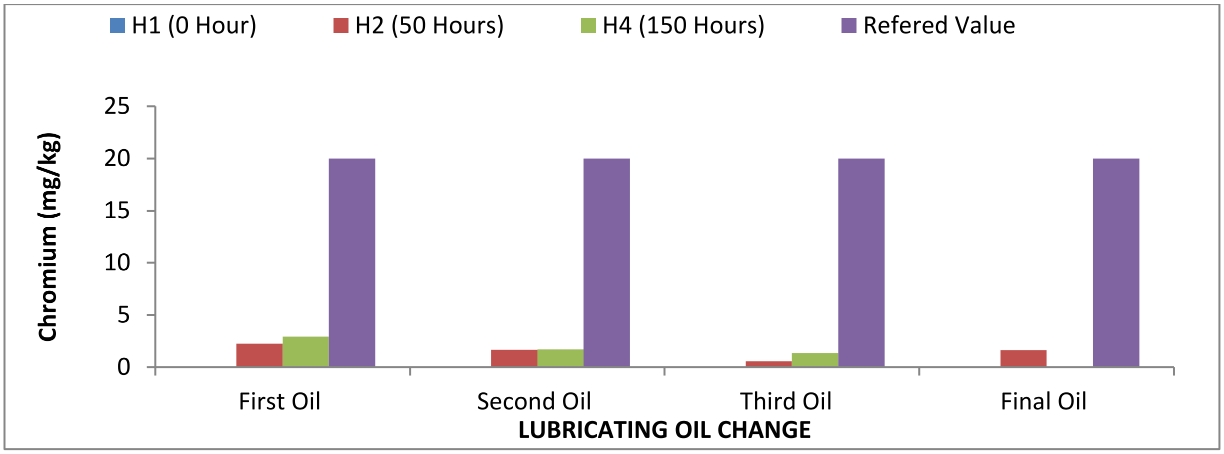


Figure 5.1 (c) Comparison of dissolved Chromium in lubricating oil for different oil changes

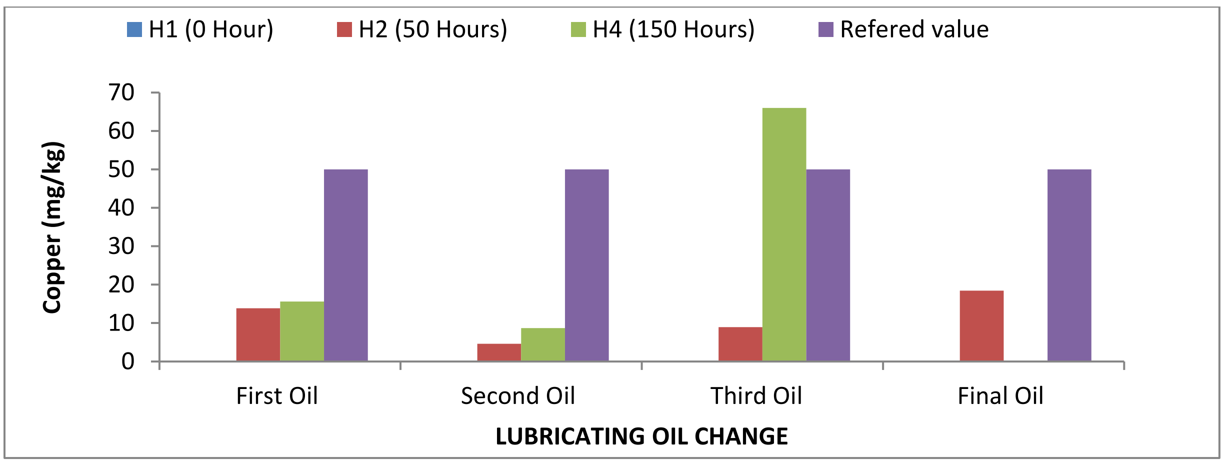


Figure 5.1 (d) Comparison of dissolved Copper in lubricating oil for different oil changes

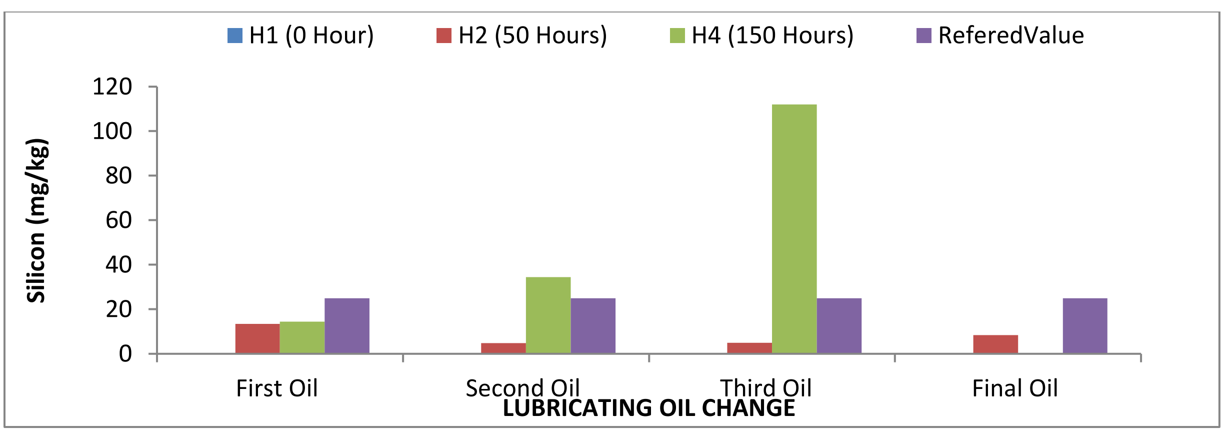


Figure 5.1 (e) Comparison of dissolved Silicon in lubricating oil for different oil changes

Figure 5.1 Comparison of dissolved material in lubricating oil for different oil change

Besides evaluating the presence of dissolved metals, another important parameter is viscosity that has to be maintained for the lubricating oil in the specified running of hours of the engine. Generally, 150 running hours were specified for identifying the change of lubricating oil in a diesel engine as per the IS 10000. The variation of the viscosity of the used oil samples with engine running hours has been shown in Figure.5.2. The results of the used lubricating oil samples indicate that there was a decrease in viscosity from 250 Centipoises of fresh oil to a minimum 187.5 Centipoises at 25^0C owing to high water content. The high water content has overcastted the effect of oxidation and contamination of the fuel. It has also been observed that the reduction in viscosity for the first lubricating oil was higher than others.

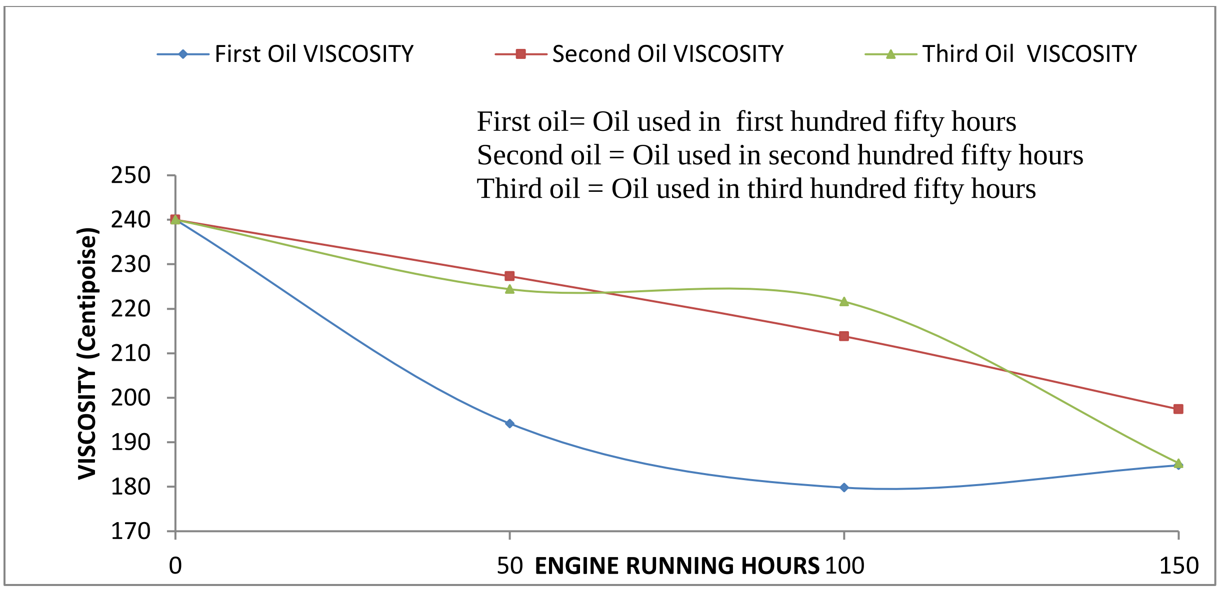


Figure: 5.2 Variation of viscosity with engine running time

5.4 VISUAL ANALYSIS OF INTERNAL PARTS BEFORE AND AFTER DURABILITY TEST

The condition of different parts both before and after the durability test is shown in Figure 5.3. Massive carbon deposition was observed on each element after the test. Though cracks and any type of damage were not found in the parts of the engine, the engine head gasket was observed to be significantly damaged after each 240-hours engine run. In the present test, after completion of each 240 hours of engine test run for each oil, it was observed that smoke is coming out with water of the engine. The water was used for engine cooling, and the smoke was present due to damage of engine head gasket. The photographic view of head gasket after 240 hours of the test run has been shown in Figure 5.4.

Engine Head before the test　　　　Engine Head after the test

Figure: 5.3 (a) Top view of Engine Head before and after the test

Fuel Injector before the test　　　　fuel Injector after the test

Figure: 5.3 (b) Fuel Injector before and after the test

Engine Suction and Exhaust Valve before the test Engine Suction and Exhaust Valve after the test

Figure: 5.3 (C) Suction and Exhaust Valve before and after the test

Top of piston before the test Top of the piston after the test

Figure: 5.3 (D) Piston top before and after the test

Figure: 5.3 Internal parts of the engine before and after the test

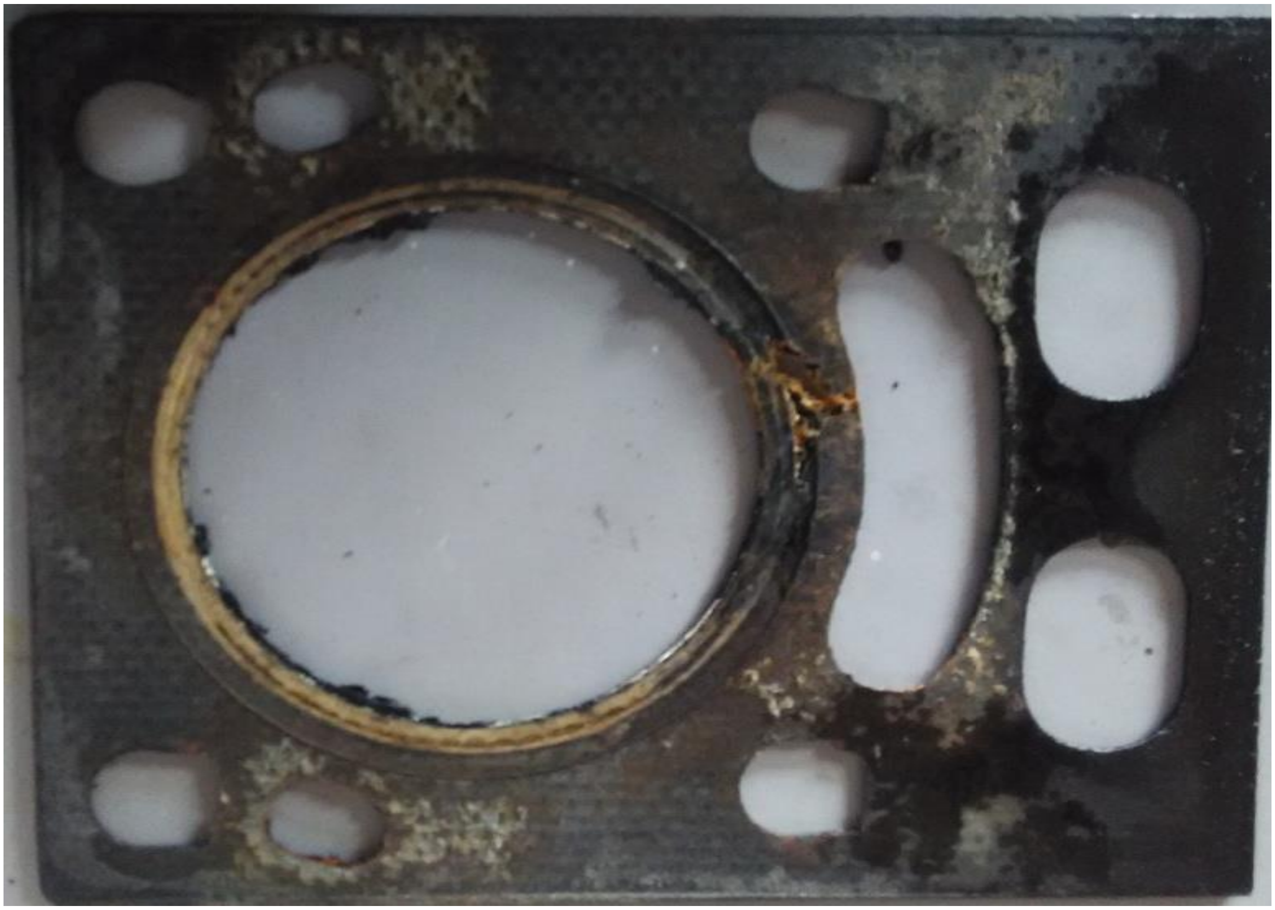


Figure: 5.4 Engine head gasket after the test

5.5 SUMMARY OF WEAR, LUBE OIL, AND VISUAL ANALYSIS

The wear on the engine parts were within the permissible limits. In the lubricating oil analysis, copper and silicon were found more than the allowable limit after 450 hours of engine test run. The change in viscosity of the lubricating oil was found within the permissible limits even if the lubricating oil was changed after every 150 hours of engine test run. Further, heavy carbon deposition was observed on the internal parts of the engine, and the engine head gasket was found to be damaged after each 240 hours engine run.

CHAPTER 6

COST ANALYSIS OF THUMBA OIL

6.1 INTRODUCTION

The analysis of the results of the experimental study revealed that the preheated mixture of 20% Thumba vegetable oil and 80% diesel can replace diesel fuel in CI engine with slight change in the maintenance schedule of the engine. The use of Thumba oil as fuel will save the diesel, which in turn will result in saving the foreign currency reserves of India. Many dangerous pollutants are emitted from CI engine fuelled with diesel. These emissions will reduce by using T20 Thumba oil diesel blend as fuel in CI engines. The use of Thumba oil will further lead to increase in the employment opportunities in the rural areas.

6.2 COST ESTIMATION OF THUMBA OIL

Vegetable oils are extracted from their plant seeds. The production cost of these oils includes the cost of seed production, seed collection, and oil extraction. The byproduct of Thumba seed can be used as animal food and can be sold in the market. The seed yield of Thumba plant is around 2500 kg/hectare to 3500 kg/hectare. Thumba seed contains approximately 19% to 25% of the oil on mass basis, but the considered yield in this study is 20%. This plant grows automatically in the fields, and no planned cultivation is needed. If Thumba plant is cultivated in an organized and planned manner, then its cost of farming will be about Rs. 300-800/hectare, which is quite less as compared to other vegetable oil plants like Jatropha and Karanj because their cost of farming is approximately Rs. 2800/hectare [20]. The cost of Thumba seed depends on planting density (number of plants per hectare), soil type, and intensity of post-planting operations. The production cost of the Thumba oil can be reduced by improving the organized cultivation of feedstock. The estimated cost of Thumba oil and its blends with diesel has been shown in Table 6.1.

Table 6.1 Estimated costs of Thumba vegetable oil and its 10% and 20% blend with diesel

Item	Quantity (kg)	Cost (Rs./kg)	Total cost (Rs.)

Thumba plant seed	5	12 (Market survey)	Rs. 60
Extracted Thumba oil	1	10 (Rs.2/kg of feedstock)	Rs. 10
Oil cake produced	4	5	Rs. 20 (revenue generated)
The net cost of Thumba oil production per kg			Rs. 50
The net cost of Thumba oil production per liter (specific gravity of Thumba oil 0.91)			Rs. 45.5
The net cost of Thumb blend T10 (10% Thumba oil and 90% diesel). (0.1 liter Thumba oil+ 0.9 liter Diesel)	1 liter T10		Rs. 58.55 {Rs. 4.55 (Thumba oil cost in a one-liter blend) + Rs. 54 (diesel cost in one-liter blend)}
The net cost of Thumba optimized blend T20 (20% Thumba oil and 80% diesel) as recommended in the paper. (0.2 liter Thumba oil+ 0.8 liter Diesel)	1 liter T20		Rs. 57.1 {Rs. 9.10 (Thumba oil cost in a one-liter blend) + Rs. 48 (diesel cost in one-liter blend)}
Present diesel cost per liter			Rs. 60
Expected economic potential T10 T20			Rs. 1.45 per liter saved if T10 Thumba oil blend is used instead of diesel. Rs. 2.90 per liter saved if optimized T20 Thumba oil blend is used in place

T100			of diesel. This is optimized and recommended blend in the current research. Rs. 14.50 per liter saved if pure Thumba oil is used in place of diesel
Crude oil price is variable, and presently, this is very low in the international market. If diesel price increases further, the bio-diesel will be more economical. Besides, if Thumba plant is regularly cultivated, its cost will reduce.			

6.3 CLIMATIC CONDITIONS AND LAND AVAILABILITY IN RAJASTHAN FOR CULTIVATION OF THUMBA PLANT

Rajasthan, a state of India, lies on the western part of India mostly in the tropical zone. The state is situated between 23^0 3' N and 30^0 12' N latitudes and 69^0 30' E to 78^0 17' E longitudes. Rajasthan has hot, dry, and desert climate. The average yearly rainfall of the state is 52 cm, most of which fall in the rainy months from July to September. Throughout the year, the temperature is extreme and the air has low humidity. On an average, the winter temperature ranges from 5^0 C to 28^0 C and the summer temperature ranges from 25^0 C to 48^0 C. Owing to less rainfall and dry climate, there is lack of forests and natural vegetation in Rajasthan [20].

Rajasthan is the largest state of India with the geographical area of 34.2229 million hectares; out of which the total wasteland is 10.1454 million hectares, which is around 29.64% of the entire geographic area. Rajasthan covers more than 10% of India's total land mass.

Different species of trees and creepers have been found to be very promising and suitable for oil production in adverse climatic conditions of Rajasthan. Apart from Neem, Jatropha, and Karanj plants, many other species like Thumba also yield oils as the source of energy in the form of fuel in Rajasthan. However, there are plenty of oilseeds that remain underutilized as fuel for CI engine. Thumba (Citrullus Colocynthis or colocynth) belongs to the species of the genus Citrullus of Cucurbitaceous family, which usually consists of many varieties that are generally known as melons. Citrullus Colocynthis (Thumba) plant belongs to the family of watermelon, and it has very high drought tolerance. Productivity is enhanced during the dry, sunny periods and reduced during the excessive rainfall and high humidity periods. The Colocynth (Thumba) plant is a native of dry soils. Thumba (Citrullus Colocynthis) is an unexploited perennial creeper that grows naturally in the hot Indian dry zone. Being a creeper, the soil binding capacity of Thumba is of considerable significance, and it has the potential to yield around 20 million tonnes of the oil-rich seeds from the arid districts of Rajasthan. Thumba plant has small crop cycle, and it has different uses that can play a pivotal role in the progress of the rural economy of Rajasthan. The raw Thumba seed oil, which is available in large quantities in Rajasthan, is presently consumed in the local soap industries [79].

Though very little information is available about the commercial cultivation of Thumba plant on a large scale, it usually grows naturally as a wild crop along with Bajra, the main crop of Rajasthan. The cultivation of Thumba plants as a pure crop for oil production on the 5.33 Mha wastelands of Rajasthan has been initiated, as it would be sufficient to meet the target of 5% replacement of diesel. The projected cultivation of Thumba plantation on wastelands of Rajasthan is shown in Table 6.2.

Table 6.2 Projected cultivation of Thumba Plantation on Wastelands of Rajasthan

1.	Average Fruits Collection (T /ha)	10-20
2.	Average Seed Yield (kg) (33 to 40% of the weight of fruits)	3300-3500 (3000 in the present study)
3.	Average Oil Yield (kg /ha) (20 to 25% on Seed Basis)	600 (20% in the present study)
4.	Wasteland available for Cultivation of Thumba Plant (million hectares)	10.1454
5.	Estimated availability of Thumba seed on the wasteland of Rajasthan/ six months (kg)	6087.24 million
6	Estimated availability of Thumba oil on the wasteland of Rajasthan/ six months {kg(liter)}	1217.448million (1337.8549 million)
7	Estimated availability of Thumba oil on the wasteland of Rajasthan/year (liter)	2675.7099 million

6.4 Life Cycle Cost Analysis

Life cycle cost is calculated based on all of the expenses generated till the fuel preparation. Life cycle cost of Thumba oil is the sum of investment cost, operating cost, and transportation cost. This total cost is the net cost of Thumba oil excludes taxes.

1. Investment cost includes cost of plant which extracts oil from the Thumba seeds. Life of this plant is to be assumed. It can be assumed 10 years.
2. Operating cost is the defined as future expenses which will be paid when the operation is running. Operating cost includes It includes the cost of feedstock, utilities, labours, maintenance and fuel, oil extracting and fuel preparation, as well as by-product credit.
3. Transportation cost is the cost of transportation of Thumba seed from cultivation area to oil extracting plant. But it may be excluded because oil extracting plants are installed in the production area. It will be included in the labour cost.

All these cost will be annualized and finally calculate the cost of Thumba oil production per kg. In the present research we have given it as one of future scope so we are including the method of Life cycle cost analysis on the suggestion examiner.

CHAPTER 7

CONCLUSIONS, FUTURE SCOPE AND RECOMMENDATION

7.1 CONCLUSIONS

After conducting an extensive experimental work on single cylinder diesel engine, which has been fuelled with diesel as well as preheated and unheated T20 Thumba oil-diesel blend, the important findings are concluded in this section. Some of the issues that need to be analyzed are discussed in the future scope of the present study with specific recommendations.

The following conclusions were derived from this rigorous and comprehensive experimental study:

- The viscosity of Thumba oil can be reduced by heating it to the temperature of approximately 90-100° C.
- After preheating, the properties of Thumba oil were found close to diesel.
- The single pass and one feet length heat exchanger was found sufficient to achieve the required temperature for preheating Thumba.
- The preheated T20 Thumba oil-diesel blend was found as an optimized blend.
- Blends having more than 30% concentration of the Thumba oil with diesel demonstrated poor results. But the results with T10 and T30 Thumba oil-diesel blends were close to T20 Thumba oil–diesel blend.
- Lowest BTE, highest BSFC, and maximum emissions were observed with use of pure Thumba oil.
- Low concentration blends of Thumba oil and diesel exhibit satisfactory results in the short-run engine operation. However, some operational difficulties like coking of injectors and piston, valves and piston sticking, carbon deposition on piston and cylinder walls and gum formation around valves and injector nozzle were observed while experimenting with regular Thumba oil as well as high concentration Thumba oil blends.
- Compression ratio of 21, injector needle lifts pressure of 203 bar, and the injection timing of 23 °CA BTDC were obtained as the optimized parameters upon operating the engine with diesel.

- Compression ratio of 22, injector needle lifts pressure of 203 bar, and the injection timing of 23 °CA BTDC were obtained as the optimized parameters upon operating the engine with preheated T20 Thumba oil-diesel blend.
- Highest BTE (26.6%) was observed upon using diesel as fuel at optimized condition of engine, and it was observed to be very close to BTE (25.93%) for preheated T20 Thumba oil diesel blend.
- Preheating improved 1.27% BTE for T20 Thumba oil diesel blend.
- Lowest smoke emissions were observed upon using the preheated T20 Thumba oil diesel blend (42.4%) as fuel. However, smoke emissions were high upon using the unheated T20 Thumba oil diesel blend (45%) and diesel (48%).
- CO emissions for the preheated, unheated T20 Thumba oil diesel blend, and diesel were 0.05%, 0.07%, and 0.06%, respectively.
- HC emissions (11PPM) were lower for the preheated T20 Thumba oil diesel blend than the unheated T20 Thumba oil diesel blend (16 PPM) and diesel (20 PPM).
- NOx emissions were lower for the unheated T20 Thumba oil diesel blend (248 PPM) than the preheated T20 Thumba oil diesel blend (259 PPM) and diesel (275 PPM). The preheated T20 blend emitted lesser NOx emissions than diesel fuel.
- The obtained maximum cylinder pressures at the optimized engine parameters for diesel, preheated, and unheated Thumba oil diesel T20 blend were 76.11 bar at an angle of 9°CA ATDC, 72.4 bar at an angle of 9°CA ATDC, and 69.78 at angle 8.5 °CA ATDC, respectively.
- The maximum heat release rate for diesel, preheated and unheated Thumba oil diesel T20 blend at the optimum condition were 46.4 J/°C , 35.28 J/ °C, and 31.2 J/°C, respectively.
- The maximum rate of pressure rise for diesel, preheated and unheated Thumba oil diesel T20 blend at optimum condition were 6.4 bar/°C, 4.2 bar/°C, and 3.8 bar/°C, respectively.
- Thumba oil is also economically viable, and the cost per litre of Thumba oil production is Rs. 45.5, which is lower than the current price of mineral diesel in India. The evaluated net cost of T20 Thumba oil blends is Rs 57.1 per litre in comparison to Rs. 60 per litre, which is the present price of diesel. The economic potential is Rs 2.90 per litre if T20 Thumba oil blends with diesel is replaced pure diesel.

- Lube oil analysis revealed that the quantity of Iron, Aluminium, and Chromium dissolved in lubricating oil was within the limits as referred by "National Biodiesel Board Report", but silicon and copper in the lubricating oil were found to be more than the allowable limit after 450 hours engine run.
- The visual inspection of internal parts of the engine showed massive carbon deposition on the piston, engine head, valves, and injector; but gum formation and ring sticking were not observed. No cracks and damage were found on the engine parts.
- Wear in piston outer diameter was observed to be 13 to 30 micron. Cylinder wear was about 80 micron. The closed gap in the oil piston ring increased up to 200 microns. Wear in other internal parts was not significant.

7.2 FUTURE SCOPE AND RECOMMENDATION

This research was initiated to resolve the problem of energy crisis in rural areas of the developing countries with the help of a source available in the rural areas itself. T20 Thumba Vegetable oil has been found to be the best alternate of conventional fuel in the present study. To reduce the viscosity of the T20 Thumba vegetable oil, a simple, economical, and sudden technique is used. The combination of blending and preheating is used to reduce the viscosity of Thumba oil. If the combination of blending, preheating, and micro-emulsion is used for reducing the viscosity of Thumba vegetable oil, then there is a possibility that the viscosity of T40 and T50 Thumba oil-diesel blend may reduce close to diesel and the effect of this combination can be observed on engine performance, combustion, and emissions.

The study is conducted on a single cylinder diesel engine, and it can be extended to multi-cylinder diesel engines, tractor engines, and other diesel engines used in agriculture and transport sector.

From the literature, it has been observed that the life cycle analysis is an essential feature of economic viability and feasibility of vegetable oil used as fuel in a CI engine. Therefore, the life cycle analysis may be carried out for the Thumba vegetable oil.

The preheated T20 Thumba oil blend with diesel is a viable and feasible alternate of diesel fuel in CI engine as per the conclusion of the current research. Therefore, it is recommended that the preheated T20 Thumba oil blend with diesel should be used as a fuel for CI engine in the rural areas for agriculture, irrigation, and power generation. However, the modified maintenance schedule may be adapted to control the carbon deposits formed during the long-term usage of this combination. National Institute for Transforming India (NITI Aayog) of the Government of India should develop a policy related to the implementation of the mixing of 20 % concentration of preheated Thumba oil with diesel for using it as fuel for engines.

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ANNEXURE 1

UNCERTANITY ANALYSIS

The values of various parameters obtained during the experiments could have errors or uncertainties due to operating conditions, environmental conditions, experimental methods adopted, calibration of equipments, accuracy and precision of equipments, human observations, test case planning, etc. Measurement errors fall into two main categories such as Systematic error and Random error. Imperfect calibration of measuring instruments (zero error), changes in environment which interfere with the measurement process and imperfect methods of observation caused the main sources of systematic error. Random error is caused by inherently unpredictable fluctuations in the readings of a measurement apparatus or in the experimenter's interpretation of the instrumental reading. In this investigation, the uncertainties were estimated from the minimum values of measured values as far as the individual measurements are concerned. To calculate the combined uncertainty of measurements a root-sum-square method was used. If a physical fundamental depends on n number of parameters, the combined uncertainties of each function was calculated by Pythagorean summation of uncertainties given by the equation.

$$z = f(x_1, x_2, x3, x_4 \ldots\ldots x_n)$$

$$\sigma_z^2 = [\partial f/\partial x_1]^2 \sigma x_1{}^2 + [\partial f/\partial x_2]^2 \sigma x_2{}^2 + [\partial f/\partial x_3]^2 \sigma x_3{}^2 + \ldots\ldots\ldots\ldots + [\partial f/\partial x_n]^2 \sigma x_n{}^2$$

σ_z = uncertanity of the function

$\sigma x_{n=}$ uncertanity of the parameter

f= function

x_n= parameter of the measurement

n= number of variable

There is possibility of error / uncertainty in the measurement of various experimental parameters obtained during the experimentation due to operating conditions, environmental conditions, calibration of equipment, accuracy and precision of equipment, human observations, test case planning, etc. The uncertainties of various parameters were calculated and uncertainty of various parameters has been presented in Table below

There is possibility of error / uncertainty in the measurement of various experimental parameters obtained during the experimentation due to operating conditions, environmental conditions, calibration of equipment, accuracy and precision of equipment, human observations, test case planning, etc. The uncertainties of various parameters were calculated and uncertainty of various parameters has been presented in Table below

Table An1: Uncertainty of various parameters

Measured quantity	Range of experiment	Resolution	% uncertainty
Load	0-12 Kg (0 -3.5 kW)	0.1kg (0.028 kW)	± 0.15
BTE	0-26.57%	-	± 0.11
BSFC	0.33 -0.78 kg/kWh	-	± 0.117
Smoke	6.5%-86%	0.1%	± 0.108
CO	0.04-0.22%	0.01%	± 0.16
CO_2	1.22-2.88%	0.1%	± 0.08
HC	6-20 ppm	≤ 2.000: 1 ppm vol. > 2.000: 10 ppm vol.	± 0.09
NO_x	132-412 ppm	1 ppm vol.	± 0.09

ANNEXURE 2

Table An2: Maximum allowable presence of metals in lubricating oil

	Iron (mg/kg)	Aluminum (mg/kg)	Chromium (mg/kg)	Copper (mg/kg)	Silicon (mg/kg)	Lead (mg/kg)	Tin (mg/kg)	Nickel (mg/kg)
DIESEL ENGINE	100	25	20	50	25	50	20	20
GASOLIN ENGINE	100	25	20	50	25	-	20	20

PUBLICATIONS

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Narayan Lal Jain, S. L. Soni, M. P. Poonia, Dilip Sharma, Anmesh K. Srivastava, Hardik Jain, 'Performance and emission characteristics of preheated and blended Thumba vegetable oil in a compression ignition engine', Applied Thermal Engineering 113 (2017) pp 970-979. **(Elsevier, SCI 3.77)**

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Publications as co author

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2. 'Effect of injection pressure on performance, combustion and emission characteristics of preheated and blended Thumba oil operated single cylinder stationary diesel engine'
3. 'Effect of injection timing on performance, combustion and emission characteristics of preheated and blended Thumba oil operated single cylinder stationary diesel engine'

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Research Paper

Performance and emission characteristics of preheated and blended thumba vegetable oil in a compression ignition engine

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HIGHLIGHTS

- Thumba straight vegetable oil is used as a fuel in CI engine.
- Engine waste heat is used to reduce the viscosity of thumba oil.
- Preheating improves thermal efficiency by 1.27% (24.66–25.93%).
- Preheating reduces emissions significantly as compared to unheated thumba.
- Drastically reduction in emissions as compared to diesel also.

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ABSTRACT

This research is motivated for making the rural areas of developing countries self-reliant in energy production. Rural areas are endowed with enormous amount of vegetation that is used to produce edible and non-edible vegetable oil. In the present study thumba oil which is non-edible vegetable oil, having high viscosity and low volatility has been used as a fuel in CI engine. Viscosity of thumba oil is reduced by blending with diesel fuel along with preheating through waste heat of engine's exhaust gases instead of transesterification, that requires chemicals and process heat, which is not available in rural area due to logistic problem. Experiment has been conducted with a single cylinder diesel engine to obtain performance and emission characteristics with various blend ratio of heated and unheated thumba oil with diesel. It has been observed that preheated thumba oil B-20 blend (20% thumba oil + 80% diesel) gives better performance and less emission as compared to all other blending combinations among all preheated and unheated thumba oil and diesel. Preheating of optimized thumba blend results an improvement in break thermal efficiency by 1.27%. It also gives reduced smoke opacity, CO, HC emissions by 2.6%, 0.2%, and 5 PPM respectively.

1. Introduction

Conventional fuels are depleting rapidly and they would be exhausted in few decades. There is a dire need to find the alternative for conventional fuel due to limited petroleum reserve and regular environment degradation. Alternative fuels should be renewable, economical, easily available and environment friendly. Vegetable oils are renewable and greener source of energy. Possibility of different edible and non edible vegetable oils as a fuel for CI engine has been accessed but these oils have not been used widely due to their limitation of high viscosity and low volatility. Vegetable oils have comparable energy density, cetane number, heat of vaporization, and stoichiometric air/fuel ratio with mineral diesel [1,2]. In addition, they are bio-degradable, non-toxic, and have a potential to significantly reduce pollution. Vegetable oils and their derivatives in diesel engines lead to substantial reductions in emissions of sulfur oxides, Carbon Monoxide (CO), Poly Aromatic Hydrocarbons (PAH), smoke, Particulate Matter (PM) [3,4]. Some of researcher had successfully studied on direct use of straight vegetable oil and its blends in CI engine for short term however; long-term endurance tests reported durability issue of the engine such as severe engine deposits, piston ring sticking,

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RESEARCH ARTICLE

A durability study of a compression ignition engine operating with Thumba (*Citrullus colocyntis*) vegetable oil

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Abstract

Vegetable oils are found suitable alternate of diesel fuel as per the results of short-run studies. Long-run studies with vegetable oil as a fuel pointed out the problems related to wear and maintenance of the engine. A single cylinder, variable compression ratio diesel engine was tested for 512 h (32 cycles of 16 h per day) to investigate longevity implications of fueling Thumba vegetable oil. Results of the study revealed that a very little damage was observed over the running surface of the cylinder liner, piston rings, valves, and valve seats. Wear in the piston outer diameter was observed to be 13 to 30 microns. Cylinder wear was about 80 microns. The closed gap in the oil piston ring increased up to 200 microns. Heavy carbon deposition was found on different internal parts of the engine, which indicates poor combustion of fuel. Amount of copper (66 mg/kg) and silicon (112 mg/kg) dissolved in the lubricating oil was found more than permissible limits (Cu 50 mg/kg, Si 25 mg/kg), after 450-h engine test run. But all the dissolve materials remain in allowable limits when the durability test conducted with diesel. Smoke, CO, HC, and NO_X emissions were found to increase initially then decrease in the further engine running hours. But these emissions were found inferior to the engine emissions fueled with diesel in all the running hours. CO2 emissions were found superior throughout the test with the preheated T20 Thumba oil blend than diesel. The maximum reduction in the viscosity of the lubricating oil, during endurance testing, was found 60 centipoises but it was found 25 centipoises when the test conducted with diesel.

Keywords Thumba oil · Wear analysis · Straight vegetable oil · Variable compression ratio · Compression ignition

Introduction

Diesel engines are widely used in the rural areas of developing countries in different agricultural activities, but the economic considerations of rural areas have restricted their extensive use (Hossain and Davies 2010; Ramning et al. 2013). People of rural areas have low energy consumption due to high price and less availability of energy resources, but they can fulfill their power requirement by generating power on their own using vegetable oils as a fuel in stationary diesel generators and tractors, vehicles, and diesel engines used for agriculture and irrigation purpose (Agrawal and Agrawal 2007; Sureshkumar et al. 2008). Vegetable oils are renewable and greener source of energy. The possibility of different edible and non-edible vegetable oils as fuel for the compression ignition (CI) engine has been accessed but these oils have not been used widely due to their limitations of high viscosity and low volatility. These oils have a comparable energy density, cetane number, heat of vaporization, and stoichiometric air/fuel ratio with mineral diesel (Agrawal and Agrawal 2007; Rakopoulos et al. 2006). Vegetable oil can replace and reduce the

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