EXPLORING THE APPLICABILITY OF *MESUA FERREA* LINN OIL AS AN ALTERNATIVE TO DIESEL IN COMPRESSION IGNITION ENGINES

A Thesis

Submitted in Partial Fulfillment of the Requirements for

the Award of the Degree of

DOCTOR OF PHILOSOPHY

By

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December, 2020

Abstract

Energy is needed for work to do, and therefore, it is required for growth and development of mankind. Energy is released when fuel reacts with other substances so that it can be used for work. Liquid fuel in the form of petrol, diesel and kerosene caters the most energy that is required to propel the automobiles, airplanes, ships, rockets and drive generators, pump, and industrial equipment. However, these energy sources are non-renewable, have limited reserves and are confined to some parts of world. This leads to frequent fluctuation of oil prices and compelling the non-oil producing countries to pay for importing oil. Moreover, the combustion of these fuels produce CO, CO_2, NOx, PM and SO_2 causing environmental pollution such as global warming, acid rain and fog. Thus, there is a need of an alternative energy source which is renewable, locally available and has lesser emissions. In this investigation, an attempt has been made to explore the feasibility of *Mesua ferrea* Linn seed oil (a type of vegetable oil) as an alternative to diesel to run the CI engines. This plant has high oil bearing seeds and it grows very well in the north eastern part of India especially in the state of Assam.

Initially, this vegetable oil (VO) extracted from the seeds of *Mesua ferrea* Linn tree is characterized and the VO is then blended with diesel at the volume percentage of 10, 20 and 30% (designated as VO10, VO20 and VO30). The VO blends are then tested in the 3.5 kW CI engine at various loads. A slight drop in brake thermal efficiency (BTE) with the use of VO is observed which up to 1.78, 3.94 and 5.47% with the use of 10, 20 and 30% blends, respectively. The brake specific fuel consumption (BSFC) increases with blending. In comparison to diesel, the VO blends produce higher CO and HC emissions. With the use of VO blends, the NO emissions is found to reduce especially in the higher engine loads. This reduction is higher with the higher amount of oil in the blend.

With the increasing proportion of VO in the oil-diesel blend, the performance of the engine tends to decrease along with the increase of CO and HC emissions. To explore the feasibility of redressing this drawback, ethanol is used as an additive to VO-diesel blend. Ethanol being of low viscosity, reduces the viscosity of the VO-diesel blend upon blending. Ethanol is added 5% and 10% by volume to both VO20 and VO30 blends. With 10% ethanol blend to VO20 and VO30, the BTE improves by approximately 1 to 4% and 1 to 1.5%, respectively. The BSFC of engine increases with the use of the

ethanol in both V2O and VO30 blends. The CO emissions of VO20 decreases and NO increases with the use of ethanol. However, the HC emissions decreases with the addition of 5% ethanol and it increases with the addition of 10% ethanol in the VO20 blend. The use of ethanol in VO30 results in an increase of CO and HC emissions; while it reduces the NO emissions up to 7 and 11% with the addition of ethanol 5 and 10%, respectively.

Later, the experiments are carried out using diethyl ether (DEE) as an additive in place of ethanol to VO-diesel blends (VO20 and VO30). DEE is added 5 and 10% to both VO20 and VO30. With the addition of DEE in the blends, the BSFC of the engine increases. The addition of DEE to VO30 increases the BTE of the engine as compared to neat VO30 which, on an average, is 0.6 and 1.6% with the addition of 5 and 10% DEE respectively. The use of DEE in VO20 and VO30 reduces the CO emissions. The addition of DEE in both VO20 and VO30 blends reduce the emissions of NO. However, the HC emissions increases with the blending of DEE with VO20 and VO30 blends.

Finally, the energy and exergy analyses of the diesel engine run on all the test fuels are studied to evaluate the quantity and quality of energy utilized and loss in the system. The shaft work increases with the increase of engine load. The fuel energy input increases with increase in the content VO in the VO-diesel blend as compared to neat diesel. On overall, the addition of ethanol and DEE in VO-diesel blends improves both the energy and exergy efficiency. The energy transferred to cooling water and uncounted loss decreases with the increasing engine load. The energy loss to exhaust also decreases with the increasing load. The availability destroyed decreases and the work or shaft availability increases with the increasing load. The exergy efficiency increases with the increasing load.

Keywords: Vegetable oil, *Mesua ferrea* Linn oil, Diesel, Ethanol, Diethyl ether, Blending, Compression ignition engine, Performance, Combustion, Emissions.

Declaration

I hereby certify that the work presented in this dissertation entitled 'Exploring the Applicability of Mesua Ferrea Linn Oil as an Alternative to Diesel in Compression Ignition Engines,' is utterly performed by myself, else stated, under the supervision of Professor Ujjwal K. Saha. Any part of this work has not earlier been submitted for the award of any degree, diploma, associate-ship, fellowship or its equivalent to any University or Institution.

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Certificate

It is certified that the work delineated in the thesis entitled 'Exploring the Applicability of Mesua Ferrea Linn Oil as an Alternative to Diesel in Compression Ignition Engines,' submitted by Mr. Maryom Dabi, a student in the Department of Mechanical Engineering, Indian Institute of Technology Guwahati, India, for the Award of the Degree of Doctor of Philosophy has been carried out under my supervision. This research work has not been submitted previously elsewhere for the award of any other degree or diploma.

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Acknowledgements

In the successful accomplishment of any work is backed by the contribution of many helping hands who directly or indirectly involves in the work. I do have received lots of unconditional support and contribution from humble personalities for successful bringing out this PhD thesis. It is my earnest pleasure to express my gratitude and thanks for their contribution in completion of thesis.

Firstly, I owe my sincere and heartfelt gratitude to my thesis supervisor Prof. Ujjwal K. Saha, for accepting and giving me an opportunity work under him as the PhD student. He supported, encouraged and guided me throughout my thesis. He led me to a new world of learning experience and open a new domain of avenue through his vision and wisdom. Best part of my working my under him is creating an environment of freehand work with unconditional support, constant guidance, counseling and encouragement to bring out my best to work. Without his support and encouragement, I could not have been able to complete this thesis.

I am truly indebted and thankful to Prof. Anoop K. Dass, the former Head of Department and Prof. Santosha K. Dwivedy, the present Head of Department of Mechanical Engineering Department for providing me research facilities and financial support. It is my privilege to associate with the members of doctoral committee, Prof. S. Kanagaraj, Prof. Chandan Das and Prof. V. Kulkarni. Their suggestions and encouragement help me in overcoming the various issues of this thesis work. I take this opportunity thanks again to Prof. S. Kanagaraj to allow me to carry out experiments at Material Science Laboratory of Mechanical Engineering Department. My special thanks go to Prof. V. S. Moholkar, the Head of Centre for Energy, IIT Guwahati, for allowing me to avail the experimental facilities.

I would like to express my thankfulness to the Scientific Officers, Mr. R. Saikia, Mr. N. Borah, Mr. J. Kakati, of Mechanical Engineering Department for providing logistic support and official works related to the IC engine Laboratory for ensuring smooth conduct of experiments. Sincere thanks go to Mr. S. Ahmed of Mechanical Engineering Department and Mr. D. Huzuri of Centre for Energy for helping me in conducting the experiments in their respective Laboratory. I cannot forget the helping hand given to me by Mr. Jaikrishna Saikia, Jr. Technical Superintendent, Department of Mechanical Engineering, for the maintenance of engine.

I would like to show my gratefulness to Dr. Achinta Sarkar, Dr. Bhaskor J. Bora and Dr. Bibhuti B. Sahoo for sharing their experiences and offering their helping hand during the course of my PhD work. I am overwhelmed by the lively work environment in the IIT Guwahati campus with my seniors and my colleagues Dr. Parag K. Talukdar, Dr. Santosh K. Hotta, Dr. Gaurav Goel, Dr. Jayanta Borah and Mr. Ojing Siram.

I am grateful to AICTE, Government of India for sponsoring me to pursue PhD under Quality Improvement Programme. I extend my earnest gratitude to Prof. P. G. Yammiyavar and Prof. S. Khijwania former Head/Coordinator and Mr. S. Debnath, Junior Superintendent of Centre for Educational Technology/QIP IIT Guwahati for their financial support.

I express my special thanks and gratitude to my parent institute NERIST, Itanagar for permitting to pursue this program. My unconditional gratitude goes to one and all of NERIST fraternity who directly or indirectly contributes and encourage me to pursue the work.

Lastly, I want to thanks all my family members who serve as my energy and morale booster to consistently carry out work even at low time.

December 29, 2020 Maryom Dabi

Contents

Nomenclature

Abbreviations

BMEP	Brake mean effective pressure
BSFC	Brake specific fuel consumption
BSEC	Brake specific energy consumption
bTDC	Before top dead center
BTE	Brake thermal efficiency
CA	Crank angle
CI	Compression ignition
CR	Compression ratio
DAD	Data acquisition device
DI	Direct injection
EGT	Exhaust gas temperature
HC	Hydrocarbon
HHO	Hydroxy gas
HRR	Heat release rate
IC	Internal combustion
IT	Injection timing
MAB	Microalgae biodiesel
MO	Mesua Ferrea Linn Oil
NHRR	Net heat release rate
PCP	Peak cylinder pressure
RoPR	Rate of pressure rise
SFC	Specific fuel consumption
SI	Spark ignition
TDC	Top dead center
THC	Total hydrocarbon
TPO	Tire pyrolytic oil
UHC	Unburned hydrocarbon
ULSD	Ultra low sulfur diesel
VO	Vegetable oil
WCO	Waste cooking oil
WPO	Waste plastic oil

Notations

CO	Carbon monoxide
NO	Nitrous monoxide
NO_2	Nitrogen dioxide
NOx	Oxides of nitrogen
SO_2	Sulphur dioxide

Symbols

%wt	Weight percentage
v/v	Volume /volume

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Chapter 1

Introduction

OVERVIEW

Fossil fuel in the form of coal and petroleum products provides most of the energy required for the production of electricity, heating and running of engines. However, this fossil fuel is non-renewable and has limited reserves for the future usage. Therefore, there is an urgent need of exploring the alternative energy sources for supplementing or replacing the fossil fuels. In this context, biofuels derived from biomass fit the best option for supporting the burden of fossil fuel because of renewability and availability. Biofuels in the form of vegetable oil, biodiesel, and others can supplement or replace the fossil fuel. This chapter starts with an overview of global and Indian energy scenario, followed by an introductory remark on internal combustion engines. The application of biofuels in CI engines together with various emissions standards are briefly summarized. The objectives of the present investigation are then presented. Finally, the chapter ends with the organization of the thesis.

Chapter Outline:

1.1 Inspiration

In general speaking, the security of a country is threatened either by enemy, economic crises, violence or crime. However, in the current context, an energy crisis can create even more than that. Many energy wars have been encountered in the world history. Energy, in the form of hot gases inside the engine cylinder, is used to propel the automobiles, airplanes, ships, rockets and drive generators, pump, and industrial equipment. Without these energy consuming devices and equipment, which are the lifeline of a country, one cannot think of living in a country with peace and feel secured. Energy is derived from fuel and main the source of fuel is fossil in the form of coal and petroleum products such as diesel, petrol, kerosene, etc. The problems with fossil fuel is that it is non-renewable, has limited reserves in the earth crust, and are mostly concentrated in certain regions of the world. The implication of this is that a fossil fuel deficit country has to import it investing huge amount of capitals. Moreover, the high dependency of energy resources through import is not at all good for the security and progress of the country. Above all, the burning of fossil fuels results in the pollution of environment affecting flora, fauna and triggering of change in the climate due to the emissions of gases such as CO_2, CO, NOx and unburned hydrocarbons (HC). This necessitates the need of an alternatives to supplement or replace the energy need which is renewable and available. In this perspective, biofuels derived from the biomass seem to be the most preferred alternative source. Biofuels, in the form of alcohol, biodiesel, vegetable oil, pyrolytic oil, and others are produced when different types of biomass are subjected to different processes.

1.2 Global and Indian Energy Overview

With the growing population due an advancement in the health care sector and improvement in the living standard across the world, energy demand also increases by about 15% in last ten years. In the year 2010, the global energy consumption was 12099.9 million tonnes of oil equivalent (mtoe) which increases to 13865 mtoe in the year 2018 as shown in the Figure 1.1. While India has registered around 50% increase in the consumption of energy at the same period as compared to the global energy consumption. The energy consumption increases from 539.2 mtoe in the year 2010 to 809.1 mtoe in 2018 as revealed the Figure 1.2. The energy is derived from oil, natural gas, coal, nuclear, hydroelectricity and renewables. The renewables include the energy derived from wind, solar, geothermal, biomass and waste. Coal and oil caters the major part of the energy demand. The combined share of coal and oil for global energy consumption, as shown in Figure 1.3, was 65, 64 and 61% in the year 2010, 2014 and 2018, respectively. The natural gas stands third rank in the energy consumption across the world with the share of 23 to 24%. The share of renewables increases from 1% in the year 2010 to 4% in the year 2018. The consumption of renewables was 170.6 and 561.3 mtoe for the year 2010 and 2018, respectively. While for India, coal and oil constitutes 84, 86 and 86% of the total energy consumed during the year 2010, 2014 and 2018, respectively (Figure 1.4). The share of renewables increases from 1% in the year 2010 to 3% in the year 2018. Figure 1.5 shows the global and Indian energy consumption per capita for the year 2010 to 2018. It increases with the progressing year. Globally it increases from 72.8 GJ in the year 2010 to 76 GJ in the year 2018. The Indian per capita energy also increases from 18.3 GJ in the year 2010 to 25 GJ in the year 2018. However, global per capita energy consumption is much higher as compared to India.

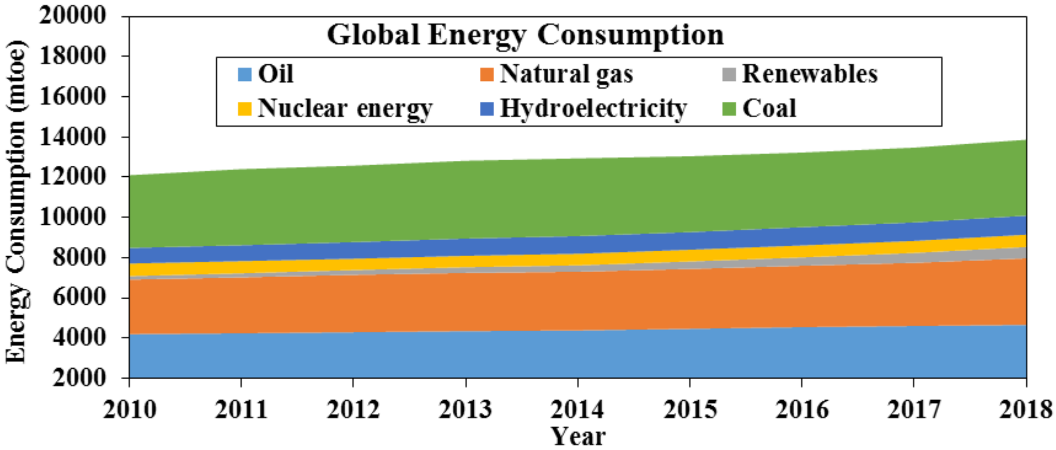


Figure 1.1 Global energy consumption for the year 2010 to 2018 (BP Statistical Review of World Energy 2019)

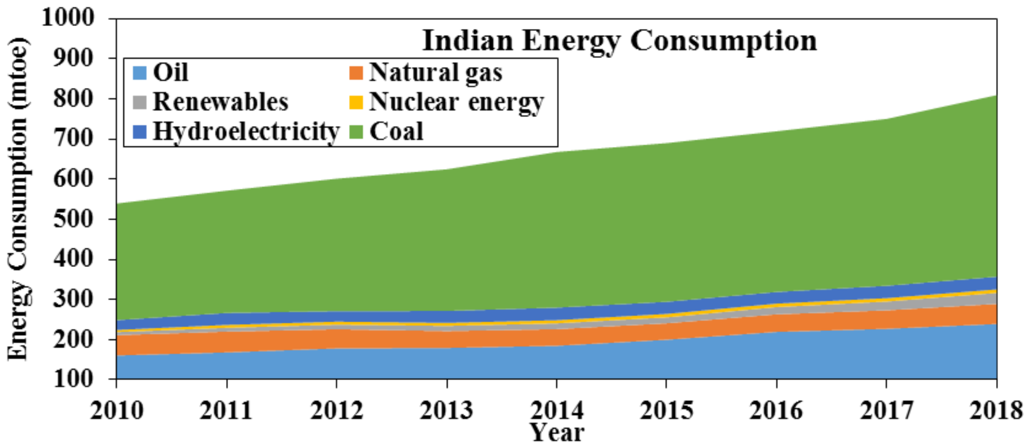


Figure 1.2 Indian energy consumption for the year 2010 to 2018 (BP Statistical Review of World Energy 2019)

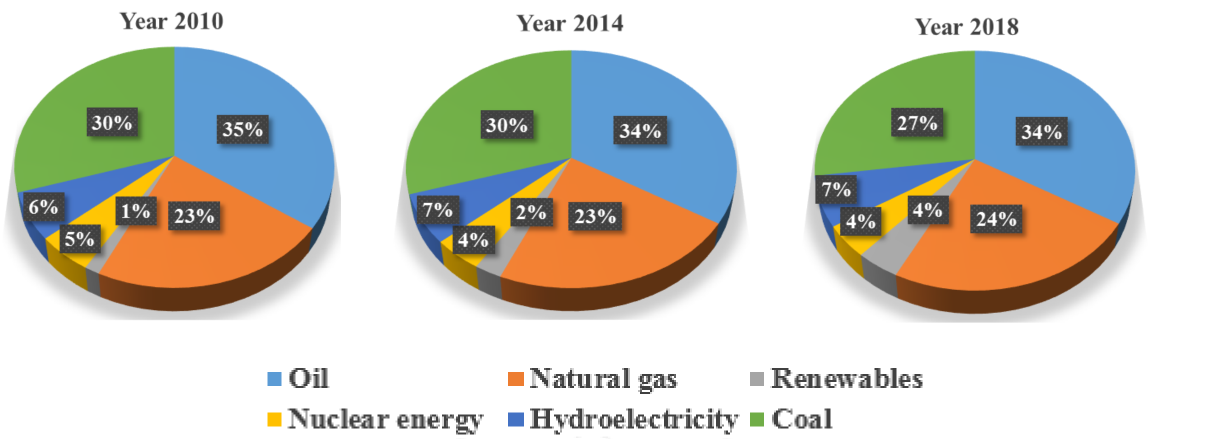


Figure 1.3 Share of global energy consumption by fuel for the year 2010, 2014 and 2018 (BP Statistical Review of World Energy 2019)

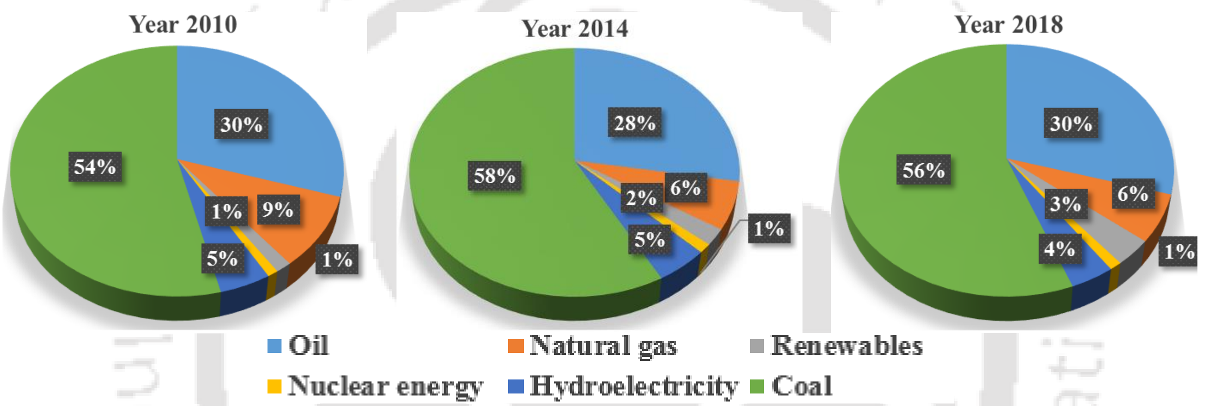


Figure 1.4 Share of Indian energy consumption by fuel for the year 2010, 2014 and 2018 (BP Statistical Review of World Energy 2019)

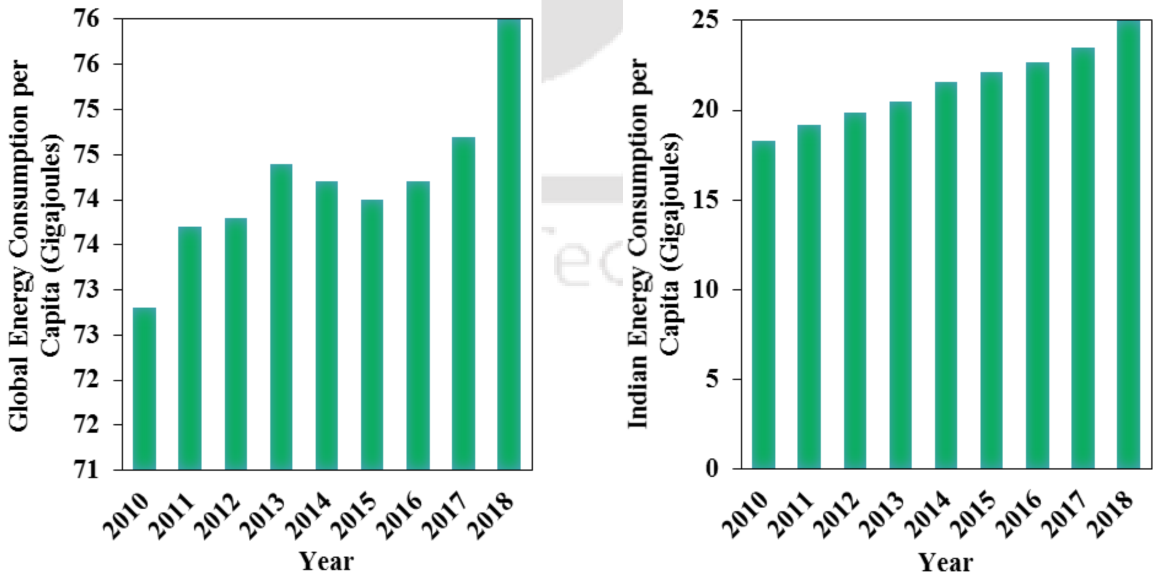


Figure 1.5 Global and Indian energy consumption per capita for the year 2010 to 2018 (BP Statistical Review of World Energy 2019)

1.3 Reciprocating Internal Combustion Engines

In a reciprocating internal combustion (IC) engine, the chemical energy stored in the fuel is converted into thermal energy through the combustion process inside the engine cylinder. This thermal energy in the form of high temperature and pressure gas expanded against the moving piston, which is in communication with crankshaft of engine through connecting rod, converts the reciprocating motion of piston to the rotary motion of the crankshaft. This rotary motion can be used to drive the vehicle, pump, generator, and others. Thus, the chemical energy of fuel is converted into mechanical energy in the internal combustion engine. The fuel used may be either liquid or gaseous. The combustion of fuel in an IC engine is initiated either by a spark or by injecting the fuel at high pressure against the highly compressed air. The engine where the combustion is initiated through an electric spark is called as the spark ignition (SI) engine; while the engine where the combustion initiated through the injection of high pressure fuel into highly compressed air is called as the compression ignition (CI) engine. In the SI engine, homogeneous mixture of air fuel is admitted to engine cylinder during the suction stoke which is then compressed and then at the end of compression, a spark plug mounted on cylinder head ignite the air fuel mixture. In the CI engine, only the air is compressed and at the end of compression, the fuel is injected into the cylinder which auto ignites due to high temperature air in the cylinder generate by the compression.

1.4 Vegetable Oil as a Fuel to CI Engines

The use of vegetable oil as fuel in CI engine was started by Dr. Rudolph Diesel. He used the peanut oil to fuel one of his engines at the Paris Exposition of 1900 (Altin et al. 2001). Though the use of vegetable oil in CI engine is almost as old as the invention of engine, it could not make its way for its utility on engine due to the discovery of petroleum fuel. The diesel derived from crude petroleum distillation emerges as the prominent fuel for CI engine because of superior quality. However, during last 20-30 years, the necessity of an alternative to diesel has been felt around the world because of non-renewability and limited reserves. As a consequence, vegetable oil (VO), as an alternative to substitute or supplement diesel, has become the focal point of engine research and development. The advantages of using vegetable oil in engine is that oil can easily be produce simply using an oil expeller, and does not require any complicated chemical reaction process. Further, as reported in literature, any type of VO bearing seeds can be used in CI engines. The oils derived from the seeds of jatropha, karanja, palm, deccan hemp, hazelnut, poon, sunflower, cotton have been extensively used in the past investigations. However, the use of neat VO is found to deteriorate the engine performance. The use of preheated VO is found to improve the engine performance as compared to the normal neat oil. The blending of VO with diesel is found to be the effective in CI engines.

1.5 Emissions Standards

The air pollution caused by the exhaust emissions from vehicles that run on fossil fuel necessitate the need of stringent laws to limit the emissions. The first European vehicle emissions limits were introduced in the year 1970 and by 1982 the emissions level for the controlled emissions was around 50% as compared to that of the year 1970 (Smith and Davies 1996). Euro 1 which was introduced in the year 1992 became the first European emissions standard for the new vehicles. In the subsequent years, further stringent emissions standard (Euro 2, Euro 3, Euro 4, Euro 5 and Euro 6) were introduced. The emissions limits of CO, NOx, HC and particulate matter (PM) under the Euro series for new diesel and gasoline vehicle is shown in Table 1.1 and Table 1.2 respectively (The AA; Keska and Janicka 2017; Frost and Smedler 1995). The CO emissions has been restricted from 2.72 g/km in Euro 1 to 0.5 g/km in Euro 6 for diesel vehicles while it is limited to 1 g/km in Euro 6 for gasoline vehicles. In India, the first initiative to regulate the emissions from vehicle was started in 1989 by Ministry of Environment and Forests under the Environment (Protection) Act (CPCB 2010). The emissions limits for passenger cars and heavy diesel vehicles under various norms are shown in Table 1.3 and Table 1.4, respectively (CPCB India).

Table1.1 European emissions norms for diesel vehicles.

Norms	CO (g/km)	HC + NOx (g/km)	PM (g/km)	NOx (g/km)
Euro 1	2.72	0.97	0.14	-
Euro 2	1.0	0.7	0.08	-
Euro 3	0.64	0.56	0.05	0.5
Euro 4	0.5	0.3	0.025	0.25
Euro 5	0.5	0.23	0.005	0.18
Euro 6	0.5	0.17	0.005	0.08

Table 1.2 European emissions norms for gasoline vehicles.

Norms	CO (g/km)	HC + NOx (g/km)	PM (g/km)	NOx (g/km)	HC (g/km)
Euro 1	2.72	0.97	-	-	-
Euro 2	2.2	0.5	-	-	-
Euro 3	2.3	-	-	0.15	0.2
Euro 4	1.0	-	-	0.08	0.1
Euro 5	1.0	-	0.005[a]	0.06	0.1
Euro 6	1.0	-	0.005[a]	0.06	0.1

[a]Direct injection only

Table 1.3 Emissions norms for passenger cars in India.

Norms	CO (g/km)	HC+NOx (g/km)
1991 Norms	14.3-27.1	2.0 (Only HC)
1996 Norms	8.68-12.40	3.00-4.36
1998 Norms	4.34-6.20	1.50-2.18
India Stage 2000 Norms	2.72	0.97
Bharat Stage-II	2.2	0.5
Bharat Stage-III	2.3	0.35 (combined)
Bharat Stage-IV	1.0	0.18 (combined)

Table 1.4 Emissions norms for heavy diesel vehicles in India.

Norms	CO (g/kmhr)	HC (g/kmhr)	NOx (g/kmhr)	PM (g/kwhr)
1991Norms	14	3.5	18	-
1996 Norms	11.2	2.4	14.4	-
India stage 2000 norms	4.5	1.1	8.0	0.36
Bharat stage-II	4.0	1.1	7.0	0.15
Bharat Stage-III	2.1	1.6	5.0	0.10
Bharat Stage-IV	1.5	0.96	3.5	0.02

1.6 Present Objectives

In most of the investigations related to the use of VO in CI engines, the researchers have focused on deriving the oil from feedstock which is available in that particular area. The north eastern (NE) region of India is endowed with large varieties of flora. Many of them have oil bearing seeds which can be used as fuel in diesel engines. The oil derived from *Mesua ferrea* Linn seeds seems to be promising because of its high oil bearing seeds and availability. These plants are grown in residents, offices, institutes, parks as ornamental plants because of their aesthetic look. The motivation behind this research work is to explore the applicability of *Mesua ferrea* Linn oil (a type of VO) in stationary diesel engines that are used in rural areas for irrigation and power generation. In this investigation, VO derived from *Mesua ferrea* Linn seeds is being considered as a fuel to run the CI engine with the following objectives.

- Production and characterization of VO derived from *Mesua ferrea* Linn seeds
- Study the effects of VO-diesel blends on performance, combustion and emissions characteristics of the CI engine
- Study the effects of VO-diesel-additives blends on performance, combustion and emissions characteristics of the CI engine
- Study the energy and exergy analysis of the CI engine run on diesel, VO-diesel blends and VO-diesel-additives blends.

1.7 Organization of the Thesis

The thesis comprises of nine chapters covering all the aspects of the study. *Chapter 1* introduces the motivation behind the present research work and the ultimate driving force to set the objectives. *Chapter 2* reviews the past literature covering the investigations on the use of VO in CI engines under different modes such as neat, preheated and blending. *Chapter 3* focuses into the production and characterization of *Mesua ferrea* Linn oil to be used in the performance, combustion and emissions studies of the CI engine. *Chapter 4* gives the details of the experimental setup, instrumentation and the procedures followed during the experiments. The experimental results of binary blend of VO and diesel are described in *Chapter 5*. In *Chapter 6*, the experimental results of ternary blends of VO, diesel and ethanol are discussed, whereas the results ternary blends of VO, diesel, diethyl ether (DEE) are discussed in *Chapter 7*. *Chapter 8* is dedicated to the energy and exergy analyses. *Chapter 9* concludes with the major findings of the entire investigation and highlights the future scope of research.

Chapter 2

Literature Review

OVERVIEW

The oil derived from non-edible vegetable feedstock has the potential to supplement or replace the fossil diesel in CI engines. This type of vegetable oil can be used in CI engines through different modes such as neat, preheated and blending. This chapter explores the implications of neat, preheated and blended vegetable oil on the performance, combustion and emissions characteristics of the engine by reviewing the experimental investigations carried out by researchers on the use of vegetable oil in CI engines. The chapter starts with the overview of vegetable oil which includes the properties of oil derived from different feedstocks. Then the effects of neat, preheated and blend vegetable oil on engine performance, combustion and emissions have extensively been covered. This is followed by reviewing the use of ethanol and diethyl ether as additives to the vegetable oil-diesel blend. Finally the chapter ends with the summary and the scope of the work.

Chapter Outline:

2.1 Vegetable Oil

Vegetable oil (VO) is the oil that is derived from plants. It includes the oils that are extracted from both edible and non-edible feed stocks such as cotton seed, corn, hazelnut, soybean, sunflower, and others. Oil is extracted from the seeds either through the methods of mechanical, solvent or enzymatic extraction process. The seeds are either sundried or oven dried before the extraction (Atabani *et al*. 2012). Since the oil is extracted directly from vegetable, it generally termed as straight vegetable oil or crude filtered oil in many of the literatures. The VO is composed of 90 to 98 % triglycerides and small amount of mono and diglycerides. Triglycerides are esters of three fatty acids (FAs) and one glycerol (Srivastava and Prasad 2000). The most common FAs that are found in VOs are shown in Table 2.1. This table also shows the nomenclature, structure, formula and molecular mass of FA. Further, FAs are classified into saturated and unsaturated based on the number of double bonds. A saturated FA acid does not have any double bond, while an unsaturated FA can have one or more double bonds.

Table 2.1 Common FAs found in VO (Sanjid *et al*. 2014; Srivastava and Prasad 2000).

FA Name (common)	FA Name (systematic)	Structure *	Formula	Molecular Mass
Myristic	Tetradecanoic	14:0	$C_{14}H_{28}O_2$	228
Palmitic	Hexadecanoic	16:0	$C_{16}H_{32}O_2$	256
Palmitoleic	Hexadec-9-enoic	16:1	$C_{16}H_{30}O_2$	254
Stearic	Octadecanoic	18:0	$C_{18}H_{36}O_2$	284
Oleic	Cis-9-Octadecanoic	18:1	$C_{18}H_{34}O_2$	282
Linoleic	Cis-9-cis-12 Octadecanoic	18:2	$C_{18}H_{32}O_2$	280
Linolenic	Cis-9-cis-12, cis-15-Octadecatrienoic	18:3	$C_{18}H_{30}O_2$	278
Arachidic	Eicosanoic	20:0	$C_{20}H_{40}O_2$	312
Eicosenoic	Cis-11-eicosenoic acid	20:1	$C_{20}H_{38}O_2$	310
Behenic	Docosanoic	22:0	$C_{22}H_{44}O_2$	341
Erucic	Cis-13-docosenoic	22:1	$C_{22}H_{42}O_2$	339
Lignoceric	Tetracosanoic	24:0	$C_{24}H_{48}O_2$	369

**xx:y* indicates *xx* carbons in the fatty acid chain with *y* double bonds.

Usually, the FA composition of VO is determined from gas chromatography. The typical composition of common FAs for different feed stocks has been shown in the Table 2.2. Stearic, oleic and linoleic FAs constitute the major composition of VO. Their composition varies with the feedstock. Canola, rapeseed, olive and hazelnut oil respectively are composed of 53.36, 64.4, 75 and 77.15% oleic FA. While corn, cottonseed, poon, sunflower and soybean oil are composed of more than 50% linoleic FA. In the study of Machacon *et al*. (2001), the coconut oil is exceptionally found to have 9.5, 4.5 and 51% of caprilic, capric and lauric respectively, along with the common FAs. Even the same feedstock shows variations in the composition of FA. This variation in the composition may be due to the growing of feedstock at different parts of the world having different climatic and geographical conditions. The availability and distribution of VO resources in India and across world has been extensively covered in some of past studies (Atabani *et al*. 2012; Atabani *et al*. 2013; Mofijur *et al*. 2013; Silitonga *et al*. 2013; Ashraful *et al*. 2014).

Table 2.2 Typical Fatty acid compositions of VO derived from different feedstock.

SVO Feed stock	Fatty acid compositions (wt%)										References
	14:0	**16:0**	**16:1**	**18:0**	**18:1**	**18:2**	**18:3**	**20:0**	**22:0**	**24:0**	
Canola	0.11	6.45	0.27	2.54	53.36	29.81	5.63	0.42	0.33	0.30	Atmanlı *et al.* (2014)
Coconut	18.5	7.50	-	3.00	5.00	1.00	-	-	-	-	Machacon *et al.* (2001)
Corn	0.09	11.17	0.15	2.20	31.80	52.36	0.91	0.40	0.16	0.16	Atmanlı *et al.* (2014)
	-	12.00	-	2.00	25.00	61.00	-	-	-	-	Rakopoulos *et al.* (2011)
Cottonseed	-	28.00	-	1.00	13.00	58.00	-	-	-	-	Rakopoulos *et al.* (2011)
	-	11.67	-	0.89	13.27	57.51	-	-	-	-	Ramadhas *et al.* (2005)
	0.79	23.13	0.20	2.28	19.08	52.50	0.22	0.29	0.51	0.13	Atmanlı *et al.* (2014)
Hazelnut	0.04	5.50	0.08	2.00	77.15	14.86	0.04	0.04	0.04	-	Atmanlı *et al.* (2014)
Jatropha	0.1	12.80	0.90	5.90	39.70	39.20	0.50	0.30	0.10	0.10	Koder *et al.* (2018)
Olive	-	5.00	-	2.00	75.00	18.00	-	-	-	-	Rakopoulos *et al.* (2011)
Poon oil	-	22.4	-	7.30	16.42	45.89		6.47	-	-	Devan and Mahalakshmi (2009)
Rapeseed	-	3.49	-	0.85	64.40	22.30	8.23	-	-	-	Ramadhas *et al.* (2005)
		4.6		3.2	60.7	20.5	9.3	0.6			Raman *et al.* (2019)
Rubberseed	-	10.2	-	8.70	24.60	39.60	16.3	-	-	-	Ramadhas *et al.* (2005)
Soybean	-	11.75	-	3.15	23.26	55.53	6.31	-	-	-	Ramadhas *et al.* (2005)
	-	10.58	-	4.76	22.52	52.34	8.19	0.36	-	-	Canakci (2005)
	0.10	10.26	0.11	3.52	26.55	51.04	7.06	0.23	0.26	0.27	Atmanlı *et al.* (2014)
	0.10	10.80	0.10	3.20	25.20	53.00	6.20	0.40	0.50	0.20	Koder *et al.* (2018)
Sunflower	-	6.00	-	3.00	17.00	74.00	-	-	-	-	Rakopoulos *et al.* (2011)
	-	6.80	-	3.26	16.93	73.73	-	-	-	-	Ramadhas *et al.* (2005)
	0.08	5.33	0.12	3.45	37.13	52.01	0.13	0.16	0.65	0.14	Atmanlı *et al.* (2014)
	-	10.22	-	2.69	24.70	62.39	-	-	-	-	Yesilyurt *et al.* (2020)

Other than the FA composition, the VOs are generally characterized on the basis of their density, viscosity and calorific value. These properties have significant importance especially in the context of applicability of it in IC engine. Table 2.3 shows the density, viscosity and calorific value of oil derived from different feedstock. The average calorific value is in the range of 36 to 39 MJ/kg, while the density and the viscosity show fluctuating values depending upon the oil temperature. The same feedstock is showing different properties due to the variation in FA compositions.

Table 2.3 Properties of VO derived from different feedstock.

Feed stock	Heating Value (MJ/kg)	Density (kg/m^3)	Viscosity[f] (cSt)	References
Canola	37.400	915.00[b]	31.80	Bayındır *et al.* (2017)
Cashew nut	35.800	958.10[a]	55.30	Kasiraman *et al.* (2012)
Corn	36.300	915.00[a]	35.00	Rakopoulos *et al.* (2006)
	37.825	915.00	46.00	Altin *et al.* (2001)
	35.10	915.00[f]	35.10	Shehata *et al.* (2015)
Cotton seed	39.600	912.00	50.00	Ramadhas *et al.* (2005)
	36.800	910.00[a]	34.00	Rakopoulos *et al.* (2006)
	39.648	912.00	50.00[c]	Altin *et al.* (2001)
	39.648	914.00	50.00	Raj *et al.* (2018)
Croton mogalocarpus	36.980	920.00[d]	33.38	Lujaji *et al.* (2011)
Deccan hemp	38.720	913.00[e]	53.00[e]	Hebbal *et al.* (2006)
Hazelnut	39.840	910.00[a]	24.00	Atmanli *et al.* (2016)
Jatropha	38.200	932.92[e]	52.76[e]	Pramanik (2003)
	37.500	918.00	37.00	Chauhan *et al.* (2010)
	39.774	918.00	49.90	Reddy and Ramesh (2006)
Karanja	34.000	912.00	27.84	Raheman and Phadatare (2004)
	41.660	938.00	35.98	Agarwal and Rajamanoharan (2009)
	37.304	913.00	27.84	Bajpai *et al.* (2009)
	-	956.00	59.84	Shah and Ganesh (2018)
	38.800	929.00	46.50	Reddy *et al.* (2018)
Lemon peel	41.510	853.00	1.06	Ashok *et al.* (2017)
Lemongrass	36.270	984.00[c]	04.18	Sathiyamoorthi and Sankaranarayanan (2017)
Linseed	39.750	864.50	16.23	Agarwal *et al.* (2008)
Mahua	38.863	904.00	37.18	Agarwal *et al.* (2008)
	38.650	912.00[a]	37.84	Sonar *et al.* (2015)
Olive kernel	37.000	925.00[a]	32.00	Rakopoulos *et al.* (2006)
Opium poppy	38.820	921.00	56.00[c]	Altin *et al.* (2001)
Orange seed	34.650	816.90[e]	3.52	Purushothaman and Nagarajan (2009)
Palm	39.849	925.00[a]	41.00	Gad *et al.* (2018)
Poon	39.650	926.40[f]	49.70	Devan and Mahalakshmi (2009)
Rapeseed	37.600	914.00	39.50	Ramadhas *et al.* (2005)
	36.995	912.00[b]	23.91	Qi *et al.* (2014)
	37.620	914.00	39.50[c]	Altin *et al.* (2001)
	36.890	918.00	-	Nwafor *et al.* (2000)
Rice bran	39.500	916.30	44.52	Agarwal *et al.* (2008)
Rubber seed	37.500	910.00	66.2	Ramadhas *et al.* (2005)
Soapnut	38.207	904.00	46.42	Misra and Murthy (2011)
Soybean	39.600	920.00	65.00	Ramadhas *et al.* (2005)
	37.000	925.00[a]	33.00	Rakopoulos *et al.* (2006)
	39.623	914.00	65.00[c]	Altin *et al.* (2001)
	39.623	914.00[f]	33.10	Shehata *et al.* (2015)
Sunflower	36.500	920.00[a]	34.00	Rakopoulos *et al.* (2006)
	39.525	918.00	58.00[c]	Altin *et al.* (2001)
	39.500	923.10[a]	34.20	Karaosmanoglu *et al.* (2000)
	39.500	918.00	58.00	Ramadhas *et al.* (2005)
Waste cooking oil	37.680	900.00	52.00[e]	Krishnamoorthy *et al.* (2018)
	38.590	904.40[a]	49.05	Milano *et al.* (2018)

a, *b*, *c*, *d*, *e* and *f* at the temperature of 15, 20, 27, 28, 30 and 40 °C respectively.

The VO is emerging as an alternative fuel to replace/supplement diesel in CI engines. Most of the investigations recommends the potential application of VOs in CI engines. The VO run engine show a lower performance in comparison to a diesel run engine due to the inferior quality of former against the later. VO has the characteristics of high density and viscosity with lower calorific value in comparison to the diesel. To improvise the performance of CI engine using VO, various methods are adopted which are shown in shown in Figure 2.1.

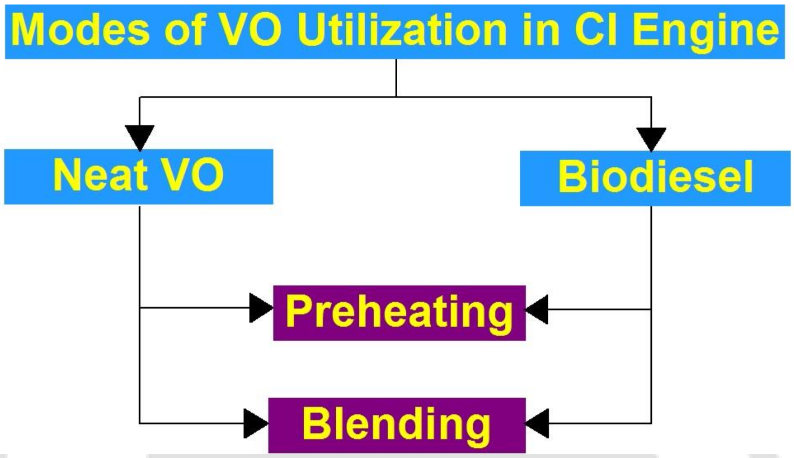


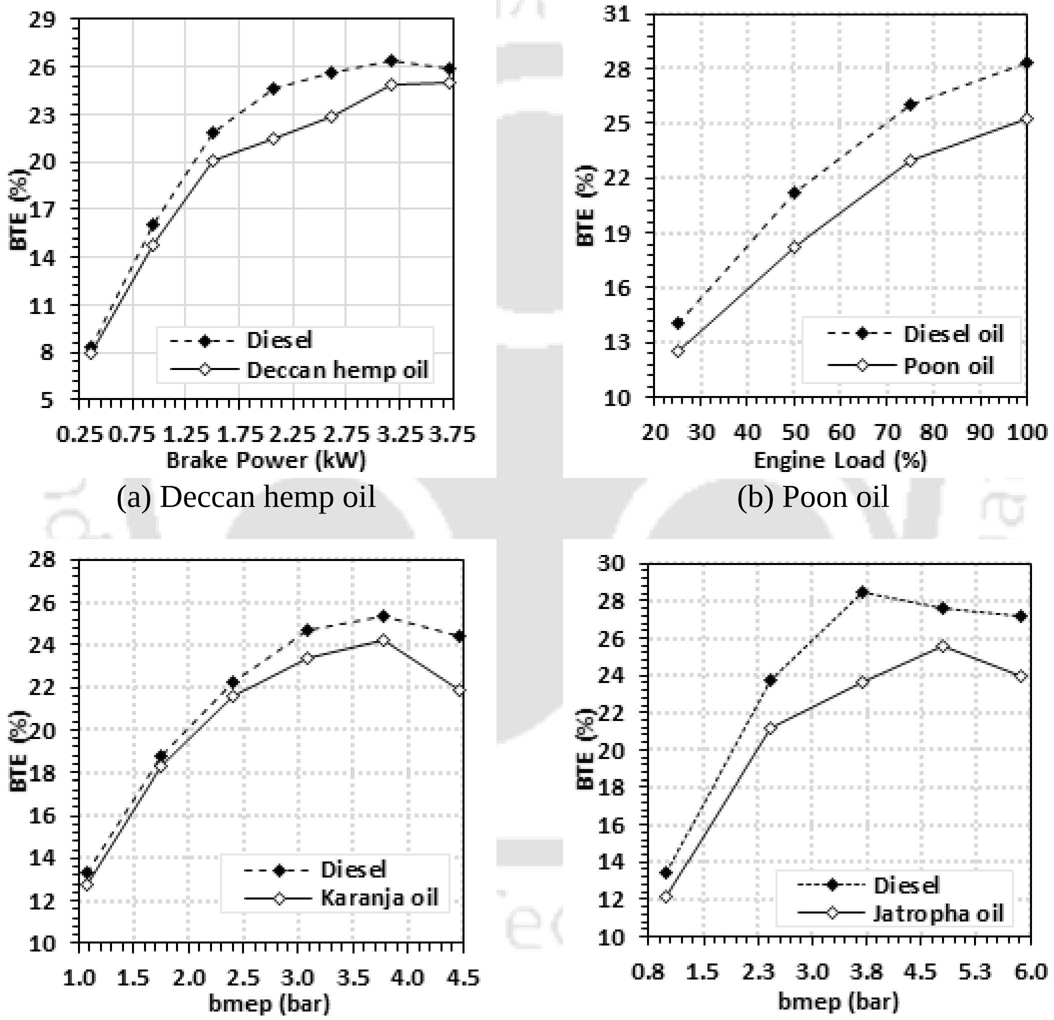
Figure 2.1 Modes of VO utilization in engine.

2.2 Impacts of Neat VO in CI Engines

Many VOs, in their neat form, have been tried and tested in diesel engines to study their effect on the performance, emissions and combustion of the engines. Altin *et al*. (2001) uses the oil, derived from sunflower, cotton seed, soybean, corn, opium poppy and rapseed, to study the effect of these oils on the performance and emissions of a diesel engine. Jatropha oil was used in the studies of Pramanik (2003); Chauhan *et al*. (2010); and Singh (2013). While Almeida *et al*. (2002); Hebbal *et al*. (2006); Cetin and Yuksel (2007); Agarwal and Rajamanoharan (2009); Devan and Mahalakshmi (2009) and Shah and Ganesh (2016) respectively used the oil derived from palm, deccan hemp, hazelnut, karanja, poon, and karanja with sunflower. The long term test of the CI engine conducted under part load condition and speed of 1600 rpm for 50 hours by Karaosmanoglu *et al*. (2000) was fuelled by sunflower oil. Rapeseed oil was used in the investigation of Nwafor *et al*. (2000) to study the effect of advanced injection timing. Purushothaman and Nagarajan (2009) used orange oil to study the performance, emissions and combustion of CI engine. The outcome of these studies has been summarized in the following sub-sections.

2.2.1 Brake Thermal Efficiency

The use of neat VO in the engine results in the drop of engine's brake thermal efficiency (BTE). This drop in BTE varies with feedstock used. The use of deccan hemp, poon, karajna and jatropha oil respectively leads to drop of 3-13%, 11-14%, 3-11% and 7-12% BTE of the engine (Figure 2.2). This drop in BTE is due to the combined effects of lower calorific value, high viscosity and poor volatility of VO, in comparison to diesel, resulting in the poor combustion characteristics (Pramanik 2003; Hebbal *et al*. 2006; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010; Shah and Ganesh 2016).



(a) Deccan hemp oil (b) Poon oil

(c) Karanja oil (d) Jatropha oil

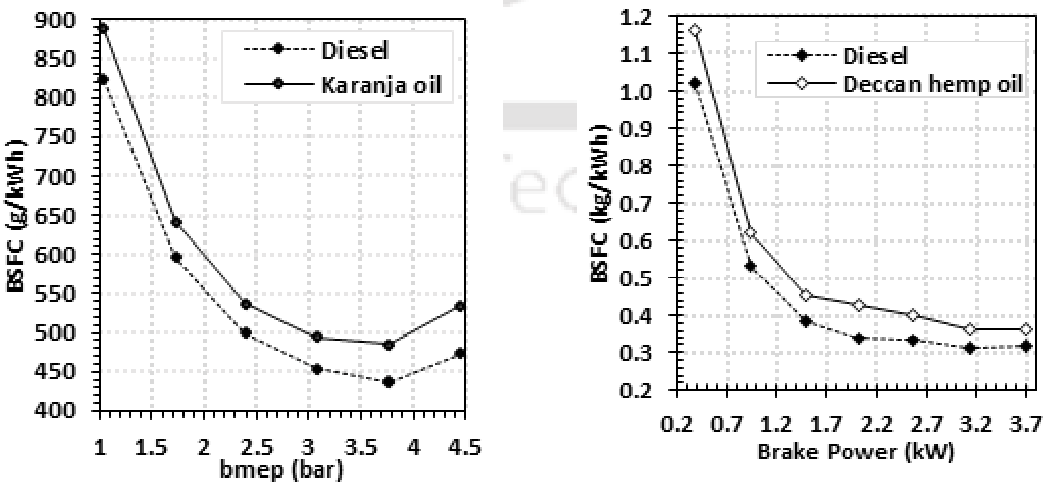
Figure 2.2 Effect of different vegetable oils on the BTE of the engine (Hebbal *et al*. 2006; Devan and Mahalakshmi 2009; Agarwal and Rajamanoharan 2009; Chauhan *et al*. 2010)

The higher viscosity of VO results in poor atomization and larger fuel droplets size in the fuel spray which leads to inadequate mixing of oil droplets and heated air (Agarwal and

Rajamanoharan 2009; Singh 2013). However, in some studies (Nwafor 2003; Yilmaz and Morton 2011), the engine BTE is found to be slightly higher or equal to diesel when neat VO is used as a fuel. According to Nwafor (2003), the viscosity of VO might have acted as a lubricant, as well as a sealant between the piston rings and cylinder wall falsifying the engine compression and leading to an efficient combustion in diesel engines.

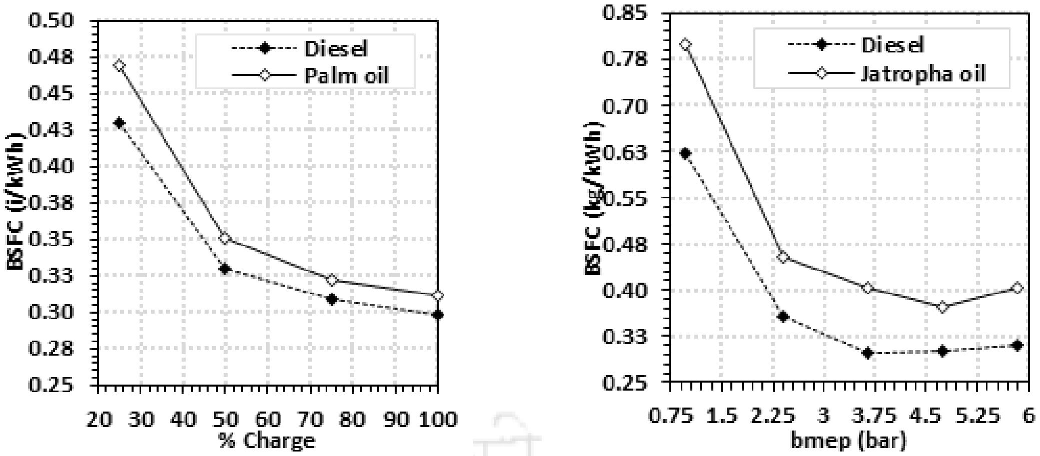
2.2.2 Brake Specific Fuel Consumption

The brake specific fuel consumption (BSFC) generally decreases with increasing engine load irrespective of fuel used. This is due to the relatively lower heat loss with the increasing engine loads (Pradhan *et al*. 2014). The comparative BSFC of the typical vegetable oils against the diesel has been shown in Figure 2.3 and 2.4. It follows a similar trend to that of the diesel with higher quantity across the all operating range of the engine. The BSFC of engine goes up to 31% higher than diesel when VO was used. It is an obvious that when a fuel with lower heating value (such as VO) is used in the engine, it bound to consume more quantity of fuel to produce same unit of power in compared with higher heating value fuel (such as diesel). Therefore, the VOs need larger mass of fuel to maintain the constant energy input to the engine (Almeida *el al*. 2002; Devan and Mahalakshmi 2009; Singh 2013; Shah and Ganesh 2016). Along with the lower heating value, the higher density and viscosity of VO is also responsible for higher BSFC. The higher density of VO leads to a greater fuel flow rate for the same displacement of the plunger in the fuel injection pump leading to an increase in BSFC (Pramanik, 2003; Agarwal and Rajamanoharan 2009; Chauhan *et al*. 2010).



(a) Karanja oil (b) Deccan hemp oil

Figure 2.3 Comparative BSFC of vegetable oil against diesel (Agarwal and Rajamanoharan 2009; Hebbal *et al*. 2006).

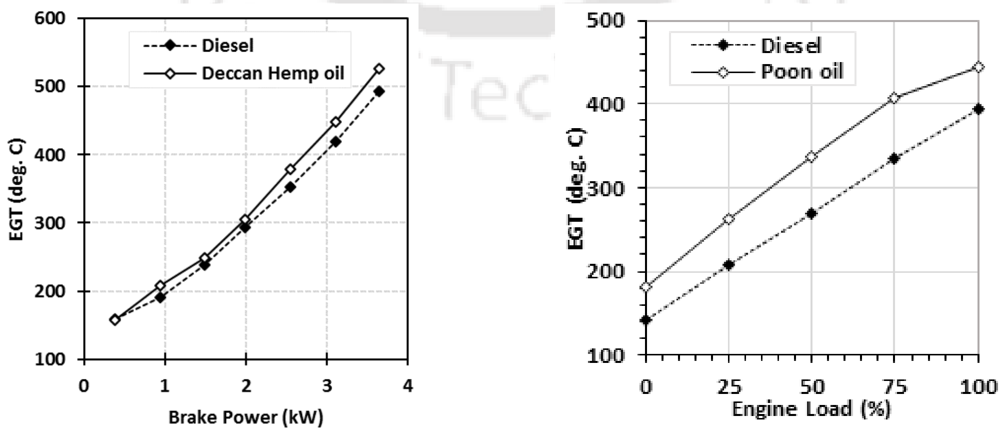


(a) Palm oil (b) Jatropha oil

Figure 2.4 Comparative BSFC of vegetable oil against diesel (Almeida *et al.* 2002; Chauhan *et al.* 2010)

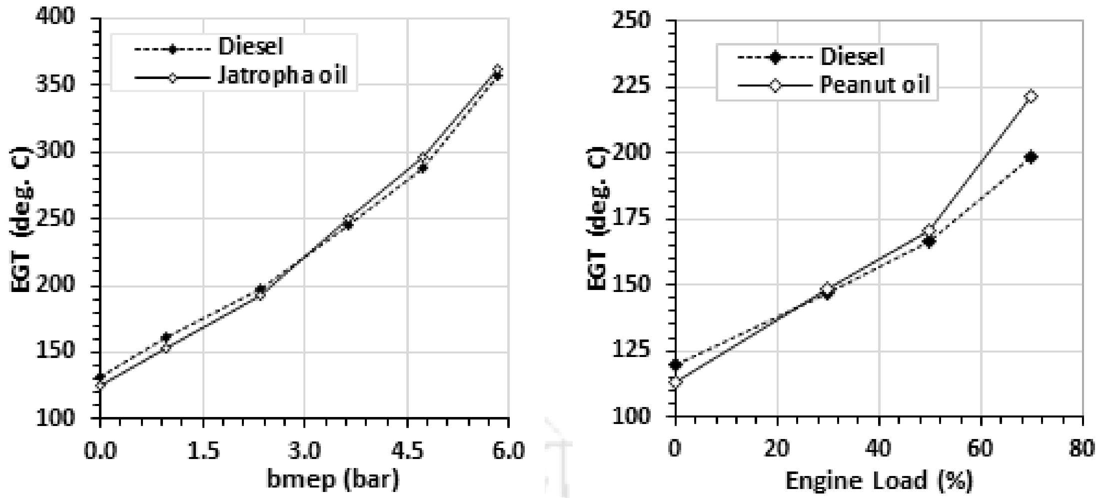
2.2.3 Exhaust Gas Temperature

Higher exhaust gas temperature (EGT) is the indicative of the lower thermal efficiencies of the engine as lesser amount of the energy input in the fuel is converted to work (Chauhan *et al.* 2010). In most cases, the VO as a fuel in engine gives higher EGT in comparison to diesel (Figure 2.5 and 2.6). The poor combustion characteristics of VO results in the higher EGT due to the high viscosity (Pramanik 2003; Chauhan *et al.* 2010). However, Hebbal *et al.* (2006) believed that the low volatility of VO affects the spray formation in combustion chamber leading to slow combustion. Devan and Mahalakshmi (2009) further added that this higher EGT may be due to the presence of constituents with higher boiling points in poon oil (VO), than in diesel, which might not have well evaporated at the time of main combustion phase and this continued to burn in the late combustion phase, thereby increasing the EGT.



(a) Deccan hemp oil (b) Poon oil

Figure 2.5 Comparative EGT of vegetable oil against diesel (Hebbal *et al.* 2006; Devan and Mahalakshmi 2009).

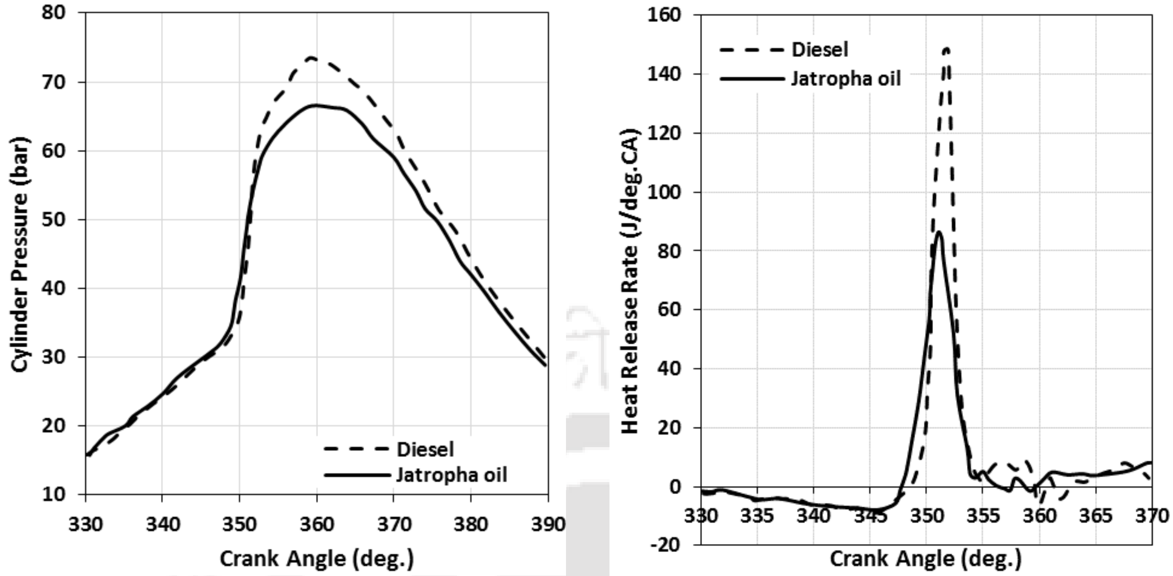


(a) Jatropha oil (b) Peanut oil

Figure 2.6 Comparative EGT of vegetable oil against diesel (Chauhan *et al.* 2010; Yilmaz and Morton 2011)

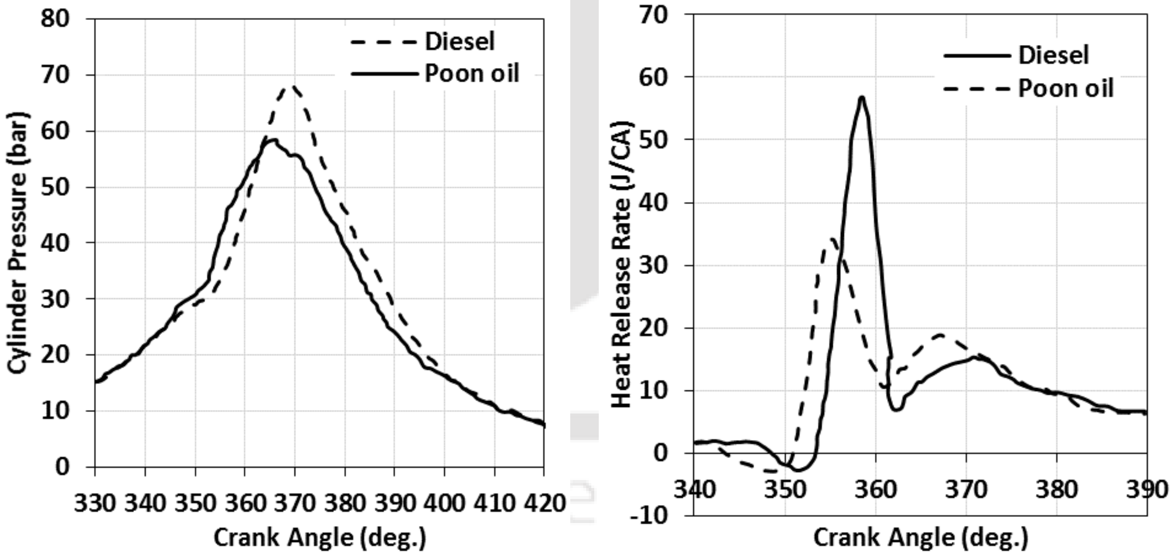
2.2.4 Combustion Analysis

The combustion studies on the use of neat VOs in the diesel engine carried out by (Nwafor and Rice 1996; Devan and Mahalakshmi 2009; Kasiraman *et al.* 2012; Nwafor 2003; Pradhan *et al.* 2014) indicates a lower cylinder pressure and heat release rate of neat VOs as compared to that of the diesel. This has been attributed to the higher viscosity of VOs compared to the diesel. Higher viscosity leads to the poor atomization of fuel and it reduces air entrainment and air-fuel mixing rates (Devan and Mahalakshmi 2009; Kasiraman *et al.* 2012; Pradhan *et al.* 2014). Figure 2.7 illustrates the variation of cylinder pressure and heat release rate with crank angle (CA) when jatropha oil was used in the engine (Pradhan *et al.* 2014). With the use of the neat jatropha oil, the peak cylinder pressure (PCP) reduces from 73.65 to 66.75 bar at full engine load while the peak rate of pressure rise (RoPR) decreases from 13.09 to 8.08 bar/deg.CA. The peak heat release rate for diesel and jatropha oil is 148.4 and 86.5 J/deg.CA respectively. Similar trend of reduced PCP and heat release rate is also being reported with the use of neat poon oil in the study of Devan and Mahalakshmi (2009) as shown in Figure 2.8. The PCP reduces from 67.5 bar to 60 bar at the full load of the engine. However, Shah and Ganesh (2016) reported a higher cylinder pressure and heat release rate (HRR) with the use of sunflower and karanj oil in compared to that of the diesel. This observation has been credited to the presence of higher amount of unsaturated FAs in sunflower and karanj which results in the longer ignition delay (ID) and allowing more time to form a flammable air fuel mixture leading to higher HRR and cylinder pressure. The higher bulk modulus and viscosity of VO has also been attributed as it advances the injection timing leading to higher cylinder pressure.



(a) Cylinder pressure variation with CA. (b) Heat release rate variation with CA.

Figure 2.7 Variation of cylinder pressure and heat release rate at full load (Pradhan *et al.* 2014)



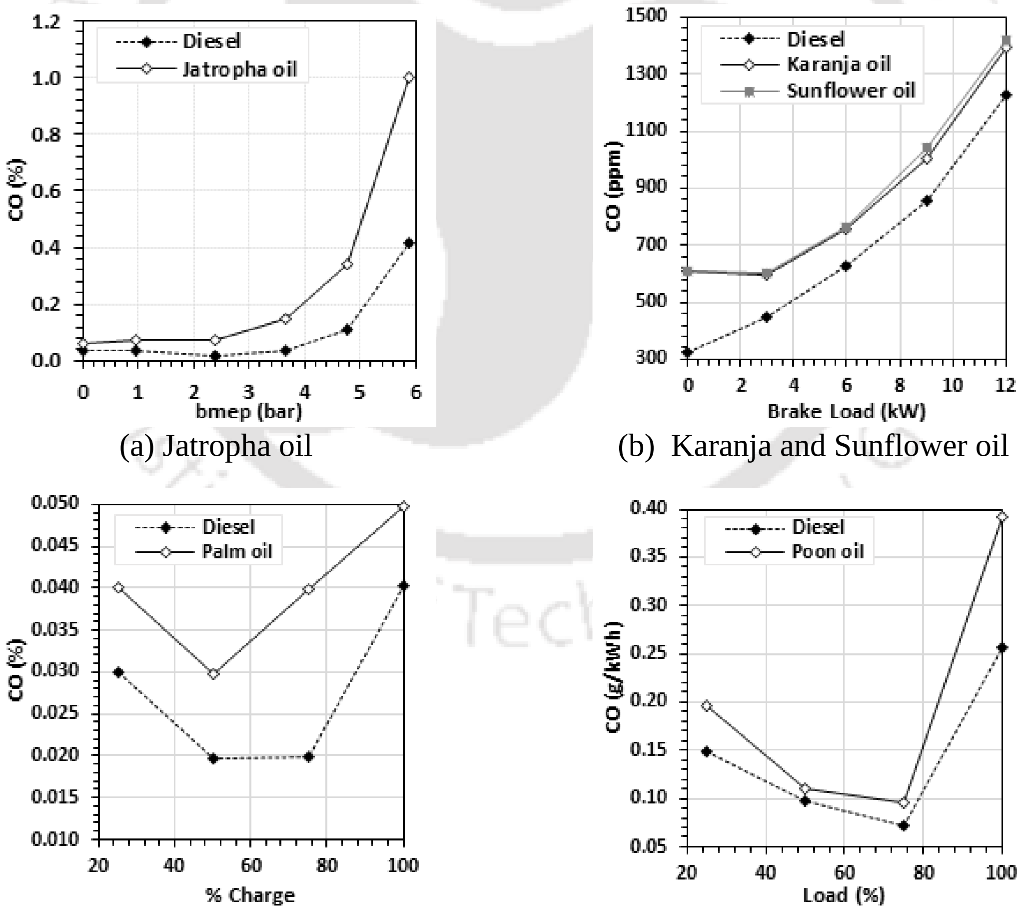
(a) Cylinder pressure variation with CA. (b) Heat release rate variation with CA.

Figure 2.8 Variation of cylinder pressure and heat release rate at full load (Devan and Mahalakshmi 2009).

2.2.5 Carbon Monoxide Emissions

The presence of carbon monoxide (CO) emissions in the exhaust gases indicates a loss in chemical energy of the fuel which is not being utilized fully to develop the engine power

(Shah and Ganesh 2016). The CO emissions in the engine increases with the increase in the engine load/bmep as shown in Figure 2.9. At higher loads, richer fuel-air mixture is burned resulting in more CO production (Almeida *el al*. 2002; Agarwal and Rajamanoharan 2009). However, in some investigations, it decreases in the intermediate and maximum loads, and they have not cited reason(s) for as such (Almeida *el al*. 2002; Devan and Mahalakshmi 2009). Neat VO increases the CO emissions through the operation range as compared to that of the diesel. The viscosity of VO has been considered as the factor responsible it. The higher viscosity of oil creates difficulty in fuel atomization leading to locally rich mixtures. This rich mixture results in an incomplete combustion and produces CO due to lack of oxygen (Almeida *el al*. 2002; Wang et al. 2006; Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010; Shah and Ganesh 2016). Shah and Ganesh (2016) further added the fact that spray cone angle gets reduced due to the higher viscosity of VO. This reduction in spray cone angle leading to less air entrainment results in an incomplete combustion and ultimately the CO formation.



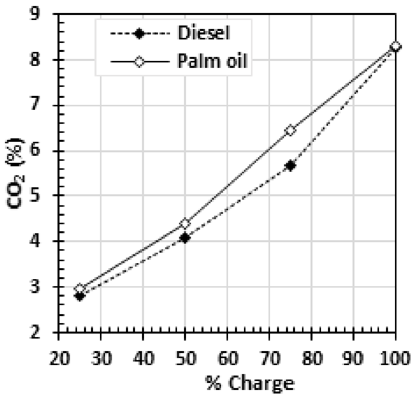
(a) Jatropha oil (b) Karanja and Sunflower oil

(c) Palm oil (d) Poon oil

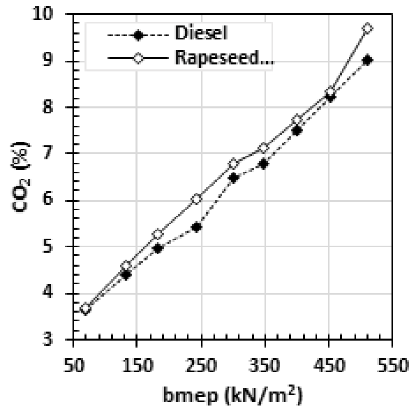
Figure 2.9 Trends of CO emissions for vegetable oil against diesel in CI Engine (Chauhan *et al*. 2010; Shah and Ganesh 2016; Almeida *et al*. 2002; Devan and Mahalakshmi 2009).

2.2.6 Carbon Dioxide Emissions

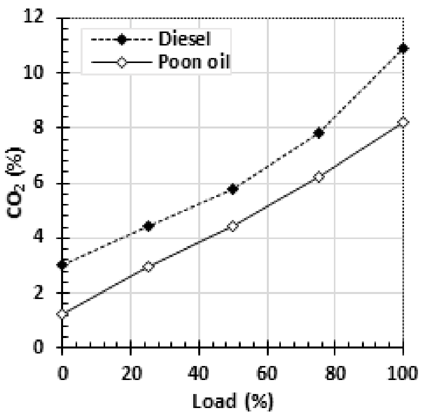
In the ideal condition, the combustion of fuel results in the heat, carbon dioxide (CO_2) and water vapour as the product, otherwise results in emissions of other gases along with CO_2. Therefore, CO_2 emissions bounds to increase with the increase in the engine load as more fuel is burned to support the increasing load (Figure 2.10) irrespective of type of fuel used. In comparison to diesel, the use of VO in engines, CO_2 emissions decreases in some cases (Wang *et al.* 2006; Devan and Mahalakshmi 2009; Singh *et al.* 2010); while it increases in most of the cases throughout the entire engine load (Almeida *el al.* 2002; Nwafor 2004; Agarwal and Agarwal 2007; Chauhan *et al.* 2010). According to Wang *et al.* (2006) and Devan and Mahalakshmi (2009), the lower in CO_2 emissions from VO (as compared to the diesel) is due to the relatively lower carbon content of the VO in the same volume of fuel consumed at the same engine load. However, for the higher CO_2 emissions from VO (as compared to diesel), authors have not pointed out any clear reasons for such cause.



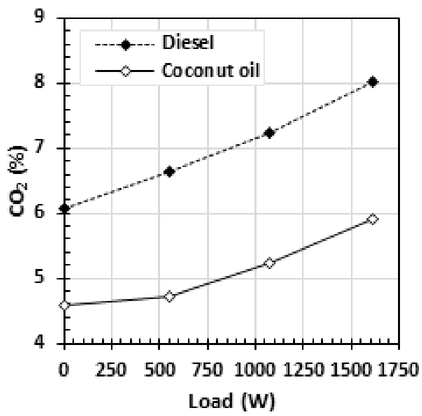
(a) Palm oil



(b) Rapeseed oil



(c) Poon oil



(d) Coconut oil

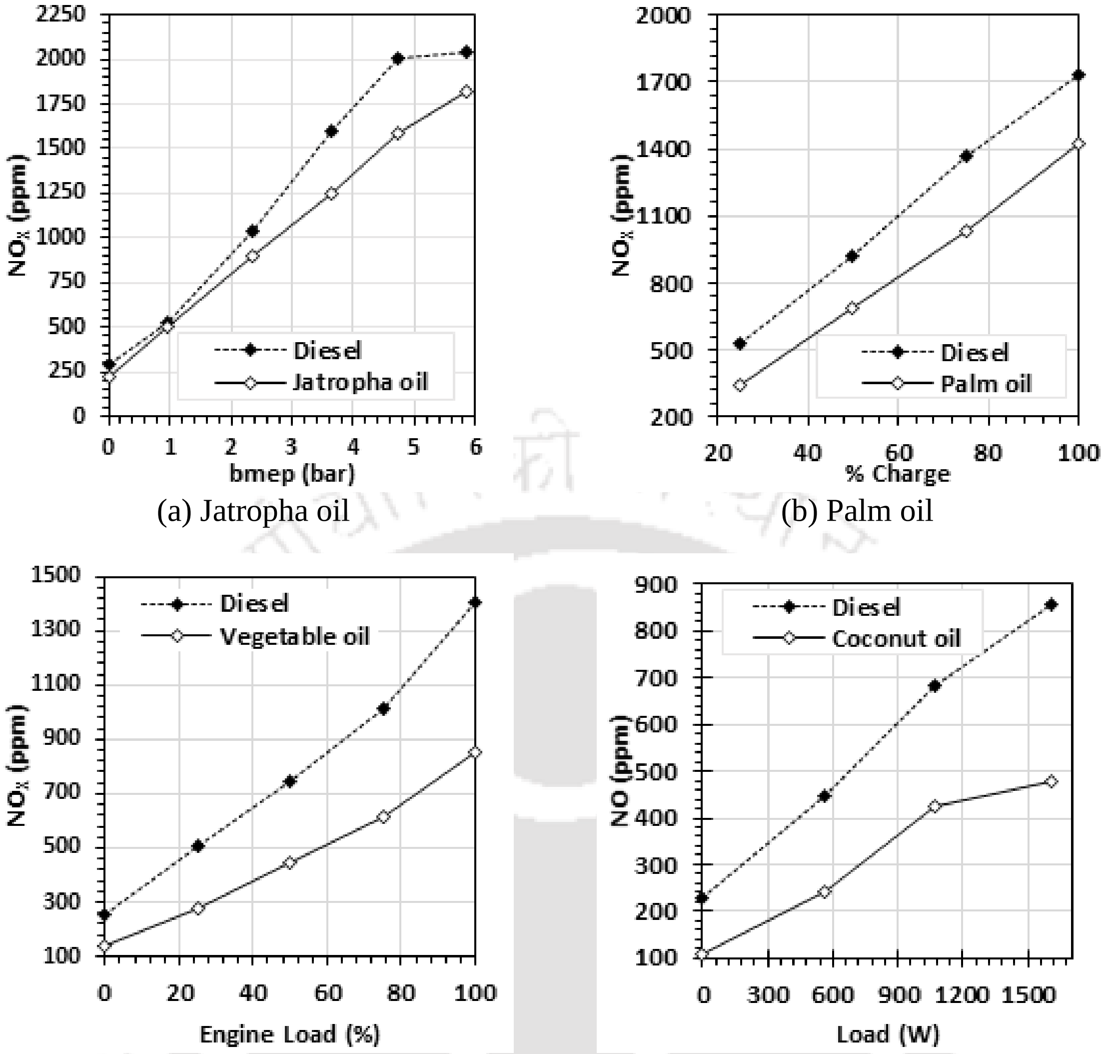
Figure 2.10 Trends of CO_2 emissions for vegetable oil against diesel in CI engine (Almeida *el al.* 2002; Nwafor 2004; Devan and Mahalakshmi 2009; Singh *et al.* 2010)

2.2.7 Oxides of Nitrogen

According to Huang *et al*. (2009), the primary sources of oxides of nitrogen (NO_X) in the combustion processes consists of thermal NO_X, fuel NO_X and prompt NO_X. Thermal NO_X, which is highly temperature dependent and formed through high temperature oxidation of nitrogen in combustion chamber, is the most recognized and relevant source from engine combustion. NO_X emissions in the engine increases with the increasing load (Almeida *el al*. 2002; Wang *et al*. 2006; Huang *et al*. 2009; Chauhan *et al*. 2010). With increased load more fuel is injected and combusted in the cylinder which results in higher gas temperature. This higher gas temperature leads to more NO_X formation in the engine cylinder and then higher NO_X emissions from the engine. The rate of NO_X formation in the engine is primarily a function of combustion (flame) temperature, the residence time of nitrogen at that temperature, and the contents of oxygen in the reaction regions in the combustion chamber (Huang *et al*. 2009).

In comparison to diesel, the use of VO in the engine reduces the NO_X emissions as shown in Figure 2.11. The lower heating value of VO is believed to cause of this reduction (Wang *et al*. 2006; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010). On the basis of oil viscosity, Agarwal and Rajamanoharan (2009) further elaborate that the higher viscosity of VO is expected to have larger fuel droplet size than diesel that will have a longer combustion duration with significant energy release during the late burning phase. Because of this, the peak combustion chamber temperature is possibly lower due to lower heat release in the pre-mixed combustion phase as well as mixing controlled combustion phase, leading to lower formation and emissions of NO.

However, in the investigations of Yilmaz and Morton, (2011), and Shah and Ganesh (2016), the NO_X emissions was found to increase with the use of VO derived from peanut, sunflower and karanja. Yilmaz and Morton (2011) opined that most VOs contain small quantities of nitrogen containing proteins which in addition to atmospheric nitrogen releases extra NO_X emissions through combustion which might be a contributing factor for VOs to have higher NO emissions than diesel fuel. While Shah and Ganesh (2016) attributed to 13.35% and 9.34% oxygen content of sunflower and karanja oil respectively which could have helped in improving oxidation of nitrogen available during combustion process. This leads to an increase in the combustion bulk temperature that is responsible for thermal NO_X formation. They further added that higher bulk modulus of elasticity of vegetable oils (sunflower and karanja) leads to an advanced injection timing which also contributes a higher NO_x emissions.



(a) Jatropha oil (b) Palm oil

(c) Vegetable oil (d) Coconut oil

Figure 2.11 Trends of NO_X emissions for vegetable oil against diesel in CI engine (Chauhan *et al*. 2010; Almeida *el al*. 2002; Wang *et al*. 2006; Singh *et al*. 2010)

2.2.8 Hydrocarbon Emissions

The hydrocarbon (HC) emitted from the incomplete combustion of fuel are lower in partial engine load and it increases with higher loads. This is because of the availability of relatively less oxygen for the reaction as more fuel is injected into the engine cylinder at higher engine loads (Almeida *el al*. 2002; Wang *et al*. 2006; Cetin and Yuksel 2007; Agarwal and Rajamanoharan 2009; Chauhan *et al*. 2010). The comparative illustration of HC emissions in the engine when diesel and VO are used as fuels is shown in Figure 2.12. It reflects significant amount of HC emissions when VO is used in the engine in comparison to diesel. Hebbal *et al*. (2006) believed that the low volatility of VO might have affected the spray formation in the combustion chamber leading to slow combustion and HC formation.

According to Devan and Mahalakshmi (2009), the higher viscosity of VO is responsible for higher HC emissions. The higher fuel viscosity may lead to higher fuel spray droplet size which affects the fuel spray quality. In some exceptional investigations, HC emissions is found to be lower when VO is used. However, they have not substantiated any support for their findings.

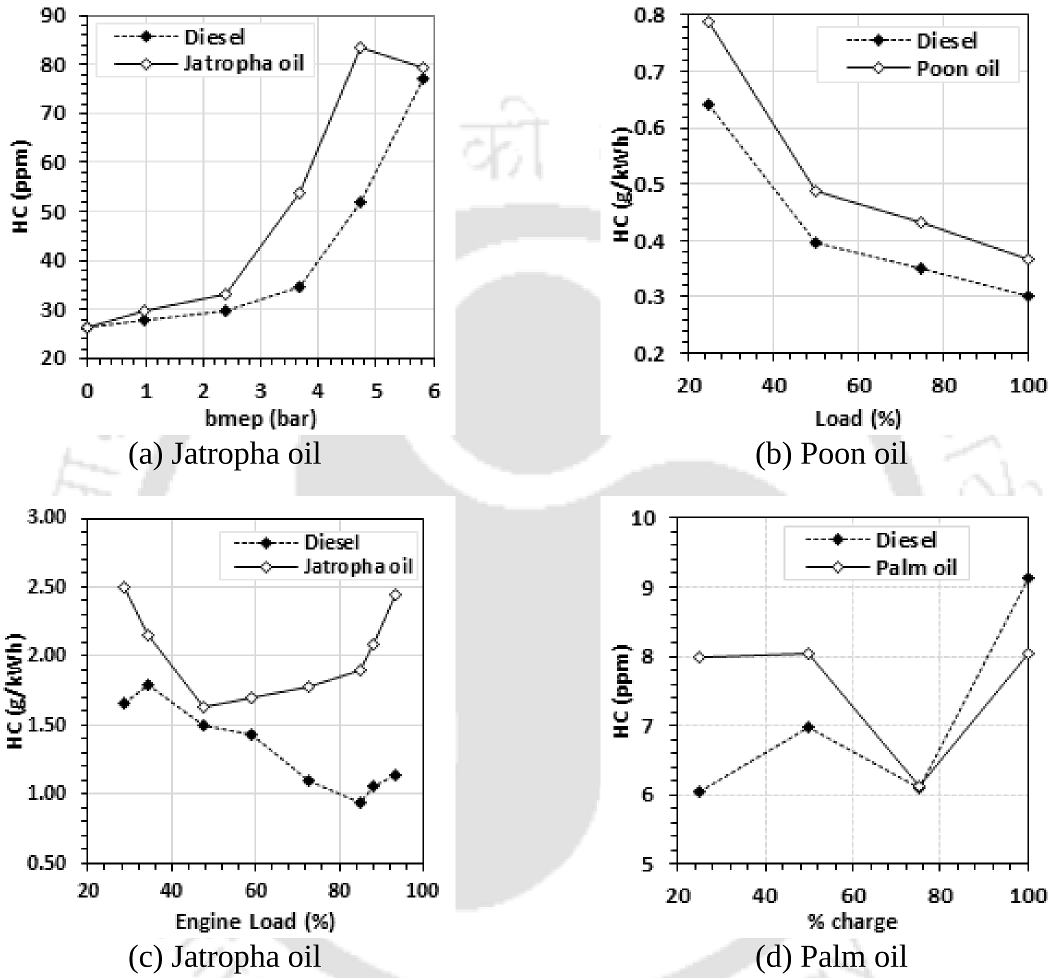


Figure 2.12 Trends of HC emissions for vegetable oil against diesel in CI engine (Chauhan *et al*. 2010; Devan and Mahalakshmi 2009; Agarwal and Agarwal 2007; Almeida *et al*. 2002)

2.3 Effects of Preheated Neat VO in CI Engines

The preheating of VO before its injection into the engine combustion chamber is done to reduce the viscosity of oil. The viscosity of oil drastically gets reduced with preheating. Figure 2.13 shows the variation in the kinematic viscosity of VO and diesel at elevated temperature (Agarwal and Agarwal 2007; Hazar and Aydin 2010; Sonar *et al*. 2015). The kinematic viscosity of mahua, rapeseed and jatropha oil is reduced by 81, 73 and 86%, respectively when the oil is heated from 40 to 100 °C. However, diesel shows a little variation in the same range of temperature. In the most of investigations, the VOs have been preheated in the range of temperature from 70 to 90 °C. However, the optimal value of this preheated temperature range could not have been ascertained as mostly the studies have been done on a single fixed preheated temperature.

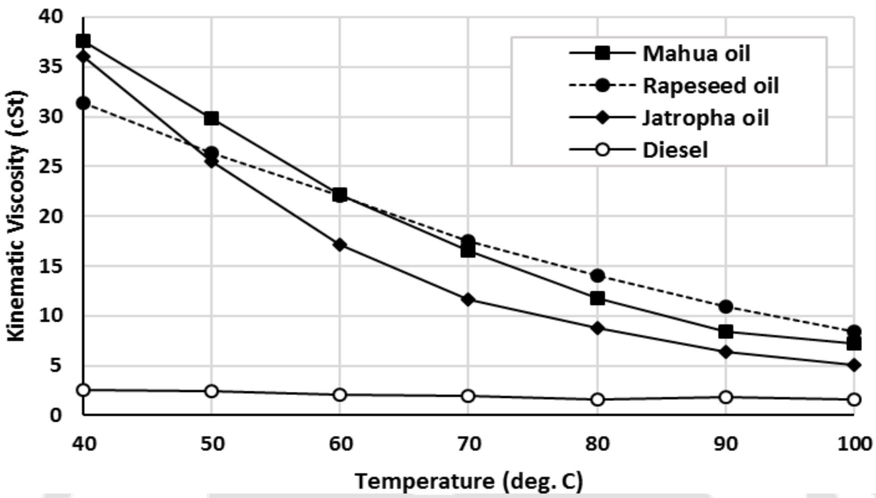
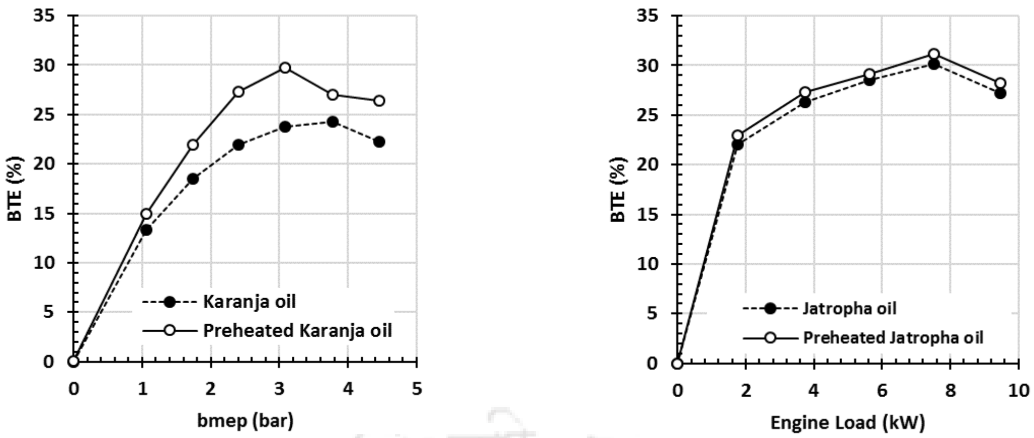


Figure 2.13. Effect of preheating temperature on the kinematic viscosity of the fuel (Sonar *et al*. 2015; Agarwal and Agarwal 2007; Hazar and Aydin 2010)

2.3.1 Brake Thermal Efficiency

The BTE of engine is found to improve with the use of preheated VO in comparison to the one without preheating. However, it remains to be lower as compared to diesel. In Figure 2.14**,** the results of neat and preheated karanja and jatropha oil are illustrated (Agarwal and Rajamanoharan 2009; Singh 2013). Preheating shows improved results in the intermediate and higher engine loads. Preheating of karanja and jatropha oil respectively results in 11 to 24% and 2 to 4% higher BTE as compared to their respective neat oil. The preheating of oil reduces the viscosity which facilitates better atomization of fuel particles ensuring better combustion and improved BTE (Pugazhvadivu and Jeyachandran 2005; Agarwal and Agarwal 2007; Yilmaz and Morton 2011; Singh 2013; Sonar *et al*. 2015). In the studies of Agarwal and Rajamanoharan (2009), the preheating of VO leads to higher BTE than that of diesel. This is because of the reduction in the viscosity and increase in volatility along with the oxygen content of VO that gives better fuel combustion which improves the BTE. However, in the study of Nwafor (2003), the use of preheated oil deteriorates the engine BTE.

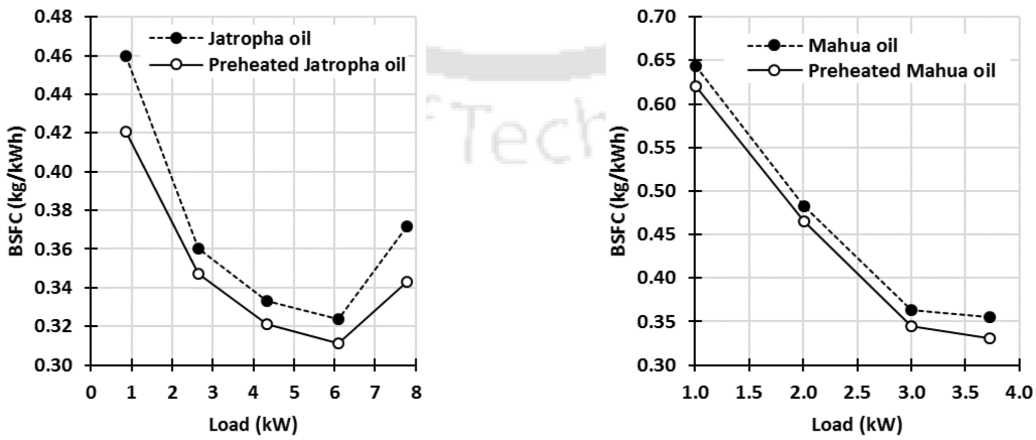


(a) Karanja oil (b) Jatropha oil

Figure 2.14 Effects of preheated vegetable oil on BTE of the engine. (Agarwal and Rajamanoharan 2009; Singh 2013)

2.3.2 Brake Specific Fuel Consumption

The preheating of VO results in the lower BSFC as compared to neat VO (Agarwal and Rajamanoharan 2009; Singh 2013; Agarwal and Agarwal 2007; Sonar *et al*. 2015). The typical results when jatropha and mahua oil have been used is shown in Figure 2.15 (Sonar *et al*. 2015; Singh 2013). The reduction in BSFC is in the range of 3 to 8% and 4 to 7%, respectively when preheated jatropha and mahua oil are used in the engine. This reduction in BSFC is due to preheating which reduces the oil viscosity leading to better combustion in the engine (Sonar *et al*. 2015). However, it is found to be higher in the study of Nwafor (2003) when preheated rapeseed oil is used. Pradhan *et al*. (2014) have found it to be lower at 25 and 50% of engine load and relatively higher at engine load of 75 and 100%.

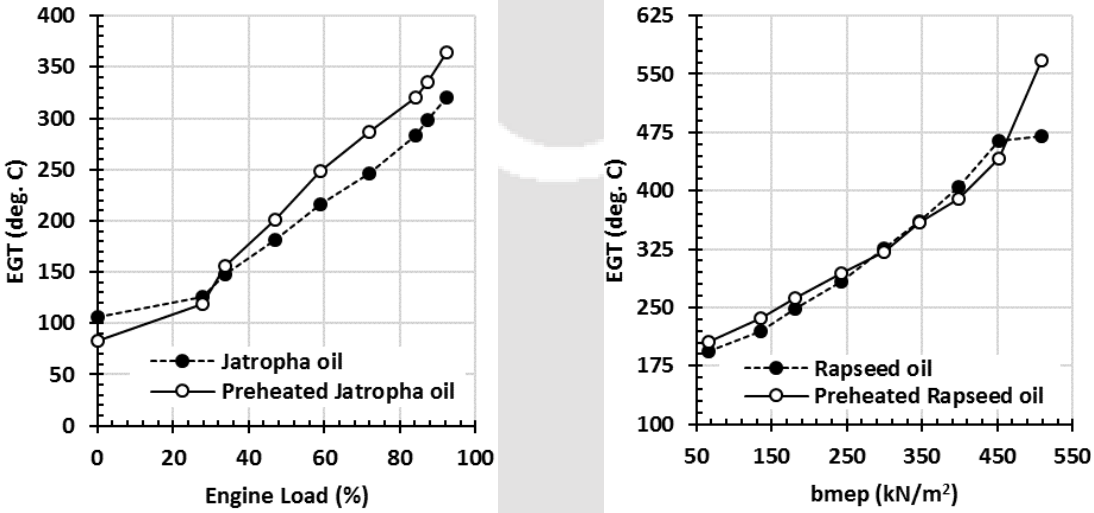


(a) Jatropha oil (b) Mahua oil

Figure 2.15 Effects of preheated vegetable oil on BSFC of the engine (Singh 2013; Sonar *et al*. 2015)

2.3.3 Exhaust Gas Temperature

The EGT is generally found to be higher when preheated VO is used in the engine as compared to the neat VO (Nwafor 2004; Pugazhvadivu and Jeyachandran 2005; Agarwal and Agarwal 2007; Yilmaz and Morton 2011; Sonar *et al*. 2015; Hazar and Sevinc 2019). The typical results of preheated jatropha and rapeseed oil against the neat oil is shown in Figure 2.16 (Nwafor 2004; Agarwal and Agarwal 2007)**.** The EGT increases by an average of around 31 °C when preheated jatropha oil is used in the engine, while preheated rapseed oil shows a very little variation. The increase in combustion gas temperature due to preheating of oil (as it increases the fuel temperature) is the probable cause for the increase in EGT (Pugazhvadivu and Jeyachandran 2005; Sonar *et al*. 2015).

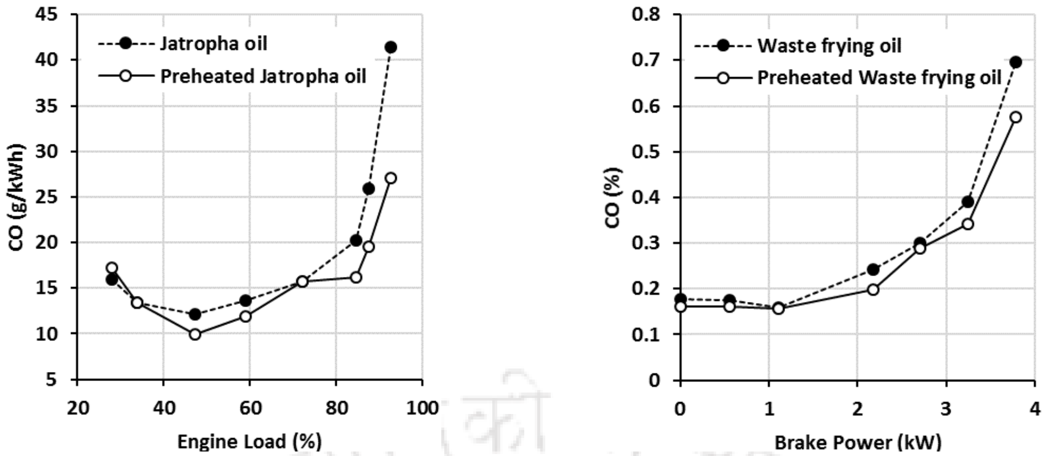


(a) Jatropha oil (b) Rapeseed oil

Figure 2.16 Effects of preheated vegetable oil on EGT of the engine (Agarwal and Agarwal 2007; Nwafor 2004)

2.3.4 Carbon Monoxide Emissions

The effect of preheated VO, jatropha and waste frying oil, vis-à-vis neat oil on the CO emissions has been illustrated in Figure 2.17 (Pugazhvadivu and Jeyachandran 2005; Agarwal and Agarwal 2007). It indicates a reduction in the CO emissions with the preheating especially at higher engine loads (Hazar and Sevinc 2019). The preheating reduces the oil viscosity that results in a better fuel atomization. This improves the spray characteristics and an improved fuel air mixing. This ultimately leads to better combustion and reduced CO emissions (Pugazhvadivu and Jeyachandran 2005; Pradhan *et al*. 2014).

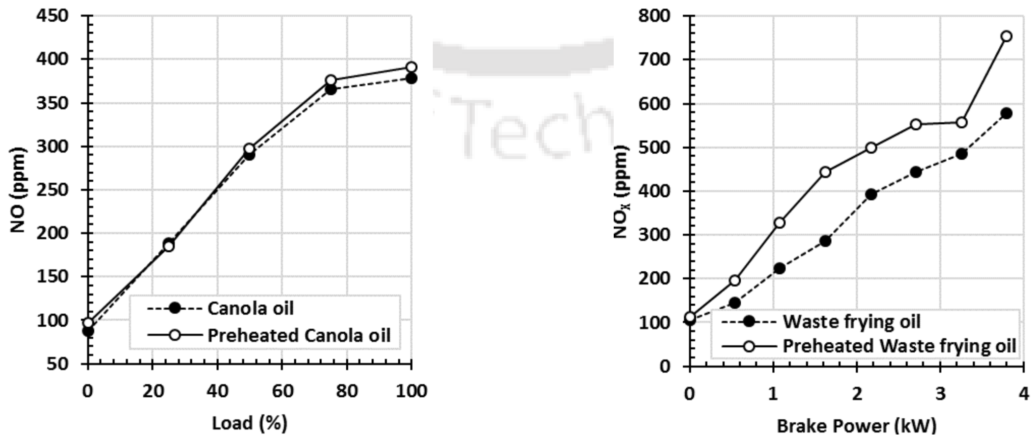


(a) Jatropha oil **(b)** Waste frying oil

Figure 2.17 Effects of preheated vegetable oil on CO emissions of the engine (Agarwal and Agarwal 2007; Pugazhvadivu and Jeyachandran 2005)

2.3.5 Oxides of Nitrogen

The effect of preheated canola and waste frying oil on NO_X emissions is presented in Figure 2.18 (Yilmaz and Morton 2011; Pugazhvadivu and Jeyachandran 2005). It reflects a relatively higher NO_X in the exhaust when preheated VO is used. Similar results have also been reported (Hazar and Sevinc 2019). Agarwal and Rajamanoharan (2009) also reported a higher NO emissions with the preheated karanja oil. However, the mass of NO emissions decreases with the increased engine load with lowest emissions at the highest load of the engine. The increase in combustion gas temperature with the preheated fuel may be attributed to the increase of NO_X (Pugazhvadivu and Jeyachandran 2005).



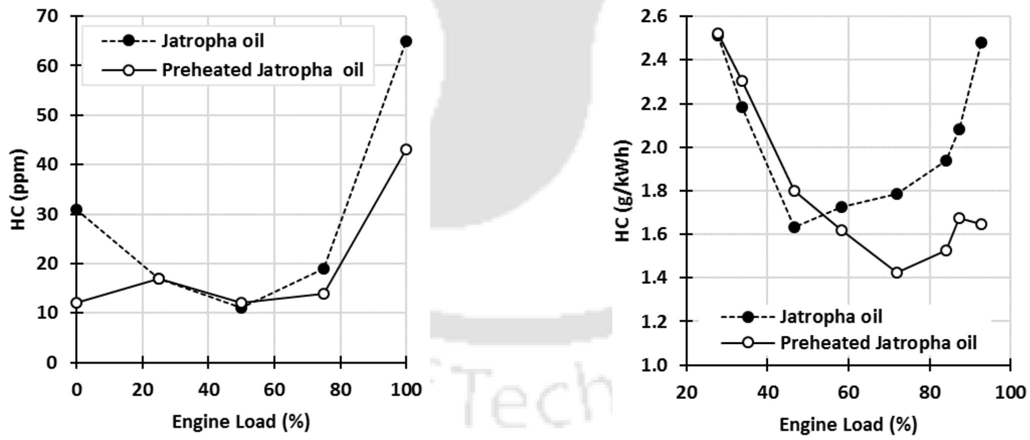
(a) Canola oil (b) Waste frying oil

Figure 2.18. Effects of preheated vegetable oil on NO_X emissions of the engine (Yilmaz and Morton 2011; Pugazhvadivu and Jeyachandran 2005)

In the investigation of Pradhan *et al*. (2014), NO_X emissions is found to be higher at 0 and 25% of engine loads; while it is found to be lower at loads of 50, 75 and 100% with preheated jatropha oil as compared to the neat oil. This lower NO_X emissions at the higher engine load has been attributed to the instantaneous chemical reaction and low air-fuel ratio of preheated oil.

2.3.6 Hydrocarbon Emissions

In comparison to the neat VO, the engine driven on the preheated VOs releases lower unburned HC (Agarwal and Agarwal 2007; Pradhan *et al*. 2014; Sonar *et al*. 2015; Hazar and Sevinc 2019). Typical results illustrated in Figure 2.19 (Agarwal and Agarwal 2007; Pradhan *et al*. 2014) indicate that preheating effectively mitigates the HC emissions at higher engine loads. It reduces by up to 34% when preheated jatropha oil is used. This reduction in HC emissions may be due to more complete and cleaner combustion with better atomization of fuel molecules because of the preheated oil (Pradhan *et al*. 2014). Nwafor (2004) reported a higher HC emissions with the preheated oil as compared to the neat oil while using rapeseed oil.



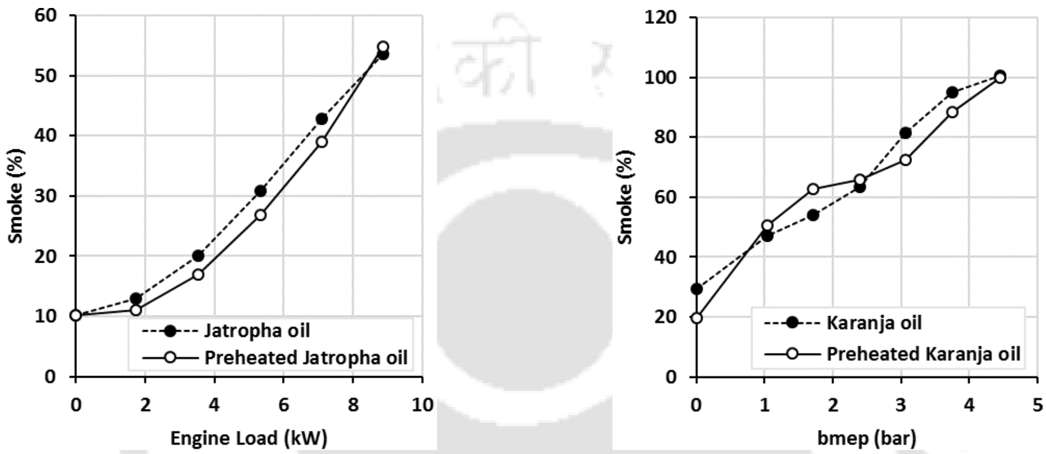
(a) Jatropha oil (b) Jatropha oil

Figure 2.19 Effects of preheated vegetable oil on HC emissions of the engine (Pradhan *et al*. 2014; Agarwal and Agarwal 2007)

2.3.7 Smoke Emissions

As compared to neat VO, the use of preheated VO reduces the smoke emissions (Pugazhvadivu and Jeyachandran 2005; Agarwal and Agarwal 2007; Agarwal and

Rajamanoharan 2009; Singh 2013; Hazar and Sevinc 2019). The preheating results in the reduction of fuel viscosity which subsequently improves spray, mixing of air-fuel and combustion characteristics (Hazar and Aydin 2010). Figure 2.20 (Agarwal and Rajamanoharan 2009; Singh 2013) illustrates the comparative smoke emissions of neat and preheated jatropha and karanja oil. It indicates a higher degree of reduction at the intermediate loads while lower variation in the higher engine loads.



(a) Jatropha oil (b) Karanja oil

Figure 2.20 Effects of preheated vegetable oil on smoke emissions of the engine (Agarwal and Rajamanoharan 2009; Singh 2013)

2.4 Effects of Vegetable Oil Blends in CI Engines

Blending of VO with diesel is another method of improvising the VO property as a fuel. Unlike preheating that requires an extra arrangement for heating, in blending VO is simply mixed with diesel in certain proportion. The VO has been used in the range of 5 to 75% by volume to blend with the diesel (Table 2.4). The blending results in the reduction in oil viscosity. The typical reduction in the viscosities of jatropha, karanja, soapnut and poon oil blend has been shown in Figure 2.21.

Table 2.4 VO blends used in engine.

Feedstock	Blending used	Researchers
Corn	10, 20	Rakopoulos *et al*. (2006)
	10, 20	Rakopoulos *et al*. (2011)
Cotton	10, 20	Rakopoulos *et al*. (2006)
	10, 20	Rakopoulos *et al*. (2011)
Deccan hemp	25, 50, 75, 100	Hebbal *et al*. (2006)
Jatropha	20, 30, 40, 50, 60, 70, 100	Pramanik (2003)
	10, 20, 50, 75, 100	Agarwal and Agarwal (2007)

Table 2.4 VO blends used in engine (Contd.)

Feedstock	Blending used	Researchers
Karanja	10, 20, 50, 75, 100	Agarwal and Rajamanoharan (2009)
	5, 10, 15, 20	Bajpai *et al*. 2009
Olive	10, 20	Rakopoulos *et al*. (2006)
	10, 20	Rakopoulos *et al*. (2011)
Poon	20, 40, 60, 100	Devan and Mahalakshmi (2009)
Rapeseed	20, 50	Qi *et al*. (2014)
Soapnut	10, 20, 30, 40	Misra and Murthy (2011)
Soya	10, 20	Rakopoulos *et al*. (2006)
Sunflower	10, 20	Rakopoulos *et al*. (2006)
	10, 20	Rakopoulos *et al*. (2011)

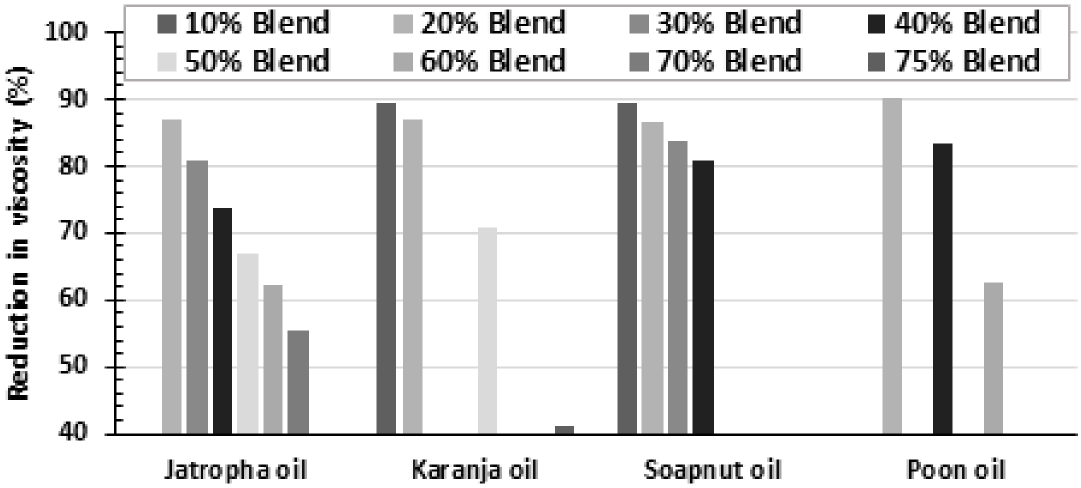
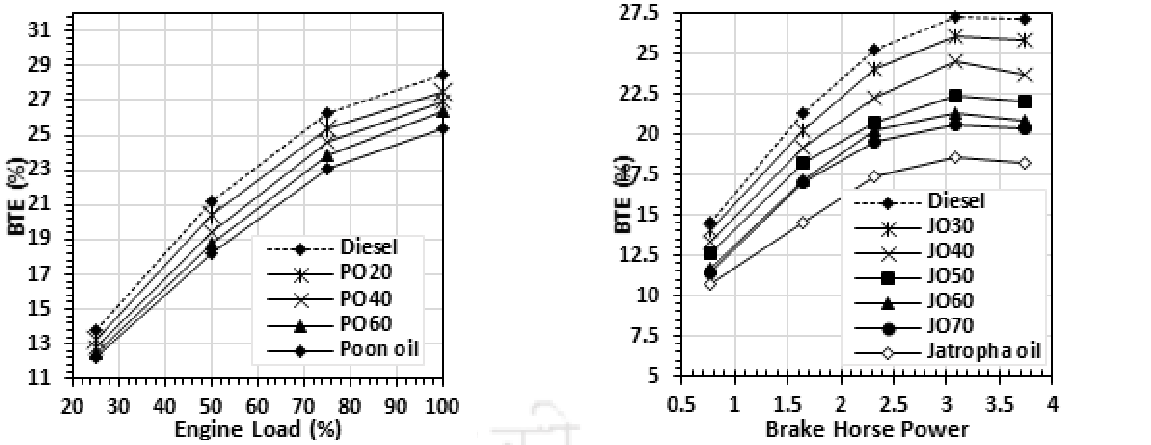


Figure 2.21 Effects of diesel blending on viscosity of vegetable oil (Pramanik 2003; Devan and Mahalakshmi 2009; Misra and Murthy 2011; Agarwal and Rajamanoharan 2009)

2.4.1 Brake Thermal Efficiency

Figure 2.22 shows the effect of VO blends on the engine BTE. It reflects a decrease in engine BTE with increasing percentage of VO in the blend. The effect is insignificant at the lower loads of the engine. The increase volume of VO in the blend increases the fuel viscosity and reduces the fuel volatility which results in the decrease of BTE (Pramanik 2003; Agarwal and Agarwal 2007; Bajpai *et al*. 2009; Devan and Mahalakshmi 2009; Misra and Murthy 2011). The vegetable blends upto 20% shows BTE comparable to that of the diesel. Rakopoulos *et al*. (2006); Bajpai *et al*. (2009); and Rakopoulos *et al*. (2011) reported even better BTE with VO than that of the diesel under 20% blend. However in the investigation of Agarwal and Rajamanoharan (2009) while using karanja oil, the blend upto 75% shows better BTE than that of the diesel.



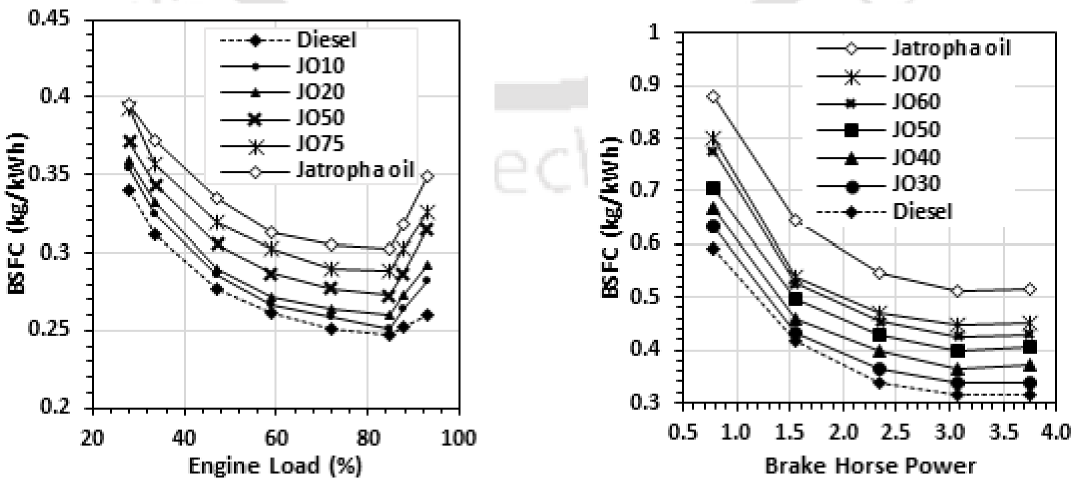
(a) Poon oil with 20, 40 and 60% blends (PO20, PO40 and PO60)

(b) Jatropha oil with 30, 40, 50, 60 and 70% blends (JO30, JO40, JO50, JO60 and JO70)

Figure 2.22 Effects of vegetable oil blending on BTE of the engine (Devan and Mahalakshmi 2009; Pramanik 2003)

2.4.2 Brake Specific Fuel Consumption

The BSFC of engine increases with the increase in blend percentage of VO. In Figure 2.23, the blending of jatropha oil 30, 40, 50, 60 and 70% by volume increases the BSFC, on an average, by 7, 15, 24 and 38% respectively. While the blending consists of 10, 20, 50 and 75% by volume of karanja oil increases it by an average of 4, 6, 12 and 17% respectively. This has been mainly attributed to the lower calorific value of VO as compared to that of the diesel. The blending reduces the calorific value of fuel which resulted in higher BSFC (Pramanik 2003; Rakopoulos *et al*. 2006; Agarwal and Agarwal 2007; Bajpai *et al*. 2009; Rakopoulos *et al*. 2011; Qi *et al*. 2014).



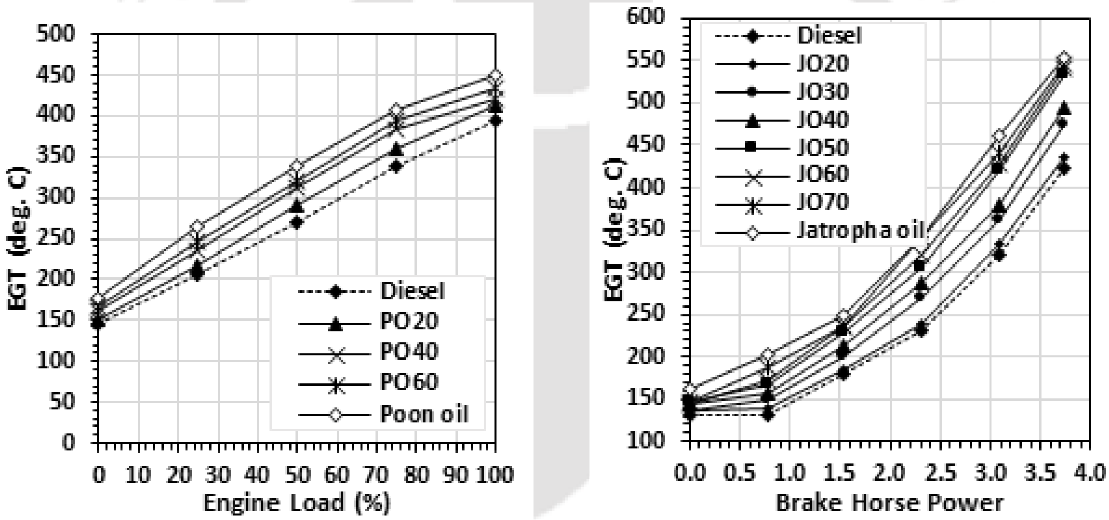
(a) Jatropha oil with 10, 20, 50 and 75% blends

(b) Jatropha oil and its blends

Figure 2.23 Effects of vegetable oil blending on BSFC of the engine (Agarwal and Agarwal 2007; Pramanik 2003).

2.4.3 Exhaust Gas Temperature

The increasing percentage of VO in the blend increases the EGT of the engine (Pramanik 2003; Hebbal *et al*. 2006; Agarwal and Agarwal 2007; Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009). The typical results of poon and jatropha oil blends used in the engine are shown in Figure 2.24. The results show a higher degree of gas temperature, throughout the entire range of engine operation, with higher amount of VO oil in the blend. As per the observation of Devan and Mahalakshmi (2009), the presence of higher boiling point constituents in the VO is responsible for the higher EGT with VO and its blend. These constituents of VO are not being adequately evaporated during main phase of the combustion and therefore it continues to burn in the late phase of the combustion resulting in slightly higher EGT.

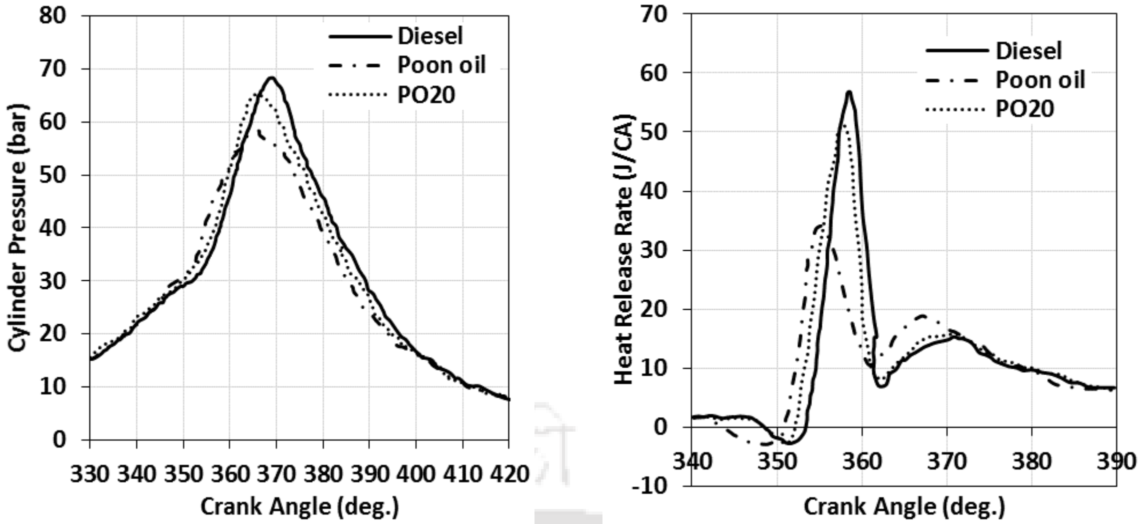


(a) Poon oil and its blends (b) Jatropha oil and its blends

Figure 2.24 Effects of vegetable oil blending on EGT of the engine (Devan and Mahalakshmi 2009; Pramanik 2003)

2.4.4 Combustion Analysis

In the investigations of Nwafor and Rice (1996), Devan and Mahalakshmi (2009), Misra and Murthy (2011), and Qi *et al*. (2014), it is observed that the blending of VOs with diesel improves the combustion which is higher with larger percentage of diesel in the blend as the diesel possesses higher calorific value and lower viscosity as compared to VOs. The variation of cylinder pressure and HRR with the use of poon oil and its blend PO20 (20% poon oil and 80% diesel by volume) in the engine is shown in Figure 2.25.



(a) Cylinder pressure variation with CA. (b) Heat release rate variation with CA.

Figure 2.25 Variation of cylinder pressure and heat release rate at full load (Devan and Mahalakshmi 2009)

With the use of PO20 blend in engine, the PCP of the engine increases to 63 bar as compared to neat poon oil which is 60 bar. The peak HRR increases from about 34.5 to 51 J/deg.CA respectively. Figure 2.26 also depicts the influence of diesel on HRR when it is blended with soap nut oil. The higher amount of diesel in soap nut oil-diesel blend improves the HRR.

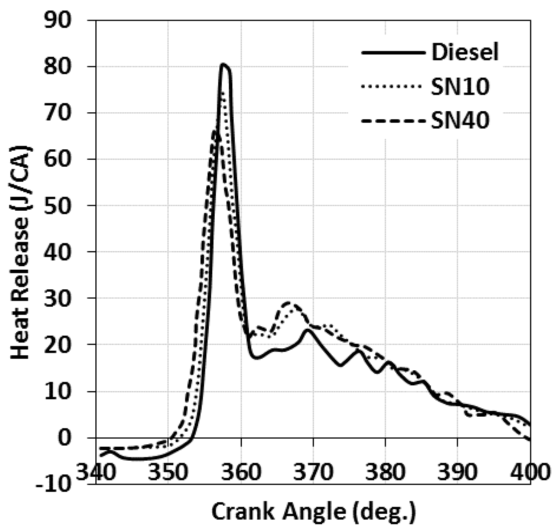
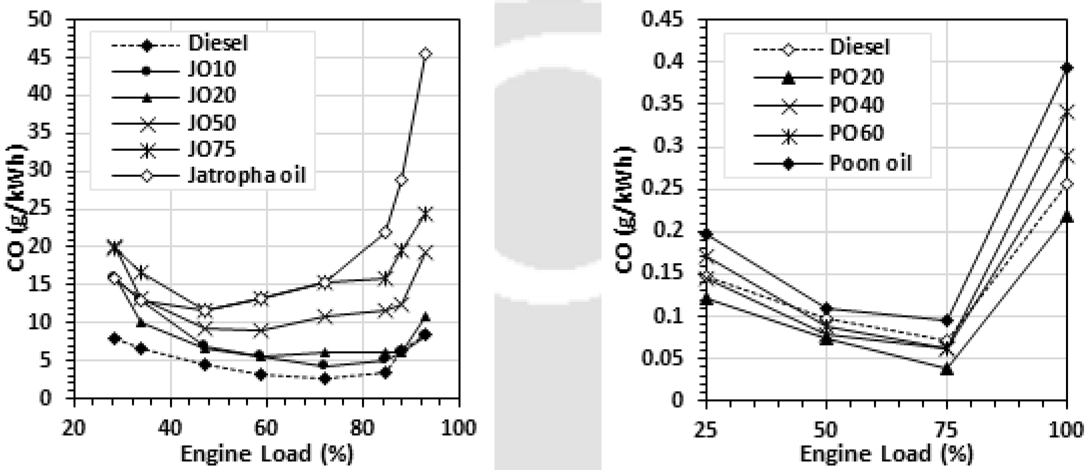


Figure 2.26. Variation of heat release rate at full load (Misra and Murthy 2011)

2.4.5 Carbon Monoxide Emissions

The blending of VO with diesel has found to improve the CO emissions in the engine although it remains higher than that of the neat diesel. At the lower blend, it is comparable to that of the diesel. The CO emissions was found to be lower than that diesel at the blend of 5, 10 and 15% (Bajpai *et al.* 2009). In Figure 2.27, the comparative results of diesel, VO and their blending with respect to the CO emissions have been presented. The CO emissions in the engine increases with the increase in the amount of VO in the blend. The increment is relatively high at the higher loads. The increase amount of VO in the blend increase the fuel viscosity results poor atomization and combustion of fuel and higher CO emissions. However, Qi *et al.* (2014) reported a lower CO emissions from the blend of 20 and 50% as compared to diesel at the high engine load.

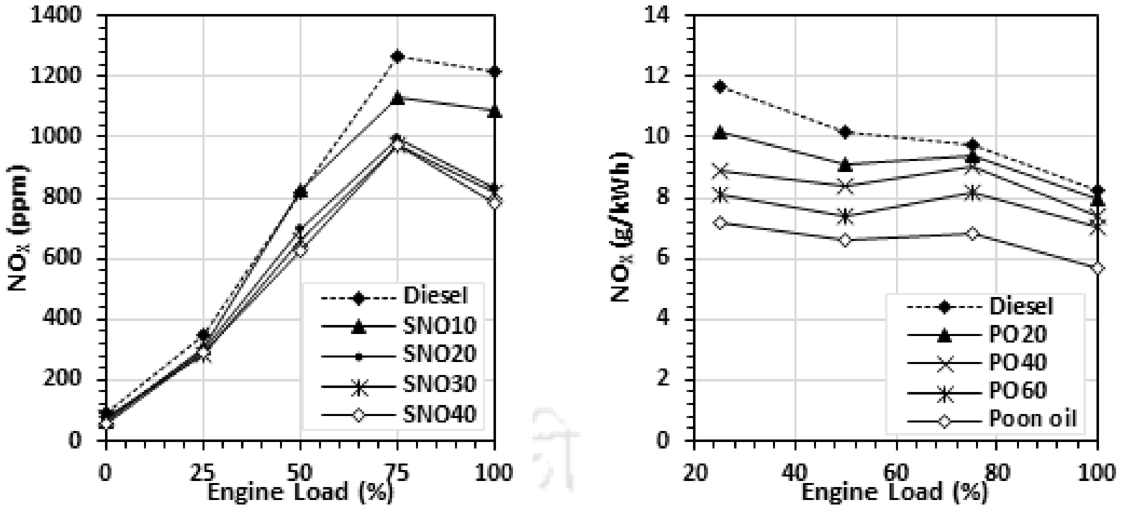


(a) Jatropha oil and its blends (b) Poon oil and its blends

Figure 2.27 Effects of vegetable oil blending on CO emissions of the engine (Agarwal and Agarwal 2007; Devan and Mahalakshmi 2009)

2.4.6 Oxides of Nitrogen

The impact of VO blend, derived from soapnut and poon, on NO_X emissions is shown in Figure 2.28. It reveals that higher percentage of VO in the blend reduces the NO_X. This trend has also been reported in the studies of Rakopoulos *et al.* (2006) and Agarwal and Rajamanoharan (2009). This reduction in emissions is probably because of the lower heating value of VO (Devan and Mahalakshmi 2009). However, Qi *et al.* (2014) found the NO_X emissions to be higher for VO blends as compared to the diesel at the higher engine loads. The increase in engine load results in more fuel injection and higher combustion temperature in the combustion chamber due to which the effect of viscosity might not become a dominating factor and the oxygen content in the reaction regions has an increased effect on the formation of NO_x (Qi *et al.* 2014). Similar higher NO_X emissions has been observed by Rakopoulos *et al.* (2011) at 10 and 20% blend with higher at higher blends.

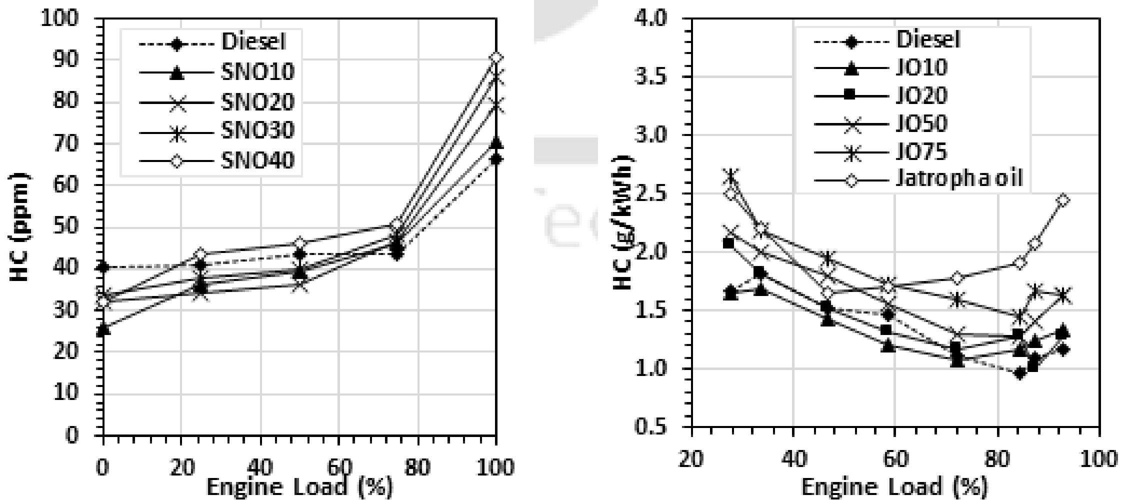


(a) Soapnut oil with 10, 20, 30 and 40% blends (b) Poon oil and its blends

Figure 2.28 Effects of vegetable oil blending on NO_X emissions of the engine (Misra and Murthy 2011; Devan and Mahalakshmi 2009)

2.4.7 Hydrocarbon Emissions

As the engine powered on VO produces higher HC emissions compared to that of the diesel fuel due to higher viscosity of former than the later, it is obvious that the blending with diesel will reduce it. It is evident from Figure 2.29 that the HC emissions reduces by blending VO with diesel. It decreases with the decreasing proportion of VO in the blends. Bajpai *et al*. (2009) reported of lower HC emissions, at 10% blend of karanja oil, compared to diesel upto 70% of engine load. However, it increases beyond 75% load.



(a) Soapnut oil and its blends (b) Jatropha oil and its blends

Figure 2.29 Effects of vegetable oil blending on HC emissions of the engine (Misra and Murthy 2011; Agarwal and Agarwal 2007)

2.5 Use of Ethanol as Additives on VO-diesel Blends

The increase in the quantity of VOs in the blends deteriorates the engine performance and increases the emissions of CO, HC and smoke due to the increase of fuel viscosity. In this context, an additive or improver can be used in the VO-diesel blend with a high VO content. Ethanol has been the most commonly used additive in the binary blend of VO and diesel. The consideration of using ethanol has an additional merit of being a fuel derived from the biomass. Qi *et al*. (2016) used 10, 20 and 30% ethanol by volume in the VO-diesel blend (BE0) consisting of 50% rapeseed oil and 50% diesel by volume. Ternary blends BE10, BE20 and BE30 consisting of VO, diesel and ethanol were prepared through micro-emulsification process using surfactant. In a study by Sathiyamoorthi and Sankaranarayanan (2017), ethanol was used in the binary blend of neat lemongrass oil and diesel (LGO25). LGO25 consists of 25% neat lemongrass oil and 75% diesel by volume. Ethanol was added to LGO25 by 2.5 and 5% to form ternary blend LGO25E2.5 and LGO25E5 respectively. In another study by Qi *et al*. (2017), tung oil-diesel-ethanol ternary blend (prepared through micro-emulsification) with different volume ratio was used in the common rail direct injection CI engine. Keeping the fraction of diesel as 30% by volume Prakash *et al*. (2018) varied the faction of VO and ethanol as 10% ethanol and 60% VO (NCO60D30E10), 20% ethanol and 50% VO (NCO50D30E20), 30% ethanol and 40% VO (NCO40D30E30) in the ternary blend composed of diesel, ethanol and castor oil. The BSFC of the engine was found to increase with the use of ethanol in the blend (Qi *et al*. 2016; Sathiyamoorthi and Sankaranarayanan 2017; Saleh and Selim 2017; Qi *et al*. 2017).

This has been attributed to the lower calorific value of the ethanol resulting in the need of greater amount of fuel to be injected into the engine cylinder for the desired output. The typical results on the effect of ethanol on the VO-diesel blend are shown Figure 2.30. Throughout the operating range of engine, the BSFC increases with the 10 and 20% blending of ethanol as compared to diesel and VO-diesel blend. However, it indicates a drop with 30% ethanol blend.

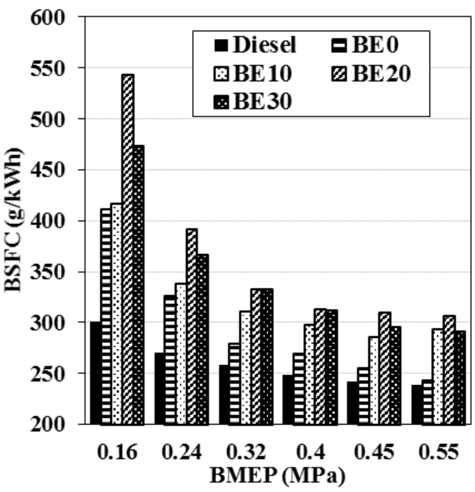


Figure 2.30 Effect ethanol on BSFC of the engine (Qi *et al*. 2016)

Figures 2.31 and 2.32 show the influence of ethanol on the BTE of the engine. The BTE of the engine improves with the blending of ethanol with VO-diesel blend. It is believed to be the improvement in the combustion due to additional supplement of oxygen, better atomization and mixture formation with the use of ethanol (Qi *et al*. 2016; Sathiyamoorthi and Sankaranarayanan 2017; Qi *et al*. 2017). The ethanol being oxygenated fuel and lower viscosity as compared to diesel and VO.

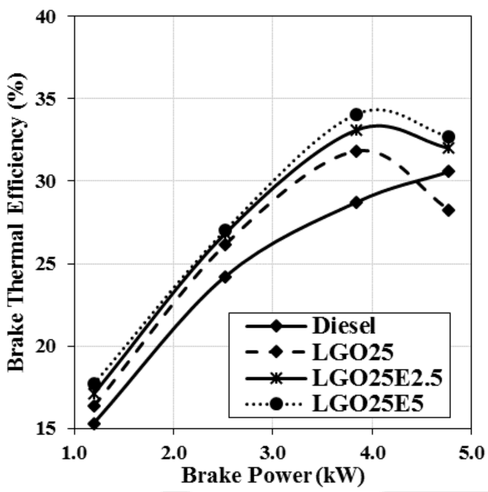


Figure 2.31 Effect ethanol on BTE of the engine (Sathiyamoorthi and Sankaranarayanan 2017)

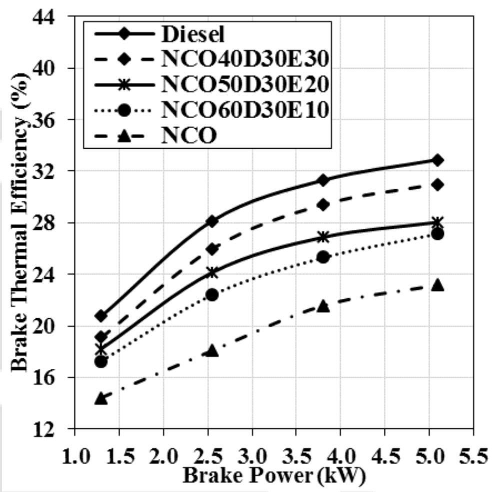


Figure 2.32 Effect ethanol on BTE of the engine (Prakash *et al*. 2018)

In most of the studies, the EGT was found to increase with the blending of ethanol with VO-diesel blend. Sathiyamoorthi and Sankaranarayanan (2017) found an increase in EGT to 17 and 18%, respectively with the blending of 2.5 and 5% ethanol respectively as depicted in Figure 2.33. This was attributed to the higher ignition delay which induces after burning of hydrocarbons in the exhaust resulting in higher exhaust gas temperature.

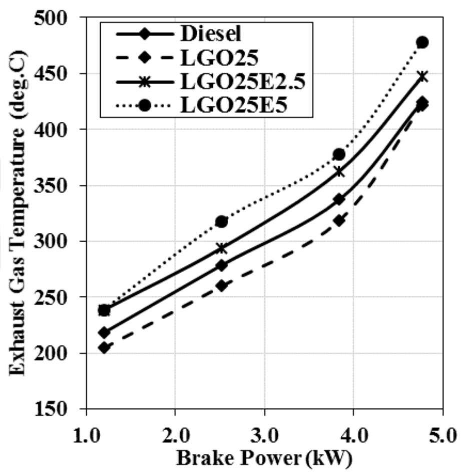


Figure 2.33 Effect ethanol on the EGT of the engine (Sathiyamoorthi and Sankaranarayanan 2017)

The use of ethanol at lower concentration is found to reduce CO emissions. The blend of ethanol up to 5% is found to reduce the CO emissions which is lower than that of the diesel (Figure 2.34). This has been credited to the enrichment of oxygen with the addition of ethanol (Sathiyamoorthi and Sankaranarayanan 2017). However, CO emissions increases with the higher content of ethanol in the blend and this is attributed to the cooling effect due to higher latent of vaporization of ethanol. The emissions of unburned HC in the engine exhaust is found to increase with the increase in the content of ethanol in VO-diesel blend. Figure 2.35 shows an increase in the emissions of HC with an increase in the content of ethanol from 10 to 30% in rapeseed oil-diesel blend. This has been attributed to the cooling effect of ethanol which is believed to reduce the in-cylinder gas temperature resulting in poorer oxidation reaction rate and thus increases HC emissions (Qi *et al*. 2016).

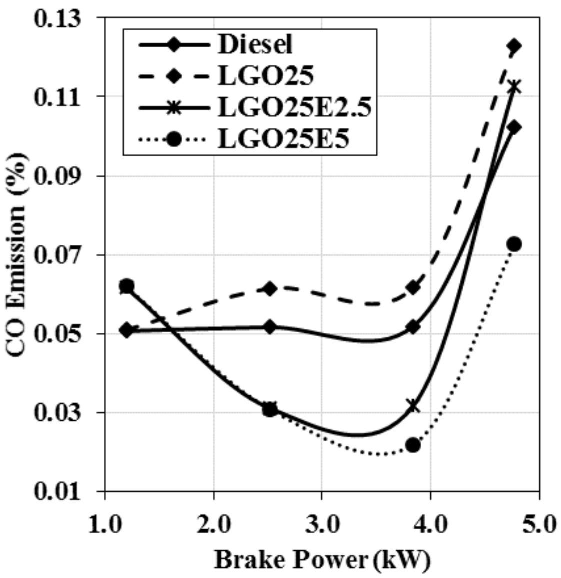


Figure 2.34 Effect ethanol on the CO emissions of the engine (Sathiyamoorthi and Sankaranarayanan 2017)

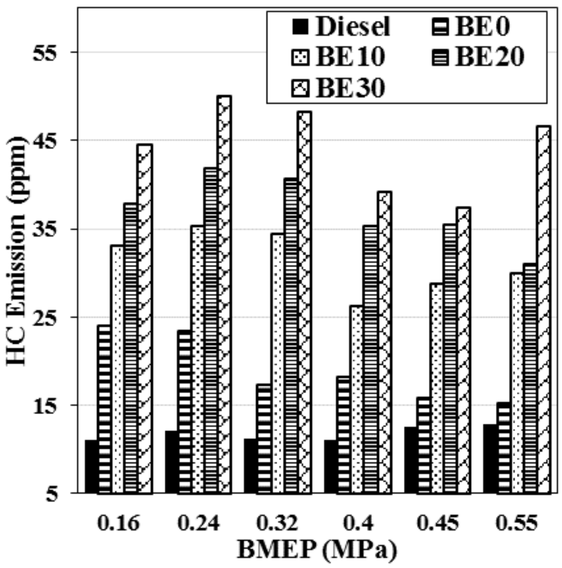


Figure 2.35 Effect ethanol on the HC emissions of the engine (Qi *et al*. 2016)

The addition of ethanol results in the increase in the NOx emissions in the engine exhaust as revealed in Figure 2.36. The improvement in the premixed combustion phase results in the increase in in-cylinder temperature which favors the NOx formation (Qi *et al*. 2017; Prakash *et al*. 2018). Sathiyamoorthi and Sankaranarayanan (2017) have cited the influence of low cetane number of ethanol on NOx emissions of the engine.

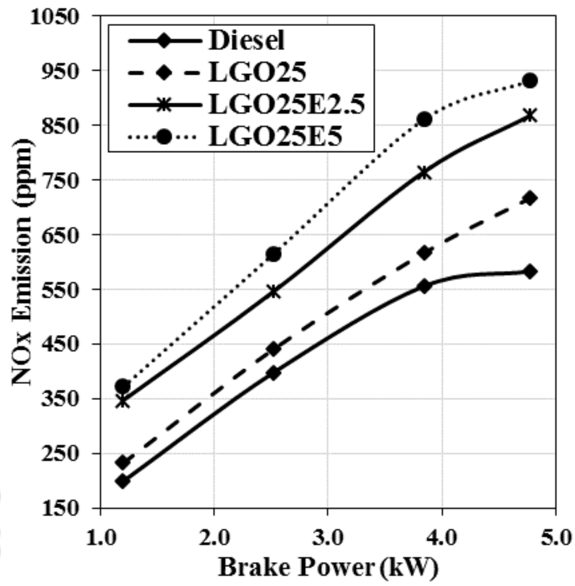


Figure 2.36 Effect ethanol on the NOx emissions of the engine (Sathiyamoorthi and Sankaranarayanan 2017)

2.6 Use of Diethyl Ether as Additives on VO-diesel Blends

The use of diethyl ether (DEE) as an additive in VO-diesel blend has not been reported in open literatures. However, the blending of DEE with diesel at different proportion has been carried out by the researchers. Paul *et al.* (2015) blended DEE with diesel by 5 and 10% to form two DEE-diesel blend consists of DEE 5 and 10% by volume respectively. While Ibrahim (2016) used 5, 10 and 15% DEE in DEE-diesel blend. Patnaik *et al.* (2017) blended 15% DEE to the diesel to form DEE-diesel blend consists of 15% DEE and 85% diesel by the volume. The blending of DEE with diesel is found to improve the BTE of engine. The CO and HC emissions are found to decrease with the blending of DEE. However, NOx emissions increases with the use of ethanol. It likely that the use of DEE as an additive in VO-diesel blend will have the similar positive effect on the CI engine run on VO-diesel blend.

2.7 Summary and Scope of Work

In this literature review, the main focus is to understand the effect of using neat and blended VOs on the performance and emissions characteristics of CI engines. The salient features of this study are summarized as below:

- The VOs are characterized by high viscosity and lower heating value compared to that the diesel. This results in the poor performance of engine on neat VO mode. The BTE of engine drops when neat VO was used in the engine. This drop extends up to 14%

which is due to the combined effects of lower calorific value, high viscosity and poor volatility of vegetable oil resulting in the poor combustion characteristics. The BSFC of engine goes up to 31% higher than diesel when vegetable oil was used due to the lower heating of vegetable oil, as compared to that of the diesel. This creates a need of larger mass fuel flow to maintain constant energy input to the engine. In most of the studies, vegetable oil as a fuel in engine gives higher EGT in comparison to diesel.

- Neat VO increases the CO emissions throughout the operation range as compared to that of diesel due higher viscosity of oil which creates difficulty in fuel atomization leading to locally rich mixtures and incomplete combustion. The NO_x emissions in the engine reduces with the use of VO. In some studies, higher amount of NO_x emissions compared to that of the diesel has also been reported. A significant amount of HC emissions is observed when vegetable oil is used in the engine.
- In comparison to neat VO mode, the performance of engine found to have improve with use of preheated neat VO, blending of VO with diesel and use of additives in VO-diesel blends.
- With the use of preheated VO, the BTE of engine increase up to 24% as compared neat VO. There is a decrease in the BSFC, CO and HC emissions due the preheating. However, the preheating of VO results in the increase in the NOx emissions.
- Blending of VO with diesel up to of 20% shows efficiency comparable to that of the diesel. The efficiency was even better than diesel when blending under 20% were used in the engine. However, it decreases with the increase in the amount of VO in the blend. The increase in the percentage of VO in the blend increases the BSFC. The increasing percentage of VO in the blend increases the EGT of the engine.
- The CO emissions in the engine increases with the increase in the amount of VO in the blend. The increment is relatively high at the higher load. However, at the lower blend it is comparable or sometimes even better than that of the diesel. The higher percentage of VO in the blend reduces the NO_x. However, opposite trend has also been reported with blending. The increase in the percentage of VO in the blend increases the HC emissions in the engine.
- For the high content VO in VO-diesel blend, the use of additives such as ethanol found to improve the engine performance. However, it has a mixed effect on the engine emissions.

This review reveals that neat VO can been used in the engine although it may need some extra maintenance of engine aroused out of higher viscous nature of the oil. This may include regular inspection and cleaning of fuel injection system, running the engine with diesel mode at the starting and stopping of engine. The blending of VO is likely to give better engine performance. Blending will be simplest and best way to use VO in the engine as it does not involve any modification or addition of supplementary system to engine. In blending, certain volume of VO is simply mixed with diesel and it does not require any skill. This simple technology can be effectively utilized especially in the rural areas for driving the stationary engines used for irrigation and power supply. The need is to exploration and systematic exploitation of these natural fuel source. An estimate based on the National Policy on Biofuels 2009, the Ministry of New and Renewable Energy, India is endowed with more than 400 species of plants that bear non-edible oil seed. Through proper research, planning and management, these resources can effectively be utilized to supplement the energy need. The blending of VO up to 20% with diesel makes a comparable performance with that of diesel. The supplement of even 10% can make a huge impact in economics of fuel supply and demand, which are in million dollars.

In most of the studies related to the use of vegetable oil in CI engines, the researchers have focused on deriving the oil from various feedstocks that are available in that particular area. The feedstocks are then sequentially subjected to various processes to obtain the final product as promising fuel. The north eastern region of India is endowed with large varieties of flora. Many of them have oil bearing seeds which can be explored as a fuel to power CI engines. In this perspective, the oil derived from seeds of *Mesua ferrea* Linn tree can be a promising source of fuel to supplement diesel in CI engines because of its high oil bearing seed and availability. The trees are planted in the residents, offices, institutes, parks, as ornamental plant because of their aesthetic look and for the shade. Hence, this thesis work is envisaged to explore the applicability of *Mesua ferrea* Linn oil in CI engines.

Chapter 3

Production and Characterization of Fuel

OVERVIEW

The present study aims at exploring an alternative fuel source either to supplement or substitute the fossil diesel. In this context, the applicability of oil derived from the seeds of Mesua ferrea Linn is focused. The consideration of Mesua ferrea Linn oil (MO) in this work is based on the fact that these oil rich seeds are non-edible and grow well in the North-Eastern part of India. This chapter is focused on the production and characterization of fuel. It describes the equipment used for determining the properties of oil such as heating value, viscosity, density and acid value.

Chapter Outline:

3.1 Introduction

In stressing the need of exploring and utilizing the renewable and locally available energy source to supplement diesel in stationary diesel engine, the oil derived from the seeds of *Mesua ferrea* Linn can be used as fuel to run the CI engines. The supplementing or replacing of a non-renewable energy sources with a renewable sources will help in managing the gap in the supply and demand of energy by serving as an alternative arrangement for compensating the energy gap. The exploring and taping the locally available energy source will open a door of employment opportunity and reduce the dependency of fossil diesel.

Figure 3.1. Tree, fruit and seeds of *Mesua ferrea* Linn.

Mesua ferrea Linn is commonly known as iron wood, Indian rose chestnut or *nagkeshar*. The plant is native of India, Myanmar, Sri Lanka, Singapore, Cambodia, Malaysia, Philippines, Thailand and Vietnam (Orwa *et al*. 2009). In India, it is commonly grown in the mid-hills of Eastern Himalaya, north eastern India, Deccan peninsula, Andaman Island and in rain forests of Konkan and Karnataka (NMPB publications). It is also known for its medicinal values such as antioxidant, analgesic, antitumor and anti-microbial. It is an ingredient to many ayurvedic and unani formulations (Chahar *et al*. 2013). The seeds produced by *Mesua ferrea*

Linn trees bear high oil which lie in the range of 58 to 75% (Kushwah *et al*. 2008). It is estimated that more than 10 million kg of seeds are being produced annually in state of Assam, India (Kushwah *et al*. 2008). The oil produced from the seed has high potential to supplement the fuel requirement of the stationary CI engines that are generally used for irrigation and power production sectors of rural areas. A typical tree of *Mesua ferrea* Linn along with its fruits and seeds is shown in Figure 3.1.

Figure 3.2 Decorticated seeds

Figure 3.3 Hand operated oil expeller.

3.2 Oil Production

For the production of oil, the *Mesua ferrea* Linn seeds have been collected from the states of Arunachal Pradesh and Assam in India. The oil production consisting of preprocessing, oil extraction and post processing phases. In the preprocessing phase, the collected seeds are decorticate to separate the kernel from the shell cover. The kernels (Figure 3.2) are then dried in the sun light or 3 to 4 days till their color change from yellow to brown. This sundried seed kernels are then fed to the expeller for oil extraction. The post processing phase consists of filtering and demoisturing the extracted oil. In this study, two oil extractors of hot screw press type are used. Initially, a hand operated oil expeller as shown in Figure 3.3 is used for the extraction of oil. This hand operated oil expeller is manufactured by Rajkumar Agro Engineers Pvt. Ltd, Nagpur, India. The construction consists of screw press with a handle for manual cranking. The heat is supplied to the press through a simple glass bottle oil lamp. During the process of oil extraction through this expeller, random samples have been collected to estimate oil yield. The oil yield of oil expeller is shown in Table 3.1. It is in the range 33.19 to 53.33% with an average yield of 42.51%. Both the manual and power operated extraction of oil are carried out in the IC Engine Laboratory of Mechanical Engineering Department, IIT Guwahati.

Table 3.1 Oil yields from hand operated oil expeller.

Sl. No.	Oil Seed used (g)	Oil Yields		Yield (%)
		(g)	(ml)	
1	100	35	50	35.00
2	180	94.54	-	52.52
3	180	96	-	53.33
4	1000	401	458	40.10
5	1000	371	430	37.10
6	1000	346	390	34.60
7	1000	542	-	54.20
8	690	229	-	33.19
Average oil yield				**42.51**

Figure 3.4 Power operated oil expeller.

At a later stage, a power operated oil expeller as shown in Figure 3.4 is used to speed up the process of oil extraction. The specification details of the power operated oil expeller is shown in Table 3.2. The expeller takes around 70 minutes to convert one kg of seed to oil. As revealed in Table 3.3, the oil yield through this expeller is in range of 44 to 65% with an overall average yield of 51% which is almost 10% higher than that of the hand operated expeller. The mechanical extraction method yields a lower oil content as compared to the solvent extraction method. Kushwah *et al*. (2008) reported an oil yield in the range 58 to 75% by the solvent extraction technique. The extracted oil is filtered for the removal of unwanted residues using filter papers. Finally, the filtered oil is demoistured by heating above 100 °C for two hours.

Table 3.2 Specifications of oil expeller

Make	Vishvas Oil Maker, Surat India
Model, Type	VI-391, hot screw press
Capacity	3-6 kg/hr (According to the material)
Voltage	220V-240V
Motor power	400 W
Material	304# Food grade stainless steel

Table 3.3 Oil yields from power operated oil expeller

Sl. No.	Oil Seed used (g)	Oil Yields		Yield (%)
		(g)	(ml)	
1	1000	477	520	47.70
2	1000	434	480	43.40
3	1000	443	500	44.30
4	1000	408	450	40.80
5	1000	478	550	47.80
6	1350	631	650	46.74
7	2000	1305	1440	65.25
8	2000	1282	1430	64.10
9	2500	1391	1460	55.64
Average oil yield				**50.64**

3.3 Characteristics of *Mesua ferrea* Linn Oil

The characteristics of produced *Mesua ferrea* Linn oil such as composition, heating value, viscosity and density have been determined using the standard procedures. The details have been discussed in the following sub-section.

3.3.1 Composition

The gas chromatography of oil carried out by Kushwah *et al.* (2008) reveals of oil being composed of both saturated and unsaturated fatty acids. The details of composition have been shown in Table 3.4. The major constituents consist of oleic acid which constitutes 55.93% of the total composition. This is followed by 14.19 and 13.68% stearic and linoleic acid respectively. Based on the proportional composition of the respective fatty acids, the chemical formula of the oil is found to be $C_{17.71}H_{33.75}O_2$.

Table 3.4 Fatty acid compositions of *Mesua ferrea* Linn oil (Kushwah *et al*. 2008).

Fatty Acid	Structure*	Formula	Composition (%)
Myristic	14:0	$C_{14}H_{28}O_2$	2.13
Palmitic	16:0	$C_{16}H_{32}O_2$	10.87
Stearic	18:0	$C_{18}H_{36}O_2$	14.19
Oleic	18:1	$C_{18}H_{34}O_2$	55.93
Linoleic	18:2	$C_{18}H_{32}O_2$	13.68
Arachidic	20:0	$C_{20}H_{40}O_2$	2.921

**xx*:*y* indicates *xx* carbons in the fatty acid chain with *y* double bonds

3.3.2 Heating Value

The heating value of fuel gives the amout of heat energy it produces when a unit mass of it is completly combusted. It is expressed as higher gross or heating value and lower or net heating value. The higher heating value of oil was determined using 1341 plain jacket bomb calorimeter follwing standard procedure given in the instrument's manual. Calorimeter is of Parr Instrument Company make having specifications of static jacket calorimetry, 2 test per hour, 0.3% precision class, 6775 digital thermometer, manual oxygen fill, manual bucket fill, manual bomb wash. The lower heating value is calculated using the standard relation. The higher and lower heating values of oil are found to 38,280 and 35,611 kJ/kg, respectively.

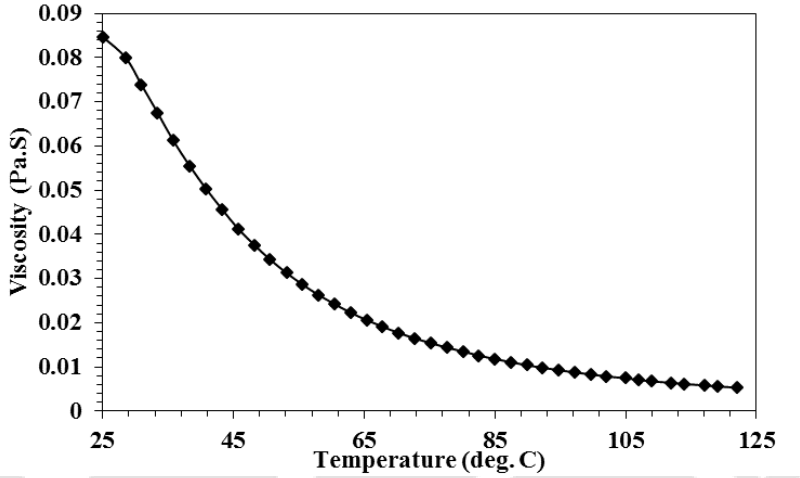


Figure 3.5 Variation of oil viscosity with different temperatures.

3.3.3 Viscosity

In CI engines, the viscosity of fuel plays a very important role in combustion. The ability of fuel to flow through pipes, atomization, vaporization and mixing with air depends on the viscosity of fuel. The lower viscosity of fuel will have better atomization, finer fuel droplets, faster vaporization and uniform mixing with air leading to improved combustion in the engine combustion chamber. For the measurement of the oil viscosity, a Physica MCR 101 rheometer (Anton Paar make) is used. The rheometer has the facility of giving the dynamic viscosity at different temperatures of the oil. In the present study, the viscosity of *Mesua ferrea* Linn oil is determined for the temperature range of 25 to 122 ºC. The viscosity profile of oil at the different oil temperature is depicted in Figure 3.5. It reveals a drop in the viscosity oil with the increasing temperature. The drop rate is very high in the lower temperature ranges of 25 to 50 ºC, and at higher temperature there is a reduction in the rate of decrease in the oil viscosity. The kinematic viscosity of oil is determined using this result and it is found to be 56.107 mm^2/s at 40 ºC.

3.3.4 Other Properties

The density of the oil was determined using specific gravity bottle of 25 ml capacity. The mass and the volume of oil was measured through this bottle and the evaluated the density. The density of oil was found to be 927.8 kg/m^3 at 15 ºC. The acid value of oil was determined titration. The stoichiometric air-fuel ratio was evaluated using the balance chemical reaction of the oil with the oxygen. The summary of the analyzed properties *Mesua ferrea* Linn oil been shown in Table 3.5.

Table 3.5 Properties of *Mesua ferrea* Linn oil.

Chemical formula	$C_{17.71}H_{33.75}O_2$
Higher heating value, kJ/kg	38,280
Lower heating value, kJ/kg	35, 611
Density at 15 ºC, kg/m^3	927.8
Kinematic viscosity at 40 ºC, mm^2/s	56.107
Acid value, mg KOH/g	33.26
Stoichiometric air-fuel ratio	12.5

3.4 Concluding Remark

This study is focused on utilizing the *Mesua ferrea* Linn oil to supplement the diesel in CI engines. Therefore, the production and characterization of oil plays a very important role in enumerating the fuel qualities and to evaluate and analyze the effect of the oil on the engine performance, combustion and exhaust emissions.

Chapter 4

Experimental Setup and Procedures

OVERVIEW

This chapter is dedicated to enumerate the experimental setup and the instruments used in the present investigation. The chapter begins with the description of the diesel engine test setup including the sensors and transmitters used. The chapter then outlines the procedures implemented and conditions maintained during the conduct of experiments. The description of instruments used for the measurement of air flow rate, fuel flow rate, pressure-crank angle and temperature is also outlined. The chapter gives the description and specifications of the exhaust gas analyzer used for the measurement of CO, NOx HC and CO_2.

Chapter Outline:

4.1 Diesel Engine Test Setup

The experiments have been performed in a single cylinder, four-stroke, water-cooled, naturally aspirated, constant speed, direct injection CI engine. The detailed specification of the engine is presented in Table 4.1. Other than the engine, the experimental set up is equipped with a dynamometer and a control panel. The dynamometer provides an electromagnetic loading on the engine crankshaft and the loading is measured through strain gauge load cell. The control panel includes the air box, manometer, separate fuel tank for diesel and vegetable oil blends, fuel measuring unit, dynamometer loading unit (DLU), data acquisition device (DAD) and digital display units for temperature, engine speed and load. The schematic diagram of the experiment set up is shown in Figure 4.1 and the photographic view is shown Figure 4.2. The setup is equipped with crank angel sensor and two piezo sensors to measure the fuel injection and the cylinder pressure. Further, there are three resistance temperature detector type sensors for water temperature measurements and two K-type thermocouples for gas temperature measurement. The signals from the sensors are routed to DAD and then transmitted to computer through USB port. In the computer, the online data are generated and displayed through the Labview based engine performance analysis software package Enginesoft. The specifications of transmitters and sensors are shown in Table 4.2., whereas those of the flow meters are presented in Table 4.3.

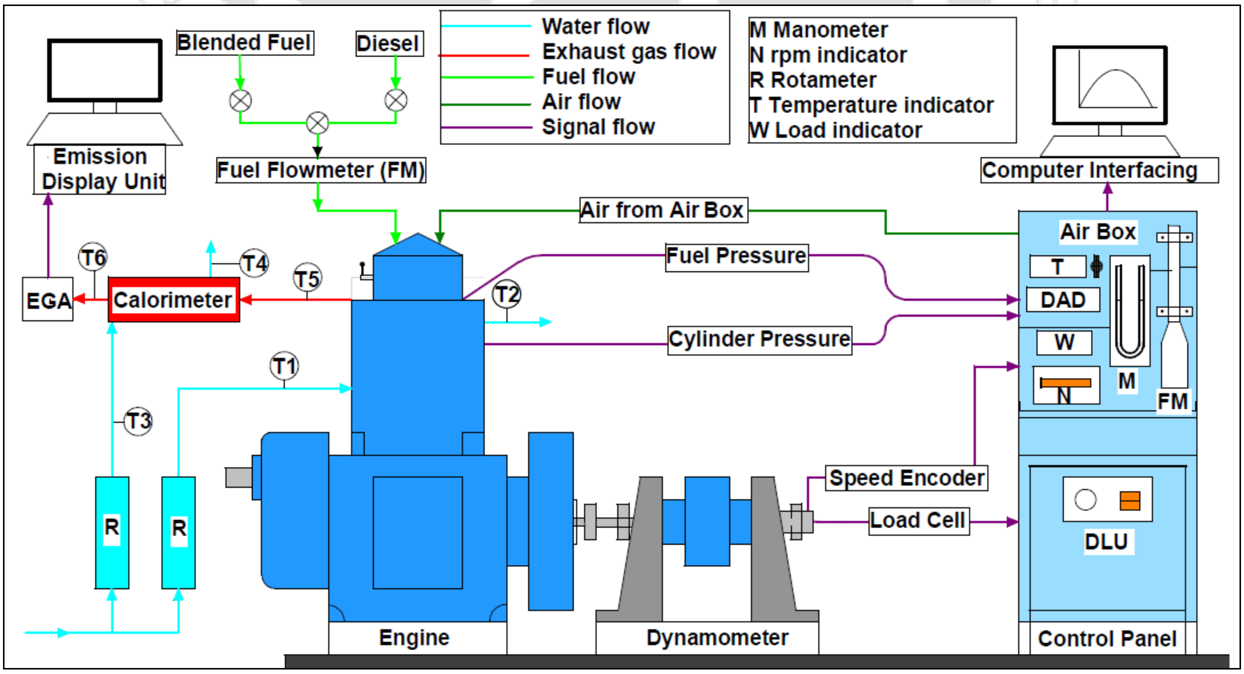


Fig 4.1 Schematic diagram of experimental setup.

Fig 4.2 Photographic view of the experimental setup.

Table 4.1. Specifications of the experimental setup.

Setup components	**Parameters**	**Specifications**
Engine	Product	Research engine test setup, Code 240
	Make and model	Kirloskar, TV1
	Type	Single cylinder, four-stroke, CI engine
	Capacity	661 cm^3
	Bore	87.5 mm
	Stroke	110 mm
	Cooling mode	Water cooled
	Air induction mode	Naturally aspirated
	Type of injection	Direct injection
	Connecting rod	234 mm in length
	Power	3.5 kW at 1500 ±50 rpm
	Compression ratio	12:1 to 18:1
	Injection variation	0 - 25^0 bTDC
	Combustion chamber	Hemispherical bowl in piston type
	Fuel tank capacity	15 litres for diesel
		5 litres for blended fuels
	Air box	MS fabricated with orifice meter and manometer
	Calorimeter	Pipe in pipe type
Data acquisition	Device	NI USB-6210, 16-bit, 250kS/s.
	Software	“Enginesoft” Engine performance analysis
Dynamometer	Type	Eddy current with loading unit, water cooled
	Arm length	185 mm
Load indicator	Type	Digital, Range 0-50 kg
Load sensor	Type	Strain gauge, Range 0-50 kg
	Input	Load cell
Speed indicator	Type	Digital with non-contact type speed sensor
	Range	5500 rpm

Table 4.2 Specifications of flow meter.

Setup components	Parameters	Specifications
Air flow meter	Type	Orifice and manometer (U tube type)
	Range	100-0-100 mm
	Oifice diameter	20 mm
	Coefficient of discharge for orifice (C_d)	0.6
Fuel flow meter	Type	Burette
	Range	0 – 450 ml
Water flow meter	Type	Rotameter
	Range	40-400 lph for engine cooling
		25-250 lph for calorimeter

Table 4.3 Specifications of the transmitters and sensors in the experimental setup.

Transmitters and sensors	Specifications
Air flow transmitter	Pressure transmitter, Range (-) 250 mm WC
Fuel flow transmitter	DP transmitter, Range 0-500 mm WC
Crank angle sensor	Resolution 1 Degree, Speed 5500 RPM with TDC pulse.
Pressure sensor (combustion)	Piezo type, Range 5000 PSI, with low noise cable
Pressure sensor (diesel line)	Piezo type, Range 5000 PSI, with low noise cable
Temperature sensors and transmitters	PT100 (RTD) type, range 0-100º C, output 4-20 mA (4 nos) K (ungrounded) type, range 0-1200º C, output 4-20 mA (2 nos)
Speed sensor and indicator	Resolution 1 deg., range 5500 rpm with TDC pulse

4.2 Experimental Procedure

The experiments have been carried out at the engine loads of 20, 40, 60, 80 and 100%. The experiment procedure starts with the warming up of engine at no load condition until the temperatures of exhaust gas and the outlet of cooling water attain a steady state condition. The load is gradually increased to 20% and then to the higher loads at an interval of 20%. On reaching a steady state condition at each load, the fuel flow rate, air flowrate, speed, cylinder pressure and fuel pressure variation have been recorded. The emissions of CO, NO, HC and CO_2 from the engine exhaust are also recorded. The experiments begin with the diesel mode which is then followed by the vegetable oil blends. For the blended fuels, initially the engine is warmed up under diesel mode and upon warming up the diesel supply is cut off through the flow control valve. The supply of blended fuel is then switched on and the engine is run till the steady state condition is reached on blended fuel mode. Upon reaching the steady state, the load is increased to 20%, 40%, 60%, 80% and 100%. On completion of experiments, the engine is run again on pure diesel mode to flush out the blended fuel in the engine. During the course of the experiments, the water is supplied at rate of 300 l/h for engine cooling and 100 l/h for calorimeter. The load variations on the engine are conducted at 1500 ± 50 rpm. Experiments have been conducted at CR of 17.5 and IT of 23^0 bTDC of the test engine, and this is not optimized condition for the blended fuels. These performance variables are substituted in the relevant expressions (described in Appendix A) to determine the engine performance parameters such as brake power (BP), brake thermal efficiency (BTE), brake specific fuel consumption (BSFC) and others. The uncertainty in the measurement of independent variables and performance parameters are provided in Appendix B.

4.3 Instruments on Engine Setup

The engine setup is equipped with various instruments for measuring various parameters for evaluation and analysis of performance, combustion and exhaust emissions. The following subsections describe the various instruments used.

4.3.1 Air and Fuel Flow Measurement

The air flow rate is measured through the manometer fixed on the control by recording the difference in the level of water column. It is interconnected across the orifice meter, through which air comes into the engine control panel box before leaving towards the engine manifold. The fuel flow rate is measured through the graduated glass burette mounted on control panel box by recording the time required for the consumption of certain volume of fuel using a stop watch.

4.3.2 Pressure-crank Angle Measurement

For the measurement of in-cylinder pressure and the fuel injection pressure, the PCB Piezotronics make dynamic pressure sensors are mounted on the cylinder head and the fuel injector. Both of them have the identical specification and capable of distinguishing the pressure of compression, combustion, explosion, pulsation, cavitation, blast, pneumatic, hydraulic, fluidic etc. An optical crank angle sensor (Kubler make) is used to measure each degree rotation of crank with TDC pulse.

4.3.3 Temperature Measurement

Four PT100 type (RTD) temperature sensors measure the inlet and the outlet temperatures of engine cooling water flow and calorimeter water flow. The inlet and outlet temperatures of exhaust gas to calorimeter are measured by two K type thermocouples. The temperatures are recorded from the digital display unit located in the control panel.

4.3.4 Emissions Measurement

The emissions from the engine are measured using AVL make exhaust gas analyzer. The analyzer uses the non-dispersive infrared (NDIR) system to measure CO, CO_2, and HC, whereas the electrochemical sensor measures NO and O_2. The detailed specifications of the exhaust gas analyzer are shown in Table 4.4.

Table 4.4 Specifications of exhaust gas analyser.

Make and model : AVL, DIGAS 444 N				
Measured Gas	Range	Resolution	Accuracy	
CO	0-15 vol.%	0.01 vol.%	0-10% ± 0.02% abs 10.01% - 15%	±3% rel ±15% rel
CO_2	0-20 vol.%	0.01 vol.%	0-16% ± 0.3% abs 16.01% - 20%	±3% rel ±5% rel
HC	0-30000 ppm	≤ 2000 : 1 ppm > 2000 : 10 ppm	0-4000ppm ± 8 ppm 4001 - 10000 ppm 10001 - 30000 ppm	3% rel 5% rel 10% rel
NO	0-5000 ppm	1 ppm	±5 ppm	1% rel
O_2	0-25 vol.%	0.01 vol.%	± 0.02% abs	1% rel

4.4 Concluding Remark

This chapter elaborates the experimental setup and the experimental procedure followed for conducting the expriments. The setup is equiped with various instruments for measurement of fuel flow rate, air flow rate, water flow rate, load, speed, pressure-crank angle, temperature and exhaust emissions. These measured data form the basis of performance, combustion and exhaust emissions charactristics of the engine run on different fuels and fuel blends and these are described in the folloing chapters.

Chapter 5

Results of Binary Blends of VO and Diesel

OVERVIEW

The literature review (Chapter 2) reveals a deterioration the CI engine performance with the use of neat VO. There is also an increase in the emissions of CO and HC. This has been attributed to the higher viscosity of VO as compared to diesel. Therefore, rather than using Mesua ferrea Linn oil (a type of VO) directly as a replacement to diesel, the VO is considered to test the stationary CI engine by blending it with diesel at certain proportions. This chapter discusses the results of the experiments carried out on three binary blends of VO-diesel and the neat diesel. Initially, the chapter describes the blend composition, the experimental matrix and the properties of test fuels. This is followed by the performance, combustion and emissions analysis of the engine.

Chapter Outline:

5.1 Introduction

In most of the literature, the blending of VO up to 20% by volume has shown a comparable performance to that of the diesel. Taking this as a guideline, an experimental matrix as shown in Table 5.1 has been designed to investigate the effect of low *Mesua ferrea* Linn oil (MO) content diesel blend on the performance, combustion and emissions characteristics of a stationary CI engine. In this study, two *Mesua ferrea* Linn-diesel blends viz., VO10 and VO20 have been considered for the experimental investigation. The VO10 consists of 10% *Mesua ferrea* Linn oil and 90% diesel by volume, while VO20 is composed of 20% *Mesua ferrea* Linn oil and 80% diesel by volume. Additionally, a blend of 30% *Mesua ferrea* Linn oil and 70% diesel by volume (VO30) has been included in the study to examine the impact of high VO content blend on the engine characteristics. Thus, the test fuels consist of VO10, VO20 and VO30 besides neat diesel. Some of the important properties of the test fuels are shown in Table 5.2. The viscosity of diesel increases with increase in the amount of *Mesua ferrea* Linn oil in the blend. The viscosity of VO10, VO20 and VO30 are 2.776, 3.7 and 4.996 mm^2/s respectively. Experiments have been carried out at the engine loads of 20, 40, 60, 80 and 100%. The outcome of the experiments has been discussed in following subsections under the performance, combustion and emissions analysis.

Table 5.1 Experimental matrix

<table>
<tr><th>Mode</th><th>Fuel used</th><th>CR</th><th>IT</th><th>Speed (rpm)</th><th>Loading conditions (%)</th></tr>
<tr><td>Diesel</td><td>Neat diesel</td><td rowspan="4">17.5</td><td rowspan="4">23° bTDC</td><td rowspan="4">1500</td><td rowspan="4">20, 40, 60, 80, 100</td></tr>
<tr><td rowspan="3">VO-diesel blend</td><td>VO10</td></tr>
<tr><td>VO20</td></tr>
<tr><td>VO30</td></tr>
</table>

Table 5.2 Fuel properties

Properties	Units	Diesel	VO10	VO20	VO30
Calorific value of fuel	kJ/kg	42000*	41361	40722	40083
Density at 15 °C	kg/m^3	828	840	860.4	869.2
Viscosity at 40 °C	mm^2/s	2.036	2.766	3.700	4.996
Stoichiometric air-fuel ratio		15.0	14.7	14.4	14.1

*Sahoo et al (2011)

5.2 Performance Analysis

The performance of the engine is analyzed on the basis of BTE, BSFC and EGT of the engine. Both BTE and BSFC are evaluated considering the mass flow rate of fuel that is consumed at each load the engine, while EGT is recorded from the digital display unit. The BSFC of engine with VO blends follow a trend that is similar to the diesel with decreasing value with the increase in the engine load as shown in Figure 5.1. However, higher the amount of *Mesua ferrea* Linn oil in the blend, the BSFC of the engine becomes higher. The increase in BSFC with blend as compared to diesel is in the range of 3 to 7% with the use of VO20, while it is even more with the use of VO30 in the range of 5 to 11%. This difference is wider with the increasing power output. The VO10 is showing very little difference with that of the diesel. The increase in BSFC with increment in the percentage of VO in the blend has also been reported in the investigations of Pramanik (2003), Hebbal *et al*. (2006), Rakopoulos *et al*. (2006), Agarwal and Agarwal (2007), Bajpai *et al*. (2009), Rakopoulos *et al*. (2011), and Qi *et al*. (2014). This increase in BSFC with the use of VO is due to the lower calorific value of VO. The lower caloric value necessitates the addition of extra fuel to produce a unit power output similar to that of diesel. The higher BSFC of VO blend also results in inducting a richer air fuel mixture into the engine cylinder as compared to that of diesel. The higher is the content of VO in the blends, the richer is the air fuel ratio as depicted in Figure 5.2. The air fuel ratio richer up to 3.2 and 4.7% with the use of VO20 and VO30, respectively. However, the VO10 exhibits a similar trend to that of the diesel except at the lower load. At the load of 20 and 40%, VO10 shows lower equivalence ratio as compared to diesel. The vegetable oil being an oxygenated fuel may have contributed additional oxygen for the combustion.

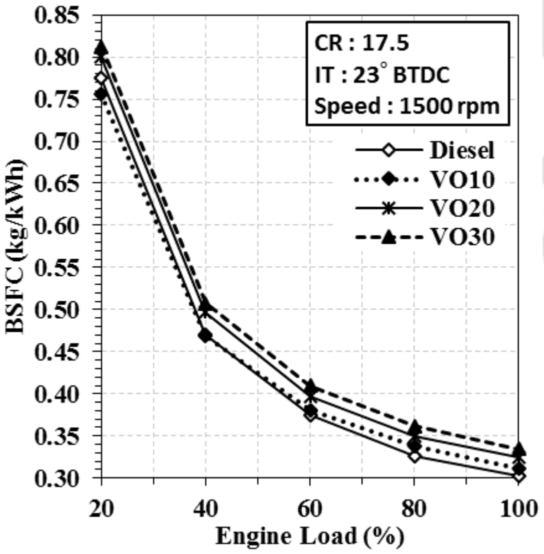


Figure 5.1 Variation of BSFC with the engine load.

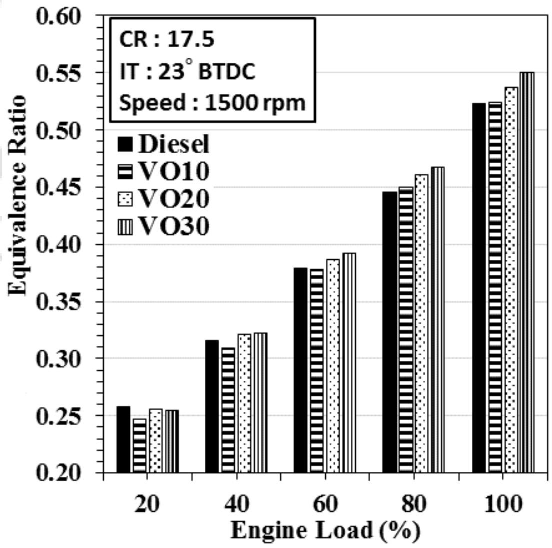


Figure 5.2 Variation of equivalence ratio with the engine load.

The variation of BTE with the engine load for diesel, VO10, VO20 and VO30 are illustrated in Figure 5.3. It increases with the increase in the engine load for all the test fuels. In general, the blending results in the reduction of engine BTE. This reduction increases with the increase in the volume of *Mesua ferrea* Linn oil in the blend. On an average, the reduction in BTE is 1.5, 3.1 and 4.4% with maximum of 1.8, 3.9 and 5.5% for VO10, VO20 and VO30, respectively. The difference in reduction increases with the increasing engine load. The VO has higher viscosity and lower volatility than the diesel which leads to poor combustion characteristics and a reduction in BTE (Pramanik 2003; Hebbal *et al*. 2006; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010; Shah and Ganesh 2016). The higher viscosity results in the formation of larger fuel droplets in the fuel spray leading to inadequate mixing of air and fuel droplets (Agarwal and Rajamanoharan 2009; Singh 2013). The low volatility affects the spray formation inside the combustion chamber resulting in slow combustion (Hebbal *et al*. 2006). Moreover, the higher amount of VO in the blend results in the richer air fuel mixture affecting the combustion process. The VO10 blend shows a comparable performance to that of diesel. At intermediate and lower engine loads, it is slightly better with VO10. Similar typical results of VO blend have been reported in the investigations of Rakopoulos *et al*. (2006), Bajpai *et al*. (2009) and Rakopoulos *et al*. (2011). Bajpai *et al*. (2009) assumes that the better combustion and additional lubricity with VO blend results in this improvement.

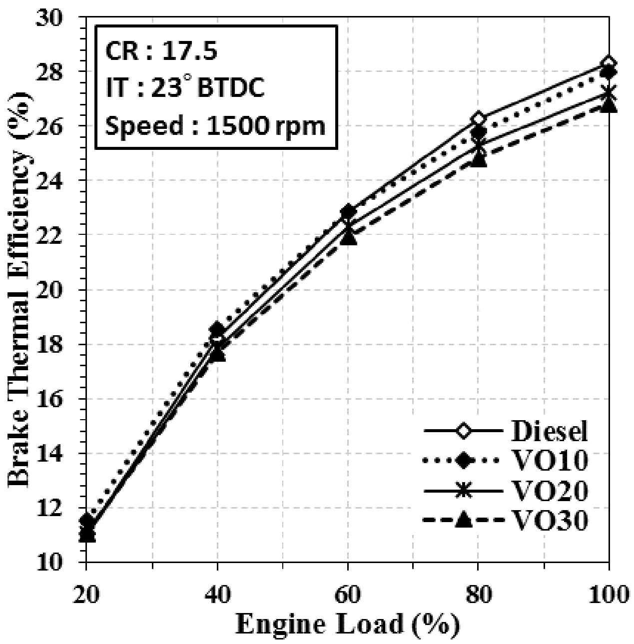


Figure 5.3 Variation of BTE with the engine load.

The blends VO10 and VO20 have registered lower EGT as compared to diesel throughout the operating range of the engine as revealed in Figure 5.4. It tends to increase with the increase in the amount of *Mesua ferrea* Linn oil in the blend. At the 100% engine load, EGT is found to be 344, 353 and 357 °C for VO10, VO20 and VO30, respectively; while diesel has shown the engine EGT to be 359 °C. While using the neat VO in the engine, in most of the studies, it has been found that the EGT is relatively higher than that of diesel (Pramanik 2003; Hebbal *et al*. 2006; Agarwal and Agarwal 2007; Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009). This may be attributed to the presence of higher boiling point components of the VO which do not sufficiently evaporate during the main combustion phase and continues to burn in the later phase of the combustion (Devan and Mahalakshmi 2009). The volumetric efficiency at different load for the test fuels are presented in Figure 5.5. Generally, it decreases with the increasing load. With the increasing load due to the higher pressure of residual gas some part of the suction stroke is utilized in re-expansion of this gas thereby reducing the amount of fresh air inducted. There is a slight drop in the volumetric efficiency with blended fuels.

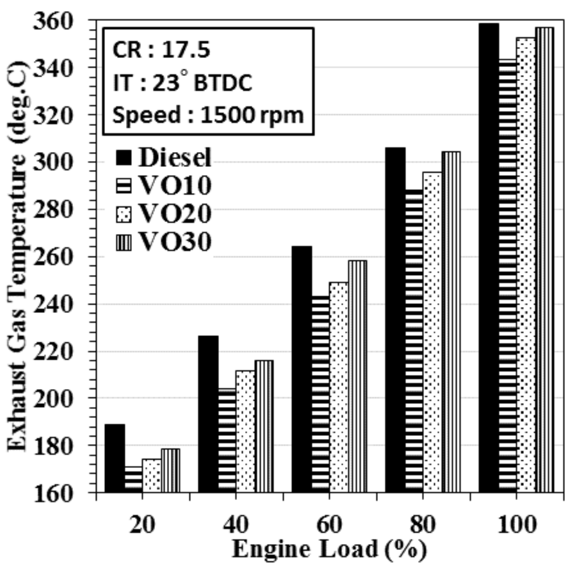


Figure 5.4 Variation of EGT with the engine load.

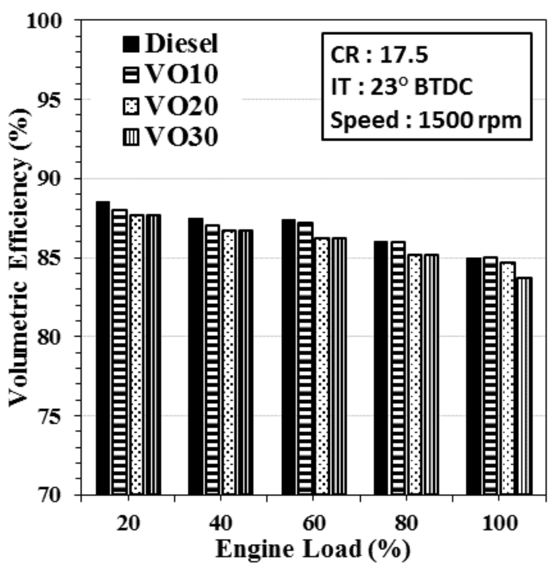


Figure 5.5 Variation of volumetric efficiency with the engine load.

5.3 Combustion Analysis

The cylinder pressure of the test fuels at 60 and 100% load of engine is presented in Figure 5.6. All the fuels exhibit a similar trend of pressure profile across the engine cycle. However, the profile of VO blends outgrows diesel in the early stages combustion which is higher with higher amount of VO in the blend.

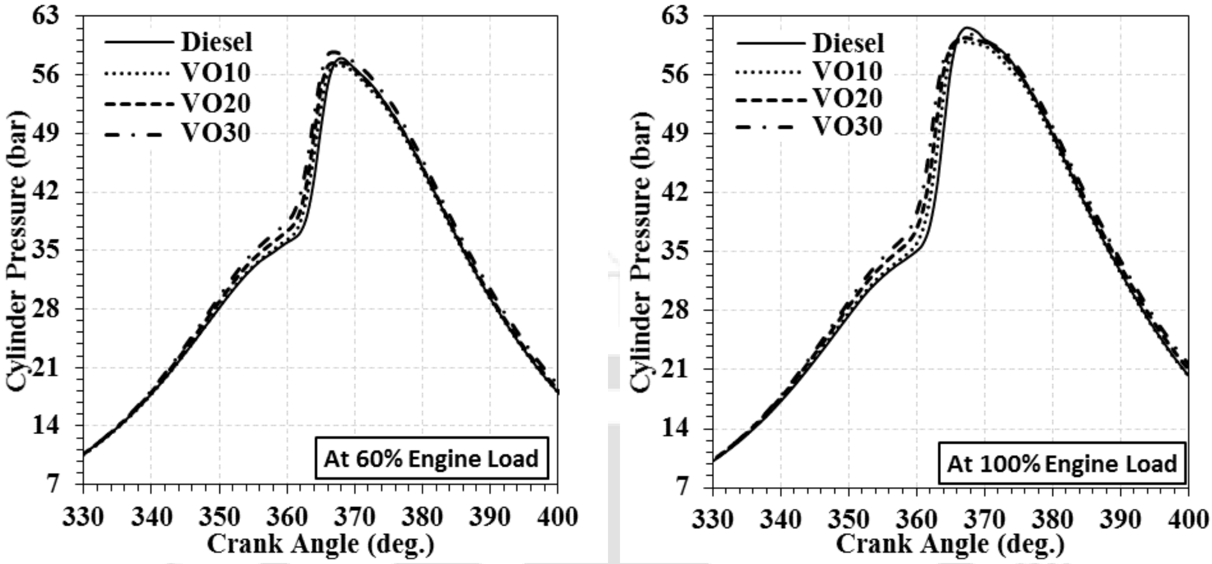


Figure 5.6 Pressure-crank angle diagram of engine at 60 and 100% load for different fuels.

Neat diesel shows a higher rate of pressure rise (RoPR) as compared to that of the VO blends (Figure 5.7 and 5.8). At 100% load, the RoPR is found to be 7.3, 6.5, 6.5 and 6 bar/deg.CA for diesel, VO10, VO20 and VO30, respectively. The differences in the RoPR between the fuels increases with the increase of engine load. At lower loads, all the fuels indicate similar RoPR. The VO blends have earlier start of RoPR compared to that of the diesel. This indicates a shorter ignition delay (ID) with the use of VO blends in comparison to diesel.

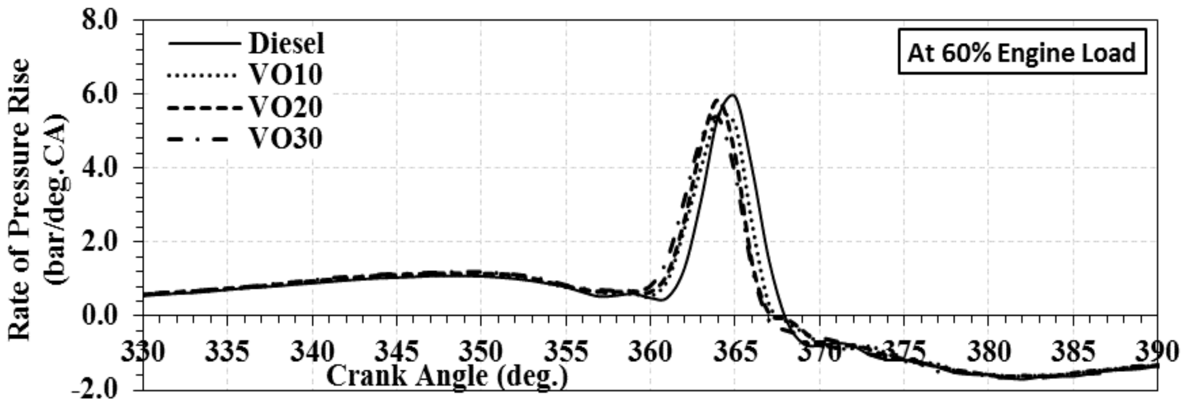


Figure 5.7 Rate of pressure rise diagram of engine at 60% load of engine for different fuels.

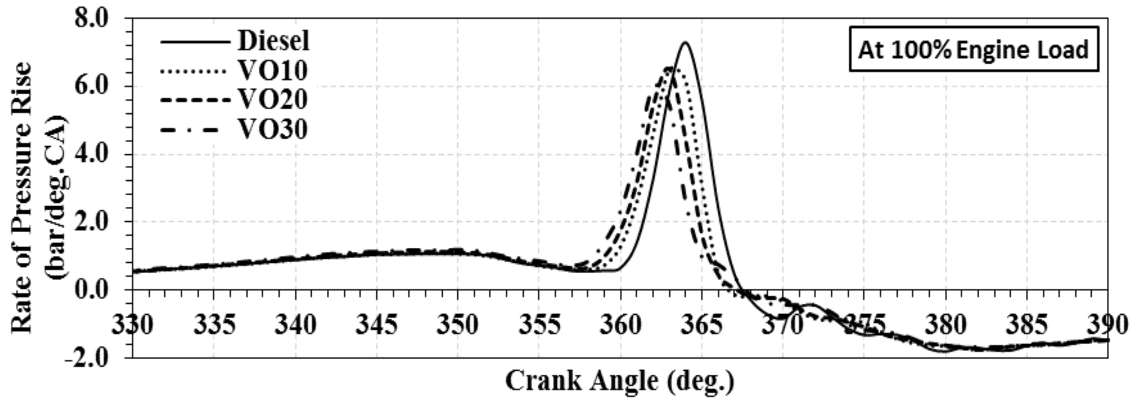


Figure 5.8 Rate of pressure rise diagram of engine at 100% load of engine for different fuels.

The variation of ID with engine is shown in the Figure 5.9, which reveals a decrease in ID with the increase of VO in the blend. At full load of the engine, ID = 21° CA for diesel and VO10, while both VO20 and VO30 have shown ID = 20° CA. For all the fuels, the ID decreases with the increase of engine load. This is due to the increase in temperature of combustion chamber and cylinder wall with the burning of more fuel at higher loads.

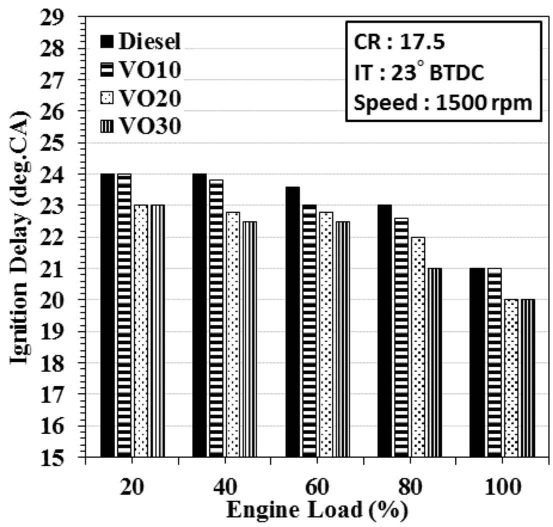


Figure 5.9 Variation of ignition delay with the engine load.

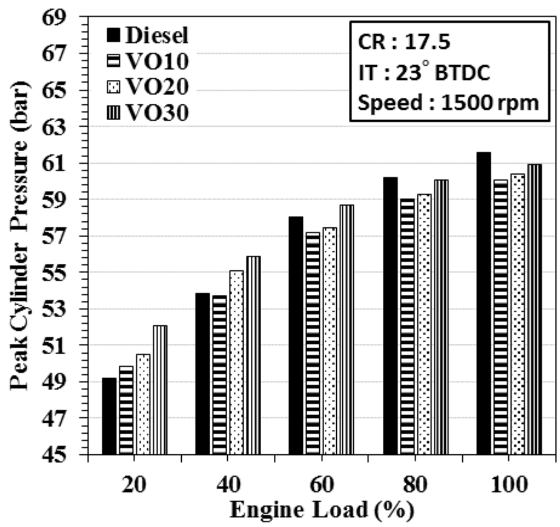


Figure 5.10 Variation of peak cylinder pressure with the engine load.

As demonstrated in Figure 5.10, with the increase of engine load, the PCP also increases for the investigated fuels. The VO blends is displaying a higher PCP as compared to diesel till the 80% of the engine load which is higher with the higher amount of VO in the blend. The shorter ID of VO blends could be responsible higher PCP at the lower and intermediate loads. At 100% load, the PCPs are 61.6, 60.1, 60.4 and 60.9 bar for diesel, VO10, VO20 and VO30, respectively. The net heat release rate (NHRR) diagram for 60 and 100% engine load displayed in Figure 5.11 reflects a drop in heat release with the increase in the share of VO in the blend. This drop is higher with increase in the engine load. This is attributed to the lower calorific value of VO in comparison to that of the diesel. The maximum NHRR at the full load of engine is 96, 84, 84 and 75 J/deg. CA respectively for diesel, VO10, VO20 and VO30. Similar trends of pressure-crank angle and NHRR were reported with the use of poon and soapnut oil blend respectively in the investigations of Devan and Mahalakshmi (2009) and Misra and Murthy (2011). The variation of combustion duration of test fuels is shown in Figure 5.12. It increases with the increase of engine loads. The blended fuel registered a shorter combustion duration than diesel.

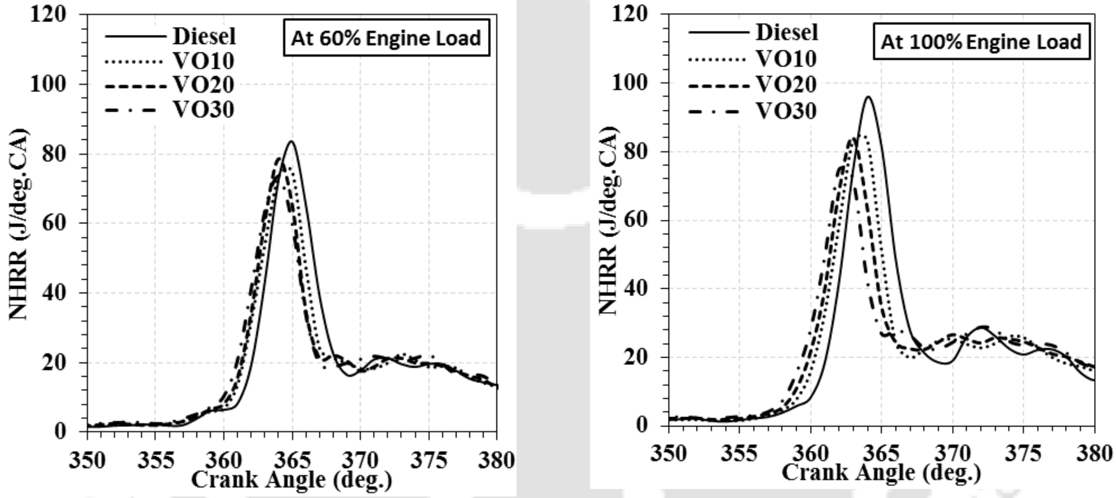


Figure 5.11 NHRR diagram of engine at full load for different fuels.

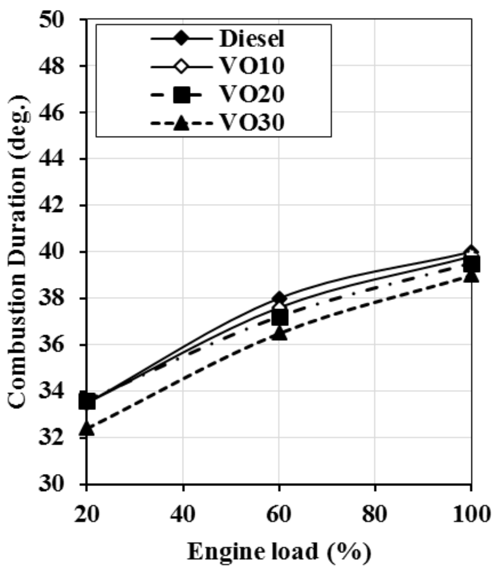


Figure 5.12 Combustion duration of fuels at engine loads.

5.4 Emissions Analysis

The CO emitted at the different engine load while using diesel, VO10, VO20 and VO30 is depicted in Figure 5.13. In overall, it reflects a lower CO emissions as the increase of engine load with little increase at the full load of the engine due to burning of richer fuel-air mixture (Almeida *et al*. 2002; Agarwal and Rajamanoharan 2009). The increasing in the amount of VO volume in the binary blend escalates the production of CO emissions in the engine exhaust. It increases, on an average, by 23 and 39%, respectively by the use of VO20 and VO30 blend in comparison to the diesel.

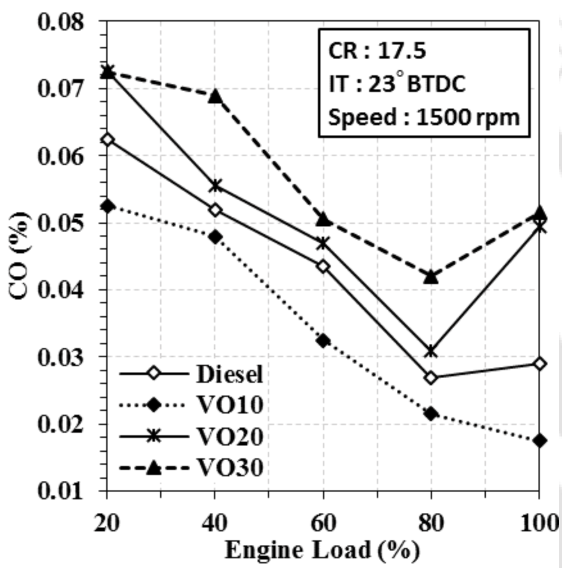


Figure 5.13 Variation of CO emissions with the engine load.

At full load of the engine, the CO emitted by diesel is 0.029 vol.% which is increased to 0.05 and 0.052 vol.% with the use of VO20 and VO30, respectively. The VOs are characterized by high viscosity which inhibits the proper atomization of fuel and triggers a locally rich mixture that consequently leads to incomplete combustion and production CO in the exhaust (Almeida *et al*. 2002; Wang *et al*. 2006; Devan and Mahalakshmi 2009; Agarwal and Rajamanoharan 2009; Chauhan *et al*. 2010; Shah and Ganesh 2016). However, while using VO10 blend in the engine, the CO emissions is found to be lower than that of diesel in the entire operating range. Similar observations of lower CO emissions as compared to diesel with lower VO blends have been reported in the studies of Bajpai *et al*. (2009), Misra and Murthy (2011) and Qi *et al*. (2014) with the use of VO blend derived from karanja, soapnut and rapeseed, respectively. The higher availability of oxygen in the fuel is believed to be the possible cause for lower CO emissions in the lower VO blends (Misra and Murthy 2011).

Figure 5.14 represents the NO emitted at various loads when the engine is run on the test fuels. The NO increases with the increase in the engine load. The injection of more fuel with increasing load results in higher gas temperature in the combustion chamber leading to production of more NOx (Huang *et al*. 2009). With the use of VO blend in the engine, the NO emissions is found to reduce especially in the higher engine loads. Higher is the amount of VO in the blend, the higher is the reduction of NO. At full load, it reduces from 657 ppm with diesel to 637, 567 and 493 ppm with the use of VO10, VO20 and VO30 blend, respectively. This may be due to the lower amount of heat released with VO blends compared to the diesel (Figure 5.11) and resulting lower in-cylinder temperature in engine cycle which reduces the NO formation during the combustion.

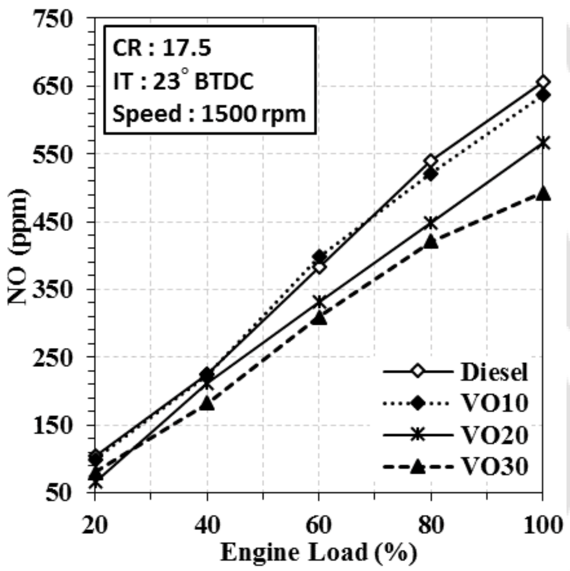


Figure 5.14 Variation of NO emissions with the engine load.

The comparative results of unburned HC emitted by the engine with the use of different fuels are shown in Figure 5.15. It shows an increase in the amount HC emissions with the increasing load irrespective of fuels used. This has been attributed to the injection of more fuel and lesser availability of oxygen for reaction with the increasing engine load (Almeida *et al*. 2002; Wang *et al*. 2006; Cetin and Yuksel 2007; Agarwal and Rajamanoharan 2009; Chauhan *et al*. 2010). The VO blends show a higher HC emissions as compared to that of the diesel. This increase is in the range of 3 to 14%, 7 to 46% and 17 to 95% which on an average is 7, 19 and 38% with 10, 20 and 30% VO blends, respectively as compared to diesel. The higher viscosity and low volatility of VO is assumed to be causes of higher HC emissions as they affect the fuel droplet size and spray development in the combustion chamber leading to slow combustion (Hebbal *et al*. 2006; Devan and Mahalakshmi 2009). The CO_2 emissions presented in Figure 5.16 shows an increasing emission with the increase of engine operating loads irrespective of the fuels used. Diesel mode gives out a little higher CO_2 emissions as compared to that of the VO blends due to better combustion with diesel. The CO_2 emissions increases with the increasing percentage of VO in the blend. This could be due to the shorter ID with increase in VO in the blend that facilitates the longer combustion period.

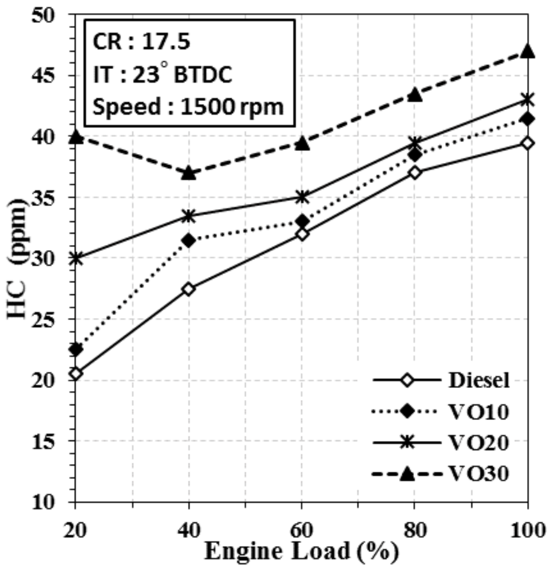


Figure 5.15 Variation of HC emissions with the engine load.

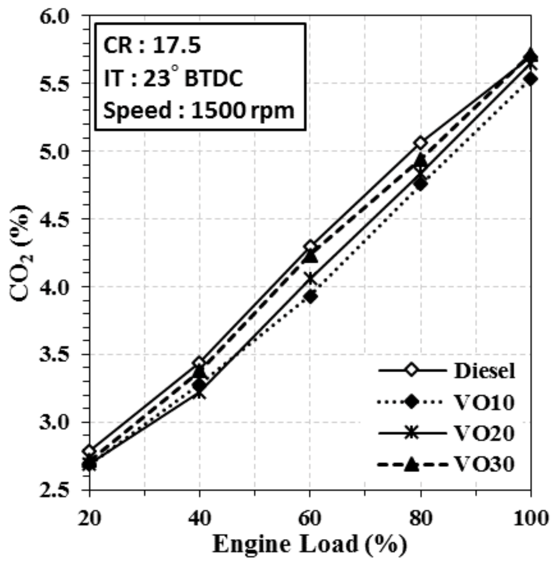


Figure 5.16 Variation of CO_2 emissions with the engine load.

5.5 Summary

This experimental investigation is aimed at exploring the applicability of non-edible VO, derived from seeds of *Mesua ferrea* Linn, to supplement the energy need of CI engines by blending *Mesua ferrea* Linn oil with diesel at the volume percentage of 10, 20 and 30%. The key findings of the study are summarized below:

- The study demonstrates a slight drop in BTE with the use of *Mesua ferrea* Linn oil and this is up to 1.78, 3.94 and 5.47% with the use of 10, 20 and 30% blend respectively. The VO10 blend shows almost the same BTE as that of the diesel.
- Due to the lower calorific value of *Mesua ferrea* Linn oil, the BSFC increases with blending and is found in the range of 1 to 3%, 3 to 7% and 5 to 11% respectively with 10, 20 and 30% blending.

- In comparison to diesel, the *Mesua ferrea* Linn oil blends produce higher CO and HC emissions. The CO emissions increases, on an average, by 23 and 39% respectively with the blending by 20 and 30%. The increase in the HC emissions is in the range of 3 to 14%, 7 to 46% and 17 to 95% with 10, 20 and 30% oil blend respectively.
- With the use of oil blend, the NO emissions is found to reduce especially in the higher engine loads. This reduction is higher with higher is the amount of oil in the blend. At full load, it reduces from 657 ppm with diesel to 637, 567 and 493 ppm with the use of VO10, VO20 and VO30 blend, respectively.
- The use of VO derived from the seeds of *Mesua ferrea* Linn has the scope of supplementing the energy need of CI engines by blend it with diesel. The diesel blended with 10% VO has almost similar performance to that of neat diesel.

Chapter 6

Results of Ternary Blend of VO, Diesel and Ethanol

OVERVIEW

In Chapter 5, it has been observed that with the increasing content of Mesua ferrea Linn oil (a type of VO) in the oil-diesel binary blend, the engine performance tends to decline. This has been due to the influence of VO viscosity that leads to the poor combustion thereby producing a larger amount of CO and HC in the exhaust. An additive is therefore added to the blend to reduce the viscosity of the blend. In this part of the investigation, ethanol which of biomass origin is used as the additive. This chapter discusses the results of the experiments carried out on ternary blend consisting of VO, ethanol and diesel. The chapter begins with the blend composition, the experimental matrix and the properties of test fuels. This is followed by the performance, combustion and emissions analysis of the engine.

Chapter Outline:

6.1 Introduction

The investigation of *Mesua ferrea* Linn oil blends (VO10, VO20 and VO30) reveals a deterioration in the engine performance with increase in the amount of *Mesua ferrea* Linn oil in the blend. This is mainly attributed to the inferior quality of oil in comparison to that of diesel. The higher viscosity and density of *Mesua ferrea* Linn oil poses a hindrance to go for higher composition of *Mesua ferrea* Linn oil in the binary blend with diesel. This necessitates a need of an alternative arrangement for overcoming this limitation. To do away with this hindrance, the simplest and best way is through the use of an additives in the binary blend. These additives have lower viscosity and density as compared to the components of binary blend. Alcohols and ethers are generally used as additives in the binary blends consisting of diesel and VO or VO derived fuels such as biodiesel. Ternary blends consisting of diesel, VO and additive reduces the fuel viscosity and density. The most commonly used alcohols are ethanol and butanol because of their renewable biomass origin. In this part of the experimental investigation, ethanol is being used as an additive in the *Mesua ferrea* Linn oil-diesel binary blend.

Table 6.1 Experimental matrix.

Module No.	Mode	Fuel used	CR	IT	Speed (rpm)	Loading conditions (%)
I	Diesel	Neat diesel	17.5	23^o bTDC	1500	20, 40, 60, 80, 100
	Binary blend	VO20				
	Ternary blend	VO20E05				
		VO20E10				
II	Diesel	Neat diesel	17.5	23^o bTDC	1500	20, 40, 60, 80, 100
	Binary blend	VO30				
	Ternary blend	VO30E05				
		VO30E10				

Two modules (Modules I and II) of experimental investigation is carried out and the details of experimental matrix is shown in Table 6.1. In Module I, ethanol is added to VO20 blend to form a two types of ternary blend, first one consists of 5% ethanol and 95% VO20 by volume (VO20E05) and second one consists of 10% ethanol and 90% VO20 by volume (VO20E10). While in Module II, ethanol is added to VO30 blend to form a two types of ternary blend, first one consists of 5% ethanol and 95% VO30 by volume (VO30E05) and second one consists of 10% ethanol and 90% VO30 by volume (VO30E10). The properties of test fuels for Modules I and II are shown in Tables 6.2 and 6.3, respectively. The addition of ethanol in VO20 and VO30 decreases the viscosity as compared to neat VO20 and VO30. For both the

Modules, experiments have been carried out at the engine loads of 20, 40, 60, 80 and 100% to study the effect of ethanol (as an additive to binary blend) on the performance, combustions and emissions characteristics of the CI engine. The results of the investigations are discussed in the following sub sections.

Table 6.2 Fuel properties for Module I.

Properties	Units	Diesel	Ethanol	VO20	VO20E05	VO20E10
Calorific value of fuel	kJ/kg	42000*	26800#	40722	40026	39390
Density at 15 °C	kg/m^3	828	788#	860.4	857.1	853.7
Viscosity at 40 °C	mm^2/s	2.036	1.2#	3.700	3.575	3.450
Stoichiometric air-fuel ratio		15	9	14.4	14.0	13.6

*Sahoo et al (2011); #Rakopoulos et al (2015)

Table 6.3 Fuel properties for Module II.

Properties	Units	Diesel	Ethanol	VO30	VO30E05	VO30E10
Calorific value of fuel	kJ/kg	42000*	26800#	40083	39418.85	38754.7
Density at 15 °C	kg/m^3	828	788#	869.2	865.14	861.08
Viscosity at 40 °C	mm^2/s	2.036	1.2#	4.996	4.806	4.616
Stoichiometric air-fuel ratio		15	9	14.1	13.7	13.4

6.2 Module I with VO20

6.2.1 Performance Analysis

The variations in the BTE of the test engine with the use of diesel, VO20, VO20E05 and VO20E10 at the operating loads have been illustrated in Figure 6.1. It indicates an increase in the BTE of engine with an increase in the load for all the tested fuels. The use of ethanol in VO20 blend shows an improvement in the BTE which is found to be better with the higher amount of ethanol in the blend. This can be attributed to the lower viscosity and higher oxygen content in the blend due to the use of ethanol that improves the combustion process (Sathiyamoorthi and Sankaranarayanan 2017; Qi *et al*. 2017; Paul *et al*. 2017; Saleh and Selim 2017; Prakash *et al*. 2018). With the use of ethanol, the viscosity of the blend reduces which facilitates a better atomization of fuel leading to formation of better combustible charge as compared to that of VO20. Moreover, ethanol being an oxygenated fuel provides additional oxygen for combustion. With the use of 10% ethanol in the VO20 blend, BTE of VO20 increases from 11.0 to 11.5%, 17.8 to 18.0%, 22.3 to 22.5%, 25.3 to 25.4% and 27.2 to 27.7% at 20, 40, 60, 80 and 100% load, respectively as compared to VO20. The increase in the BTE is in the range of 0.1 to 3.5% with the 5% ethanol blending. While it is in the range of 0.6 to 4% with the 10% ethanol blending. As shown in Figure 6.2, with the use of ethanol in the VO20, the fuel-air mixture is leaner which facilitates the formation of a better combustible mixture and improves the combustion. This formation of leaner mixture is attributed to the reduction of density of fuel due to the addition of ethanol.

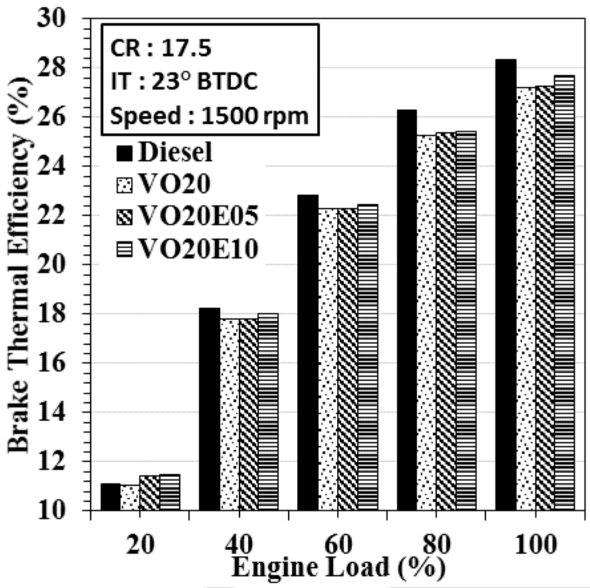


Figure 6.1 Variation of BTE with the engine load.

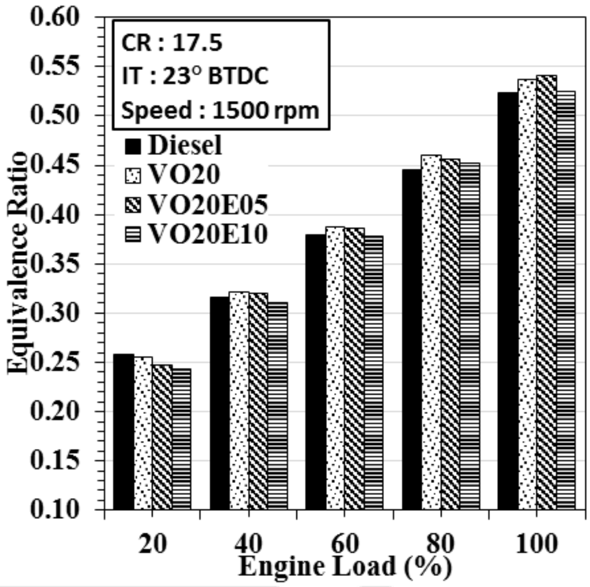


Figure 6.2 Variation of equivalence ratio with the engine load.

The BSFC of all the test fuels decrease with the increase of engine load as revealed in Figure 6.3. The combustion temperature increases with the increasing engine load which results in the faster rate of increase in the brake power than the increase in fuel consumption leading to lower BSFC with the increasing load (Qi *et al*. 2014). The BSFC of engine increases with the use of ethanol in the blend, and it is higher with higher in amount of ethanol in the blend. However, this increment is almost negligible which on an average of 1.7 and 2.5% respectively with the use of 5 and 10% ethanol in the VO20 blend. The lower heating value of ethanol in comparison to diesel and VO is responsible for the increase in BSFC with VO20E05 and VO20E10 blends. The lower calorific value of fuel necessitates more amount of fuel to produce the same amount of power as compared to higher calorific value fuel of diesel. The use of ethanol blend results in the slight increase in engine EGT as compared to that of the VO20 blend as depicted in Figure 6.4. The increase is in the range of 1.3 to 4.8% and 2.1 to 5.4%, respectively with the use of VO20E05 and VO20E10 blends. This could be due to the cooling effect of ethanol which prevents the combustion of some fraction of fuel during main combustion phase and the burning of this fuel during later phase resulting in increase in EGT. An increase in EGT with the use of ethanol by 2.5 and 5% in the binary blend LGO25, consists of 25% neat lemongrass oil and 75% diesel, has also been reported in the experimental investigation of Sathiyamoorthi and Sankaranarayanan (2017). Figure 6.5 depicts the variation of volumetric efficiency at different load for the test fuels. The volumetric efficiency decreases with the increasing load for all the tested fuels. The ethanol blended fuel shows a marginal decrease in efficiency as compared to VO20 blend.

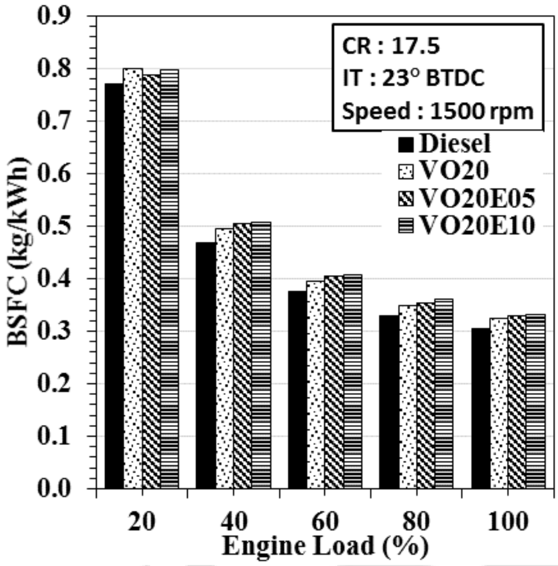


Figure 6.3 Variation of BSFC with the engine load.

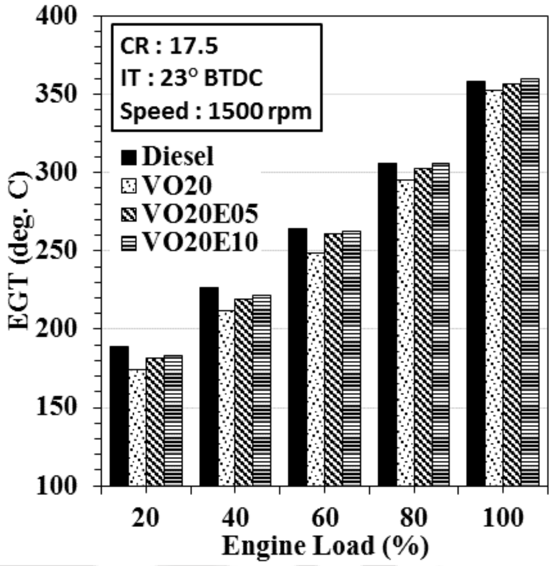


Figure 6.4 Variation of EGT with the engine load.

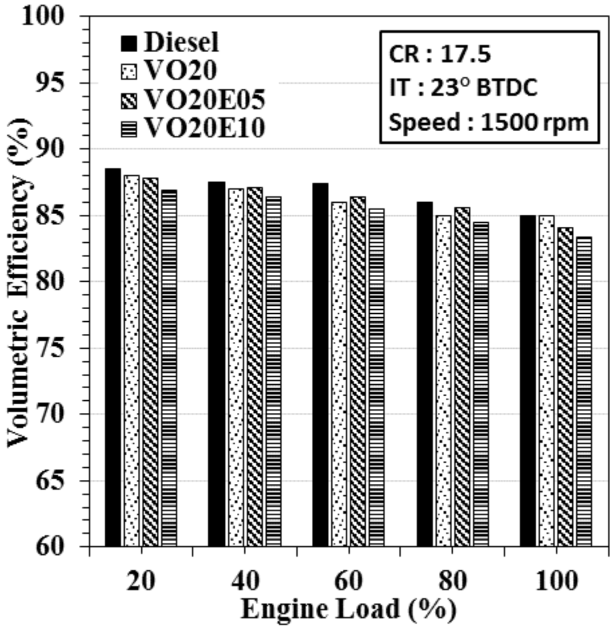


Figure 6.5 Variation of volumetric efficiency with the engine load.

6.2.2 Combustion Analysis

The cylinder pressure at various crank angles at 60 and 100% loads is shown in Figure 6.6. All the test fuels indicate similar trend of pressure variation across the given period of engine cycle irrespective of the engine load. At the intermediate load of 60%, the ethanol blended fuels show an earlier onset of pressure rise as compared to that of VO20. This indicates an earlier start of combustion or shorter ID of ethanol blend as compared to diesel and VO20. This may be attributed to the lower boiling of the ethanol which vaporizes in the earlier stage of combustion enabling an earlier combustion of fuel.

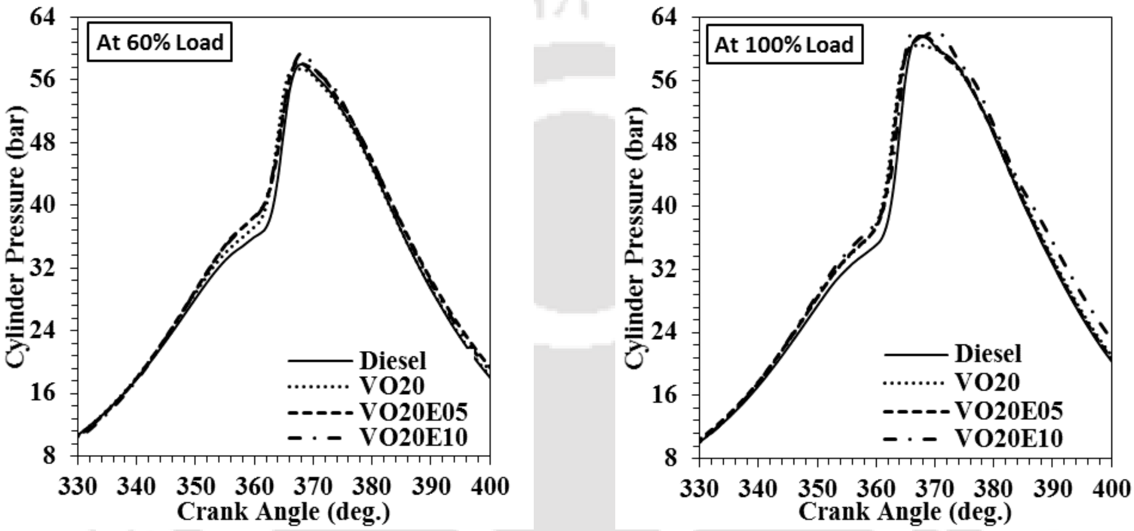


Figure 6.6 Pressure-crank angle diagram of engine at 60 and 100% load for different fuels.

However, at the full load, both ethanol blended VO20 and neat VO20 show a similar trend of pressure variation because of the prevailing of high combustion temperature. Although the ethanol blends have shorter ID in comparison to VO20 and diesel, the RoPR of ethanol blends is lower as indicated in Figures 6.7 and 6.8. This could be due to the lower calorific value of the ethanol blends as compared to the other fuels. At the 60% engine load, the peak RoPR is 5.9, 5.8, 5 and 4.6 bar/deg.CA; while at 100% load, it is 7.3, 6.5, 6.7 and 7 bar/deg.CA respectively for diesel, VO20, VO20E05 and VO20E10. The IDs of test fuels at various loads of the engine are shown in Figure 6.9. It generally decreases with the increase of load due to the increase in the combustion temperature with the increasing load. Both VO20 and VO20E05 exhibit similar IDs in the most of the loads, while VO20E10 has shortest ID. The addition of ethanol reduces the viscosity of VO20 leading to better atomization of fuel. Since the vegetable oil is a complex compound composed various types of hydrocarbons, the better atomization might have resulted in the immediate combustion of highly volatile components and reduces the ID. In comparison to VO20 blend, the PCP of ethanol blended VO20 is higher which is greater with the amount of ethanol in the blend (Figure 6.10). At 60% load, the PCP increases from 57.4 bar with VO20 to 58.2 and 59.5 bar with VO20E05 and VO20E10, respectively. While with 60.4 bar with VO20, it increases to 61.6 and 62.2 bar, respectively with VO20E05 and VO20E10 at the 100% load of the engine. This may be due to the better combustion with the ethanol in the blend.

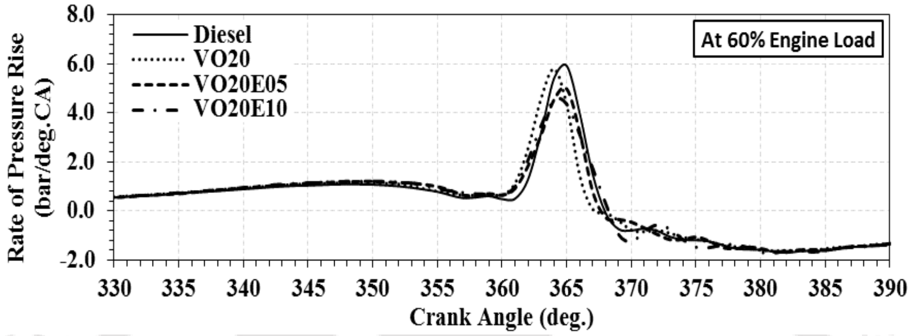


Figure 6.7 Rate of pressure rise diagram of engine at 60% load of engine for different fuels.

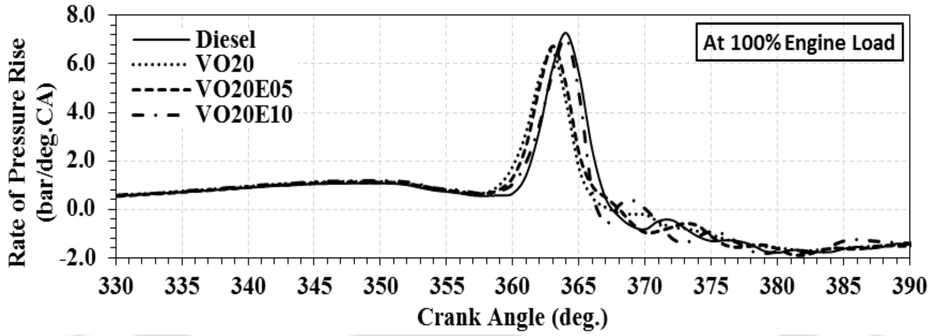


Figure 6.8 Rate of pressure rise diagram of engine at 100% load of engine for different fuels.

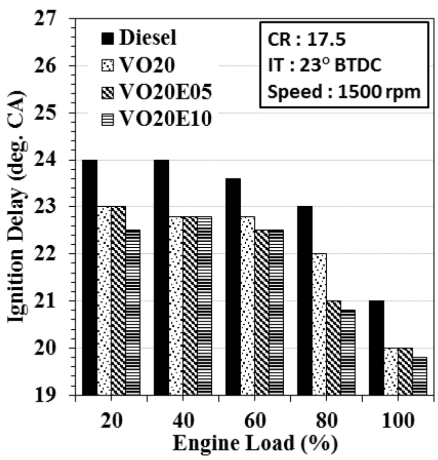


Figure 6.9 Variation of ignition delay with the engine load.

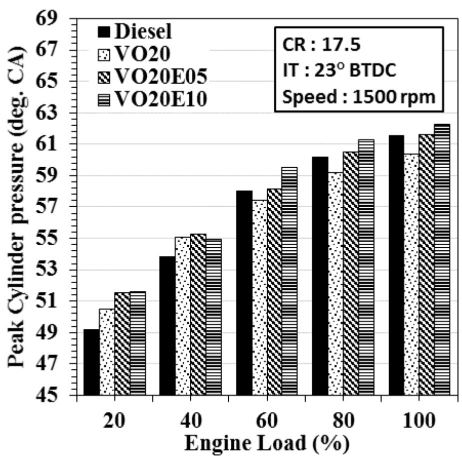


Figure 6.10 Variation of peak cylinder pressure with the engine load.

Figure 6.11 illustrates the NHHR diagrams of the test fuels at 60 and 100% loads of the engine. It reflects an increasing influence of ethanol on the combustion and subsequent heat release rate with the increasing load. At 60% load, the peak NHRR is 83.6, 78.6, 73.0 and 65.7 J/deg.CA with diesel, VO20, VO20E05 and VO20E10, respectively; whereas at 100% load, this increases to 96.0, 84.1, 86.7 and 94.0 J/deg.CA for diesel, VO20, VO20E05 and VO20E10, respectively. This higher NHRR of ethanol have resulted in an improvement of BTE with VO20E05 and VO20E10 as compared to VO20. This higher NHRR with ethanol in VO-diesel blend has also been reported in the studies of Qi *et al*. (2017) and Sathiyamoorthi and Sankaranarayanan (2017). The variation of combustion duration with the engine load is depicted in Figure 6.12. The combustion duration increases with the addition of ethanol as compared to neat VO20.

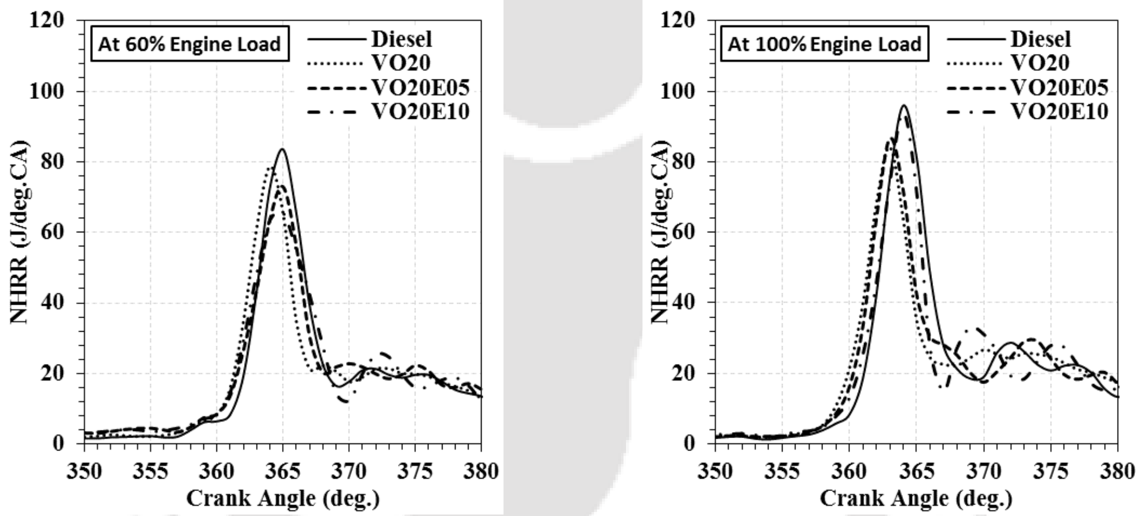


Figure 6.11 NHRR diagram of engine at 60 and 100% engine load for different fuels.

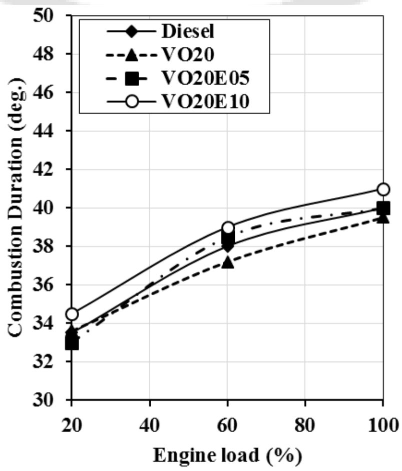


Figure 6.12 Combustion duration of the tested fuels.

6.2.3 Emissions Analysis

The CO emissions associated with diesel, VO20, VO20E05 and VO20E10 at the engine loads of 20, 40, 60, 80 and 100% are demonstrated in Figure 6.13. The test fuels are showing a decrease in the emissions with increase in load having the least value at 80% load. This is due to an improvement in the combustion process with the increasing load. There is a slight increase CO emissions at 100% load as richer fuel-air mixture is burned resulting in a higher CO production. The addition of ethanol in VO20 blend reduces the CO emissions. It reduces from 0.073 to 0.061%, 0.056 to 0.053%, and 0.047 to 0.044% and 0.050 to 0.034%, with the use of 10% ethanol in the blend (VO20E10) at the loads of 20, 40, 60 and 100%, respectively as compared to VO20. While with 5% ethanol in the blend, the CO emissions reduces to 0.053, 0.042, 0.030, 0.021 and 0.024% at the engine loads of 20, 40, 60, 80 and 100%, respectively. The addition of ethanol increases the oxygen content and reduces the viscosity of fuel resulting in a better combustion and reduction of CO emissions (Qi *et al*. 2014; Sathiyamoorthi and Sankaranarayanan 2017).

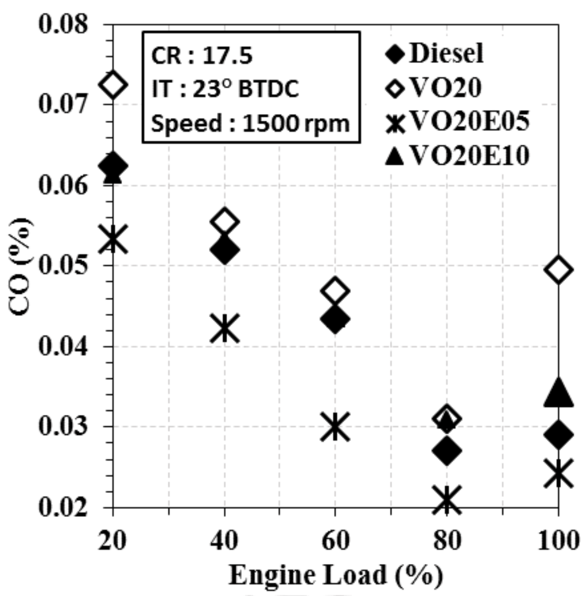


Figure 6.13 Variation of CO emissions with the engine load.

The NO emissions from engine increases with the increasing load irrespective of type of fuel used as revealed in Figure 6.14. This is due to the increase of combustion temperature with the increasing load. Diesel and VO20E05 blend respectively have registered a highest and lowest NO emissions throughout the operating ranges of the engine. The NO emitted by diesel is in the range of 105 to 657 ppm, while it is 28 to 485 ppm for VO20E05 blend. There is a drastic reduction in the NO emissions at 5% ethanol blend of VO20. This is possibly due to the cooling effect resulted from higher latent of vaporization and lower calorific value of ethanol that reduces the combustion temperature leading to production of lower NOx (Qi *et al*. 2014). The NO emissions increases, in the range of 86 to 547 ppm as compared to the range of 28 to 485 ppm as the ethanol content in VO20 increases from 5 to 10%.

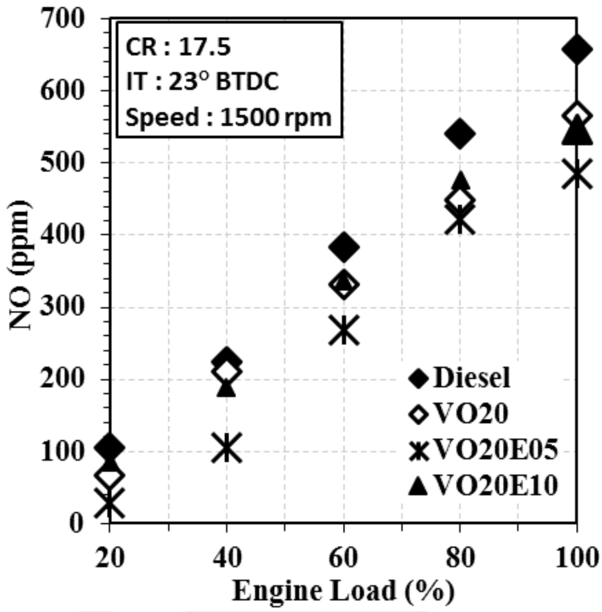


Figure 6.14 Variation of NO emissions with the engine load.

The unburned HC emitted in the engine exhaust with the use of tested fuels is illustrated in Figure 6.15. With the use of 10% ethanol in the blend, the engine produces a higher HC emissions as compared to other fuels. The increase is in the range of 6 to 13% as compared to VO20. This increase in HC emissions with ethanol is believed to be due the expanding of the lean spray flame out region at ID period because of presence of ethanol which is an oxygen bound fuel (Rakopoulos *et al*. 2015). However, 5% ethanol blend has registered the lowest HC emissions at 20 to 60 % engine loads. Similar results of low and high HC emissions respectively are observed in the study of Sathiyamoorthi and Sankaranarayanan (2017). As illustrated in Figure 6.16, the CO_2 emissions of the engine increases with the increase of engine load for all the test fuels. The diesel mode is showing higher emissions as compared to the other fuels throughout the tested loads. At the loads of 20 and 40%, the VO20, VO20E05 and VO20E10 blends exhibit almost a similar CO_2 emissions. However, with the increasing load, the CO_2 emissions from ethanol blended fuels are found to be lower as compared to VO20 blend. The emissions increases with the increase in the volume of ethanol in VO20.

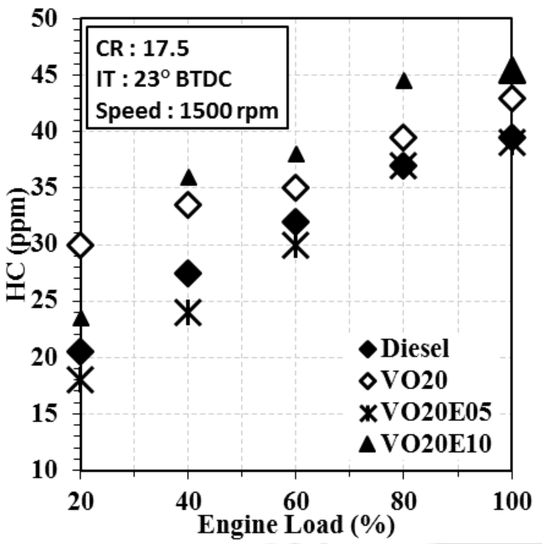


Figure 6.15 Variation of HC emissions with the engine load.

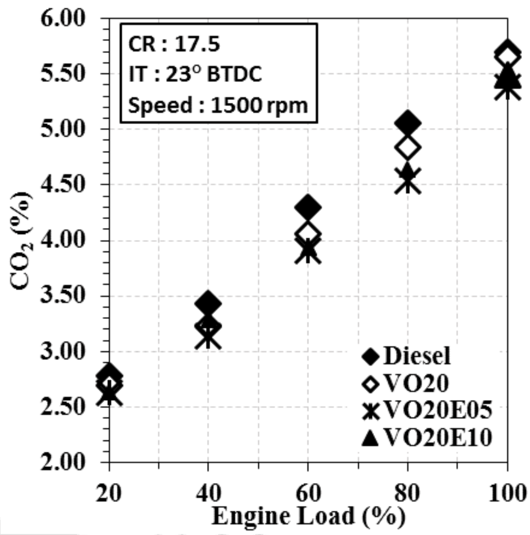


Figure 6.16 Variation of CO_2 emissions with the engine load.

6.2.4 Summary of Module I

The blending of 5 and 10% ethanol to *Mesua ferrea* Linn oil-diesel blend composed of 20% *Mesua ferrea* Linn oil and 80% diesel by volume reveals the following:

- The use of ethanol is found to have slight improvement in the engine performance as compared to that of VO20. With the use of 10% ethanol in the VO20 blend, the BTE of VO20 increases from 1 to 4%; while with VO2OE05 blend, the increase of BTE is in the range of 0.1 to 3.5%.
- The BSFC of engine increases with the use of the ethanol in the V2O blend and it is higher with higher in amount of ethanol in the blend. On an average, the increment is 1.7 and 2.5%, respectively with the use of 5 and 10% ethanol in the VO20 blend.
- There is a slight increase in EGT with use of ethanol as compared to VO20. The increase is in the range of 7 to 12 ˚C and 8 to 14 ˚C, respectively with the use of 5 and 10% ethanol blend.
- The CO emissions of VO20 decreases and NO increases with the use of ethanol. However, HC emissions decreases with 5% ethanol addition and increases with the use of 10% ethanol in the VO20 blend.

6.3 Module II with VO30

6.3.1 Performance Analysis

The BSFC of the test fuels at various loads of the engine is shown in Figure 6.17. It indicates a reduction in the BSFC with the increasing load for all the test fuels. This has been attributed to the increase of combustion temperature in the combustion chamber with increasing load. This has resulted a higher rate of increase in brake power as compared to increase in the fuel consumption with increasing load (Qi *et al*. 2016). The addition of ethanol in VO30 leads to higher BSFC and the value is higher with the larger fraction of ethanol in the blend. In comparison of VO30, the average increase in the BSFC for VO30E05 is 1.8% with maximum increase of 4.4% at 40% load. While it is 3% for VO30E10 with maximum increase of 5.2% at 40% engine load. As VO30E05 and VO30E10 blends have lower calorific value as compared to VO30 blend, this imposes a need of consuming higher amount of VO30E05 and VO30E10 fuel to produce a same unit of brake power in the output. The BTE of the tested fuels depicted in Figure 6.18 shows an increase in the BTE of engine with the increasing load. Diesel mode recorded a higher BTE compared to other fuels throughout the entire range of experiment. The blending of 5% ethanol to VO30 (VO30E05) showing an increase in the BTE which is in the range of 0.3 to 1.3% as compared to neat VO30. With the use of VO30E10, the BTE increases by 0.5 to 1.5%. It increases by 1.5, 0.5 and 1.2% compared to VO30 at the engine loads of 60, 80 and 100%, respectively. This indicates a significant influence of ethanol on VO30. The addition of ethanol in VO30 reduces the viscosity and density of fuel. This leads to an improved atomization of the fuel and formation combustible mixture thereby improving the combustion.

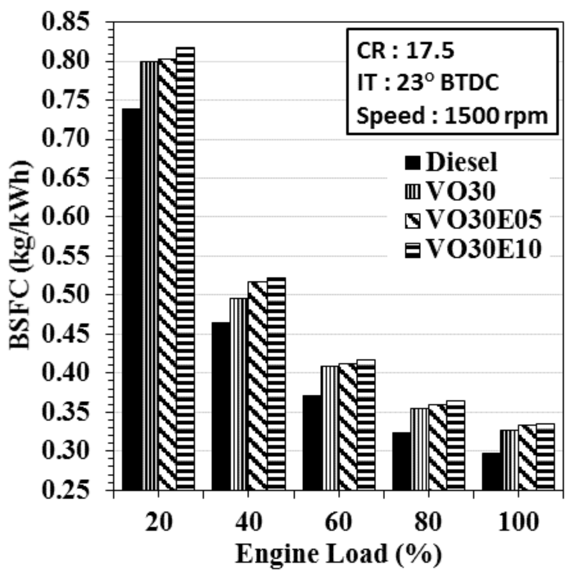


Figure 6.17 Variation of BSFC with the engine load.

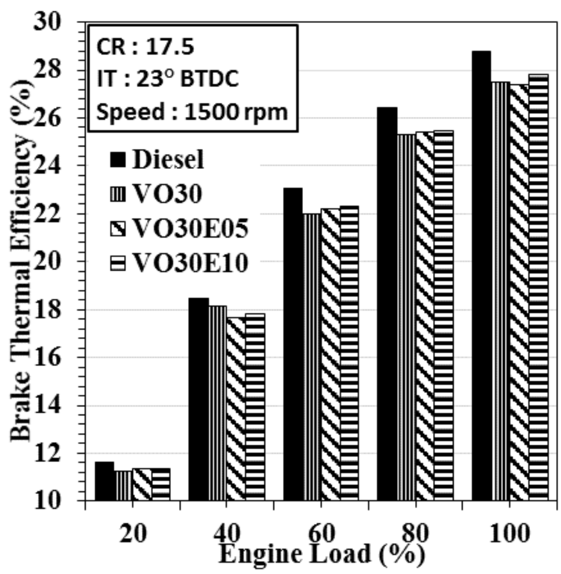


Figure 6.18 Variation of BTE with the engine load.

Moreover, the ethanol blending results in the formation of leaner fuel air mixture, as revealed in Figure 6.19, as compared to VO30 due to the reduction in the density of fuel and helps in the better combustion. Equivalence ratio decreases by 0.3, 2.1 and 2.4% with the use of VO30E10 as compared to VO30 for the engine loads of 60, 80 and 100%, respectively. Figure 6.20 represents the variation of EGT at various loads of the engine. In general, the VO blends display higher EGT in comparison to diesel. This has been attributed to the presence of higher boiling constituents in the VOs which do not effectively evaporate during the main combustion phase and this constituents continue to burn in the late combustion phase resulting in a higher EGT (Devan and Mahalakshmi 2009). Prakash *et al*. (2018) assumes that the poor volatility and higher viscosity with slow combustion of VOs is responsible for higher EGT. The ethanol blended VO30 does not show a remarkable consistency in EGT trend. It varies from a little higher to lower values as compared to neat VO30. This could be due to the lower percentage of ethanol in the VO-diesel to cause any remarkable effect on EGT. Prakash *et al*. (2018) reported a lower EGT with the use 20 and 30% ethanol in VO-diesel blend. The variation of volumetric efficiency of the engine at various loads for the test fuels are shown in Figure 6.21. It decreases with the increasing load irrespective of the fuel used due to the higher pressure of residual gas with the increasing engine load. The blended fuels show almost the similar volumetric efficiency. Diesel mode has a higher volumetric efficiency as compared to the blended fuels.

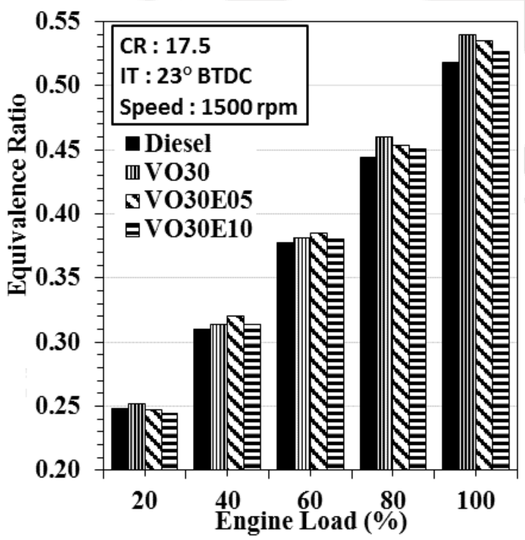


Figure 6.19 Variation of equivalence ratio with the engine load.

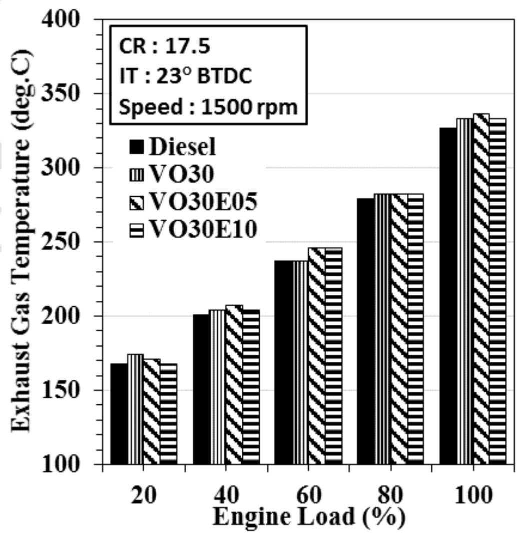


Figure 6.20 Variation of EGT with the engine load.

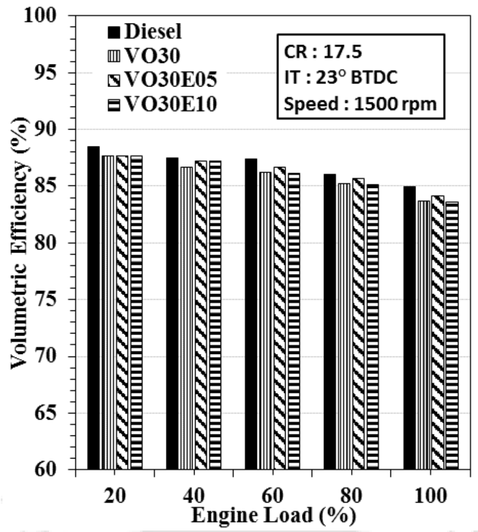


Figure 6.21 Variation of Volumetric efficiency with the engine load.

6.3.2 Combustion Analysis

The in-cylinder pressure variations of tested fuels with engine crank angle (CA) at the 60% (part load) and 100% (full load) of engine load is illustrated in Figure 6.22. All the tested fuels exhibits a similar pressure profile trend across the given range of the engine crank angle. The VO30 maintains a lower cylinder pressure profile in comparison to the other fuels especially in the full load of the engine. This could be due to the higher viscosity of VO30 which results in the formation larger fuel droplets affecting the proper combustion of the fuel. With the addition of ethanol in VO30, the combustion of fuels improves leading the increase in the cylinder pressure, in comparison to neat VO30, as revealed in Figure 6.22.

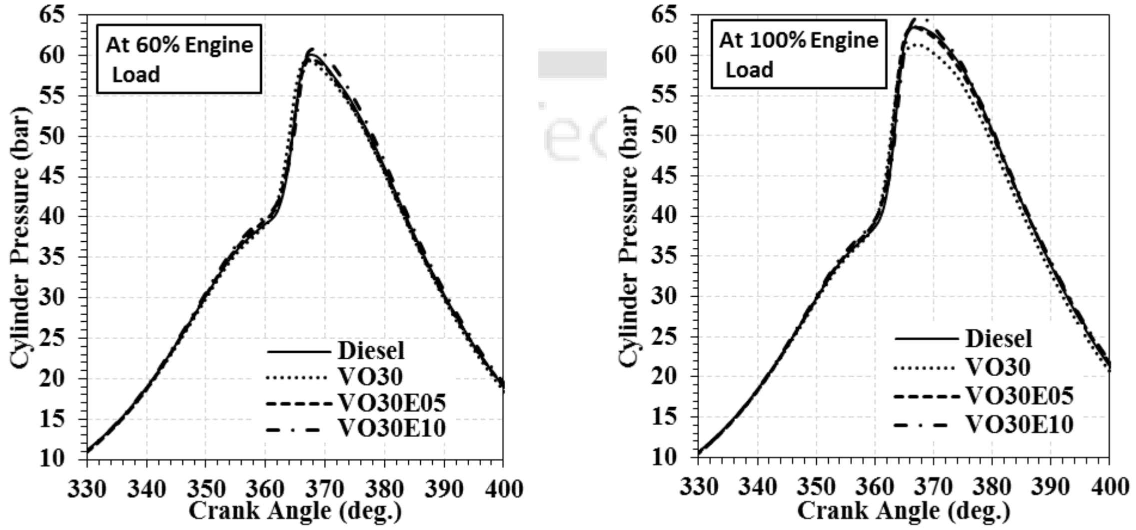


Figure 6.22 Pressure-crank angle diagram of engine at 60 and 100% load for test fuels.

Diesel recorded a highest rate of RoPR in both the part and full load of the engine as depicted in Figures 6.23 and 6.24. At 60% load, the peak RoPR is 5.9, 5.5, 5.2 and 5.2 bar/deg.CA for diesel, VO30, VO30E05 and VO30E10, respectively. While it is 7.3, 6.3, 7.1 and 7.0 for diesel, VO30, VO30E05 and VO30E10, respectively at full load of the engine. However, the start of the RoPR of diesel takes a little longer time than that of the other fuels.

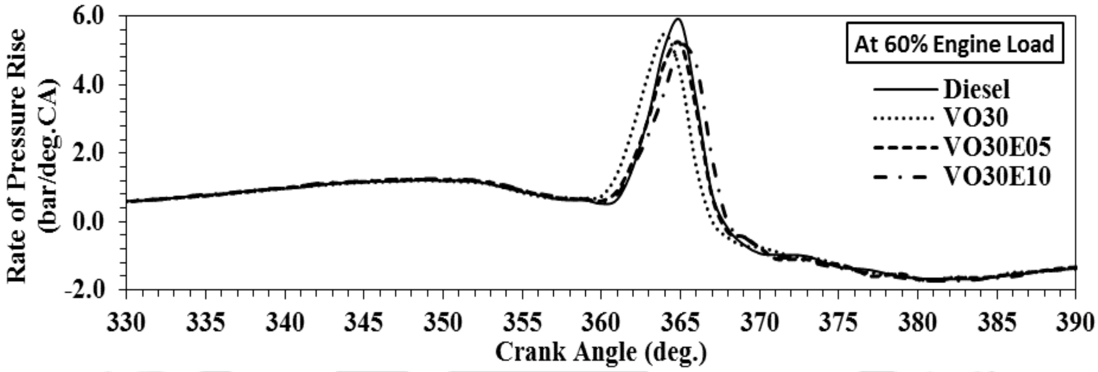


Figure 6.23 Rate of pressure rise diagram of engine at 60% load for different fuels.

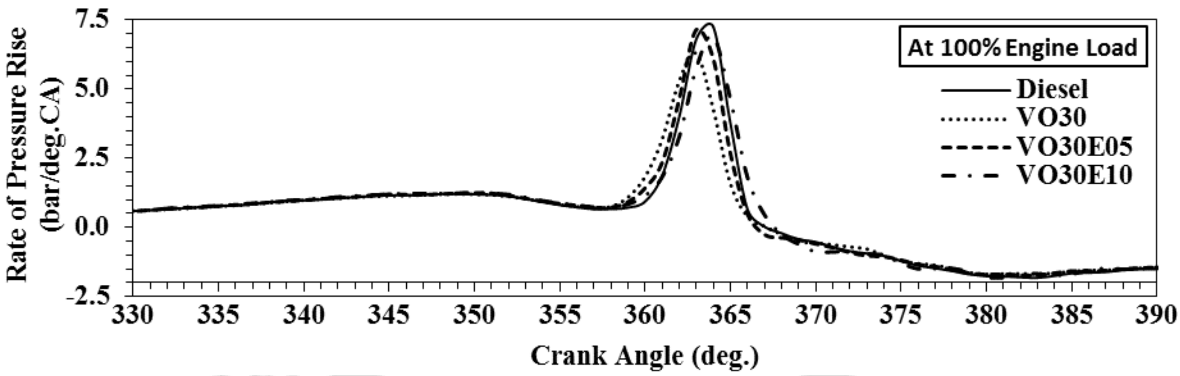


Figure 6.24 Rate of pressure rise diagram of engine at 100% load for different fuels.

The ethanol blended fuels gives shorter ID compared to the diesel and VO30 as shown in Figure 6.25. This is probably due to the better atomization, faster vaporization of fuel droplets and formation of improved mixture of fuel vapor and air. As ID consists of physical and chemical delay, the low viscosity and very high volatility of ethanol reduces the physical delay and results in the shorter ID. The ID decreases with the increase of engine load for the test fuels. Figure 6.26 shows the variations of PCP for tested fuels at the different engine loads. The VO30 recorded a lower PCP in comparison to the other fuels. This is possibly due to the incomplete combustion due to the higher viscosity of fuel. However, with the use of ethanol in the blends PCP increases and it is greater with larger amount of ethanol in the blend. At full load of the engine, the PCP of VO30 increases from 61.4 bar to 63.5 and 64.7 with the addition of 5 and 10% ethanol respectively.

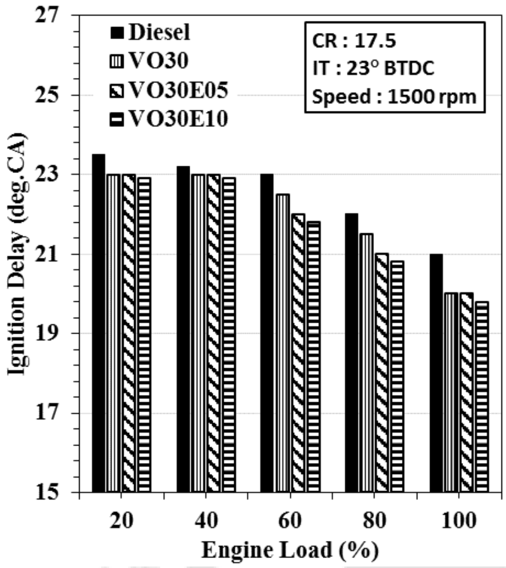


Figure 6.25 Variation of ignition delay with the engine load.

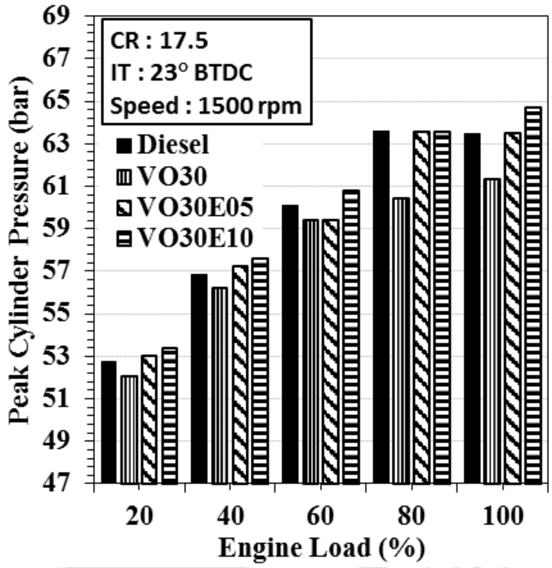


Figure 6.26 Variation of peak cylinder pressure with the engine load.

In Figure 6.27, the net heat release rate (NHHR) for the test fuels at loads of 60 and 100% has been shown. Diesel mode showing of highest NHRR and while VO30 has lower NHRR as compared to the other tested fuels. Further, it indicates an impact of ethanol content in VO30 on the combustion and the consequent heat release rate. The peak NHRR is 75.2, 75.7 and 76.2 J/deg.CA for VO30, VO30E05 and VO30E10 respectively at the engine load of 60%. While for the full load of engine it is 82, 91.6 and 92.9 J/deg.CA for VO30, VO30E05 and VO30E10 respectively. The higher NHRR of ethanol blended VO30 might have resulted in the better BTE of the engine in comparison to neat VO30. Qi *et al*. (2016) and Sathiyamoorthi and Sankaranarayanan (2017) have also reported higher NHRR with ethanol in VO-diesel blend (Qi *et al*. 2016; Sathiyamoorthi and Sankaranarayanan 2017). The variation of combustion duration of test fuels at different loads is presented in Figure 6.28. All the fuels exhibit longer combustion duration with the increase of engine load. The combustion duration increases with the addition of ethanol in the VO30 blend which is longer with higher the amount of ethanol in the blend.

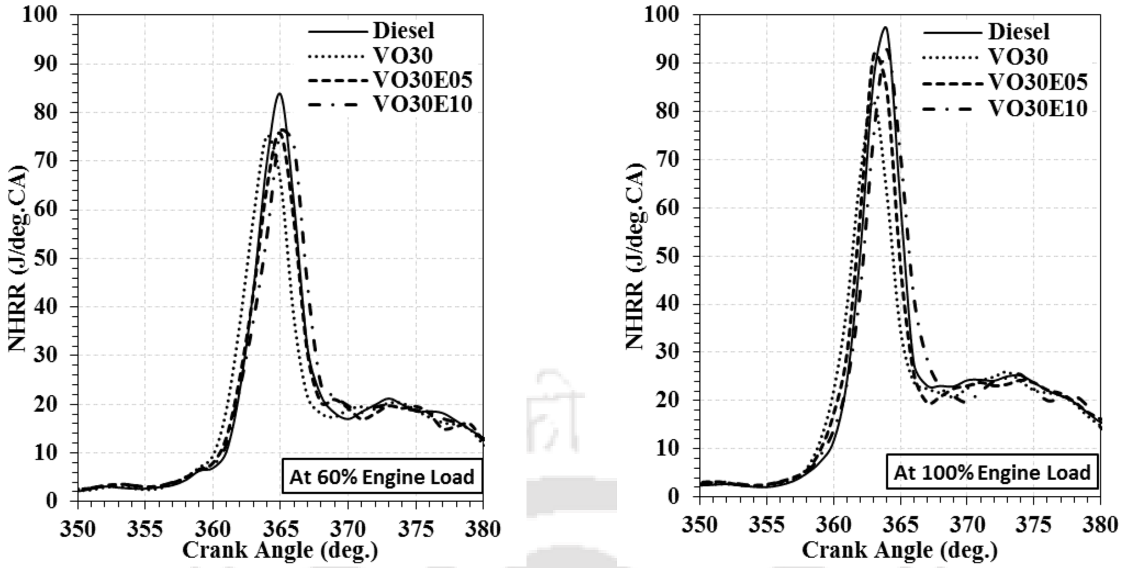


Figure 6.27 NHRR diagram of engine at 60 and 100% engine load for different fuels.

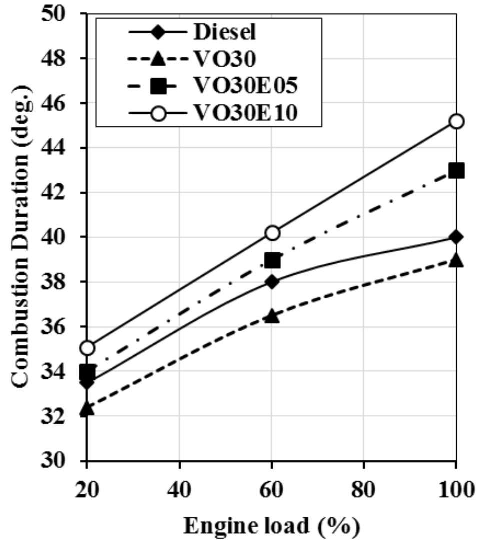


Figure 6.28 Combustion duration of different fuels.

6.3.3 Emissions Analysis

The CO emissions from the engine exhaust at the different load for tested fuels are shown in Figure 6.29. It reveals a higher CO emissions with VO30 as compared to the diesel. This increase is in the range of 28 to 81%. This is probably due to the higher viscosity of VO30. The higher viscosity results in the formation larger fuel droplets leading to incomplete combustion and formation of CO. With the addition of ethanol in VO30, the CO emissions is found to increase further. Although the addition ethanol reduces the viscosity of the fuel remarkably, it also increases the CO emissions (Qi *et al*. 2016; Qi *et al*. 2017; Prakash *et al*. 2018). This has been attributed to the cooling effect of ethanol due to which in-cylinder gas temperature reduces and causing poor oxidation reaction rate (Qi *et al*. 2016). Moreover, VO30 being a viscous fuel, it is bound to produce high CO emissions. The NO emissions of engine increases with the increase of engine load for all the test fuels as indicated in Figure 6.26. With the increasing load, more fuel is being injected and combusted in cylinder leading higher combustion gas temperature and the formation of NOx (Qi *et al*. 2016). In comparison to diesel, the VO30 exhibits a lower NO emissions as indicated in Figure 6.30. At the engine load of 60, 80 and 100%, NO emitted by diesel is 27, 46 and 146 ppm higher than VO30 respectively. This has been attributed to the lower calorific value of VO30 leading to the reduction in peak in-cylinder pressure and temperature (Qi *et al*. 2017)**.** The addition ethanol in VO30 on overall records a marginal reduction in the NO. At full load, the NO emissions is found to be 472, 438 and 421 ppm for VO30, VO30E05 and VO30E10, respectively. The combined effect of high VO content in blend and high latent heat of vaporization of ethanol might have influenced the combustion and this results in the lower NO emissions.

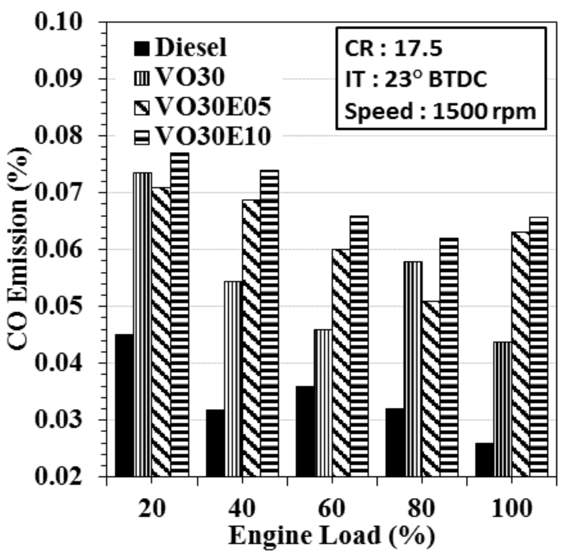


Figure 6.29 Variation of CO emissions with the engine load.

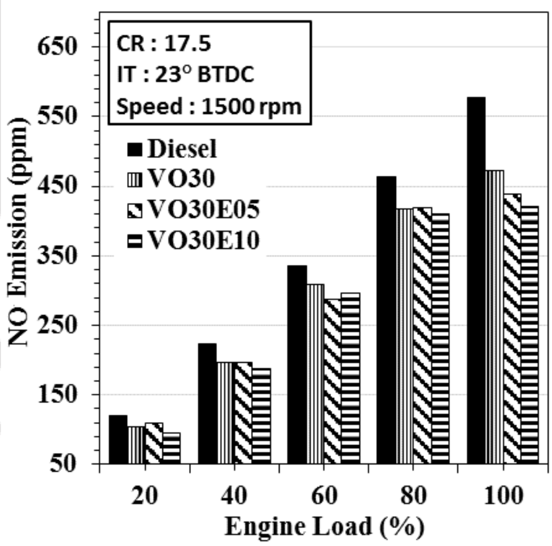


Figure 6.30 Variation of NO emissions with the engine load.

In Figure 6.31, the HC emitted from the engine exhaust for the test fuels has been depicted. It increases with the increase of engine load for all fuels. This is due to increase in fuel-air equivalence ratio with increasing load, as shown in Figure 6.19, which results in relatively lesser availability of oxygen for combustion process (Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010). The VO30 yields higher HC emissions as compared to diesel. It increases from 12 to 30 ppm, 21 to 24 ppm and 27 to 33 ppm at the engine loads of 40, 60 and 100%, respectively. The higher viscosity of VO30 effects the fuel spray quality resulting in an incomplete combustion and the formation HC. The addition of ethanol to VO30 increase the HC emissions and it is higher with greater the fraction of ethanol in the blend. Qi *et al*. (2016) assumes that the cooling effect of ethanol reduces the in-cylinder gas temperature that results in the poorer oxidation reaction rate and a higher emissions of HC. Figure 6.32 represents the variation of CO_2 emissions at various engine loads. It increases with the increase of engine load. As more fuel is injected into combustion chamber, the in-cylinder gas temperature increase with the increasing load, the CO_2 emissions is bound to increase. However, the VO30 shows a comparatively lower CO_2 emissions as compared the other fuels. This is possibly due to the poor combustion of fuel which is evident from the more CO and HC emissions.

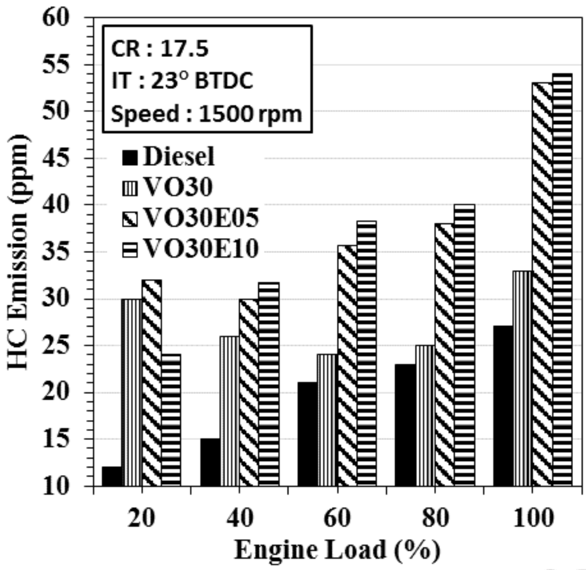


Figure 6.31 Variation of HC emissions with the engine load.

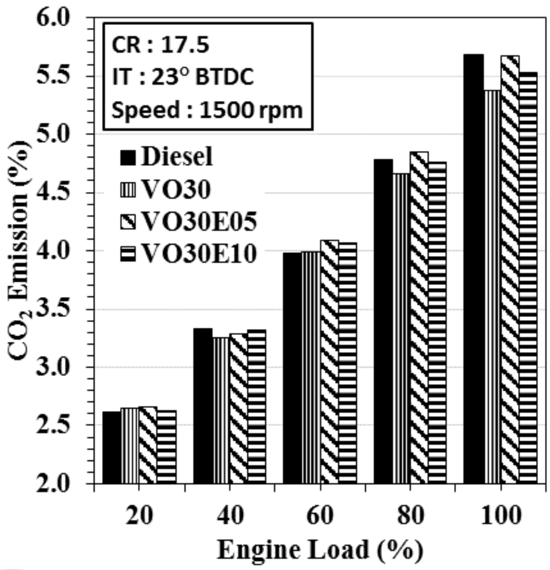


Figure 6.32 Variation of CO_2 emissions with the engine load.

6.3.4 Summary of Module II

In this module of investigation, ethanol is added 5 and 10% to VO30, consisting of 30% *Mesua ferrea* Linn oil and 70% diesel by volume, to study the influence of ethanol on the engine behavior. The findings of the investigation are noted below:

- The BTE of the engine records a marginal increase with 10% ethanol added to VO30. This increase is in the range of 0.5 to 1.5% as compared to neat VO30. However, 5% addition does not show any significant change.
- BSFC of engine increases on an average by 1.8 and 3% with the use of 5 and 10% ethanol respectively due to the lower calorific value of the ethanol.
- The use of ethanol also results in increase of CO and HC emissions. However, it reduces the NO emissions up to 7 and 11% with the addition 5 and 10%, respectively.

6.4 Overall Summary

With the increasing proportion of *Mesua ferrea* Linn oil in the oil-diesel blend, the performance of the engine tends to decrease along with increase of CO and HC emissions as revealed from the testing of VO20 and VO30 in the engine. This is could be due to higher viscosity of blend with increase in the content of *Mesua ferrea* Linn oil which results in injection larger fuel droplets into engine combustion chamber leading to formation of poor combustible mixture and incomplete combustion. To explore the feasibility of redressing this drawback, ethanol is used as an additive to *Mesua ferrea* Linn oil-diesel blend. Ethanol being less viscous, it leads to the reduction in the viscosity of *Mesua ferrea* Linn oil-diesel blend upon blending. In this investigation, ethanol is added 5% and 10% by volume to both VO20 and VO30 blends. The key findings of the VO20 and VO30 with additives are summarized below:

- With 10% ethanol blend to VO20 and VO30 BTE improves by approximately 1 to 4% and 1 to 1.5% respectively. The blending of 5% ethanol blend increases BTE of engine in the range of 0.1 to 3.5% and 0.3 to 1.3% as compared to neat VO20 and VO30 respectively.
- The BSFC of the engine increases with the use of the ethanol in both the V2O and VO30 blends and is found to be higher with higher in amount of ethanol in the blend. On an average, the increment is 1.7 and 2.5% respectively with the use of 5 and 10% ethanol in the VO20 blend. While it is increases, on an average, by 1.8 and 3% with the blending of 5 and 10% ethanol respectively in VO30. This due to the lower calorific value of the ethanol.

- The EGT of the engine records slight increment with use of ethanol in VO20 blend which is in the range of 7 to 12 $^{\circ}$C and 8 to 14 $^{\circ}$C, respectively with the use of 5 and 10% ethanol blend. There is not much significant changes in the EGT with ethanol in VO30 blend.
- The CO emissions of VO20 decreases and NO increases with the use of ethanol. However, HC emissions decreases with 5% ethanol addition and increases with the use of 10% ethanol in the VO20 blend. The use of ethanol in VO30 results in the increase of CO and HC emissions. While, it reduces the NO emissions up to 7 and 11% with the addition 5 and 10% respectively.

Chapter 7

Results of Ternary Blend of VO, Diesel and DEE

OVERVIEW

In Chapter 6, investigation of the CI engine run on Mesua ferrea Linn oil (VO)-diesel binary blend with ethanol as an additive has been reported. This chapter reports the investigation of the diesel engine run on the same blend of VO-diesel blend with diethyl ether (DEE). Thus, the results of the experiments carried out on the ternary blend consists of VO, diesel and DEE are discussed. The chapter begins with the blend composition, the experimental matrix and the properties of test fuels. The experiment matrix consists of two modules having different percentages of DEE. This is followed by the performance, combustion and emissions analysis of the engine. At the end, a summary of important findings on the use of ethanol and DEE as additives to binary blends of VO-diesel has been made.

Chapter Outline:

7.1 Introduction

The use of ethanol in *Mesua ferrea* Linn oil (VO)-diesel blends show an improvement in the performance that has been attributed to the reduction in blend viscosity and contribution of additional oxygen by ethanol. The engine BTE improves while CO emissions decreases with the blending of ethanol with VO20 and VO30. In this part of the study, the focus is to examine the impact of low viscous additive (even lower than ethanol) in VO20 and VO30 blends. Diethyl ether (DEE) having a very low viscosity and high cetane number is considered as an additive for the two blends viz., VO20 and VO30. The addition of low viscosity DEE to VO20 and VO30 would reduce their viscosity. The experimental matrix consists of two modules viz., Modules I and II (Table 7.1.). The modules are considered to observe the effect of DEE on increasing the content of *Mesua ferrea* Linn oil in the blends.

In Module I, DEE is added to VO20 to form two samples of ternary blend VO20DEE05 and VO20DEE10. The VO20DEE05 consists of 5% DEE and 95% VO20 by volume, while VO20DEE10 contains 10% DEE and 90% VO20. While in Module II, DEE is added to VO30 blend to form a two types of ternary blend, first one consists of 5% DEE and 95% VO30 by volume (VO30DEE05) and second one consists of 10% DEE and 90% VO30 by volume (VO30DEE10).

Table 7.1 Experimental matrix.

Module No.	Mode	Fuel used	CR	IT	Speed (rpm)	Loading conditions (%)
I	Diesel	Neat diesel	17.5	23° bTDC	1500	20, 40, 60, 80, 100
	Binary blend	VO20				
	Ternary blend	VO20DEE05				
		VO20DEE10				
II	Diesel	Neat diesel	17.5	23° bTDC	1500	20, 40, 60, 80, 100
	Binary blend	VO30				
	Ternary blend	VO30DEE05				
		VO30DEE10				

The properties of test fuels for Modules I and II are shown in Table 7.2 and Table 7.3 respectively. For both the Modules, the experiments have been conducted to investigate the performance, combustion and emissions behavior of the CI engine when DEE is added to the test fuels. The experimental results are discussed in the following subsections.

Table 7.2 Fuel properties for Module I.

Properties	Units	Diesel	DEE	VO20	VO20DEE05	VO20DEEE10
Calorific value of fuel	kJ/kg	42000*	33900#	40722	40381	40040
Density at 15 °C	kg/m^3	828	713#	860.4	852.7	845.7
Viscosity at 40 °C	mm^2/s	2.036	0.23#	3.700	-	-
Stoichiometric air-fuel ratio	-	15	11.2	14.4	14.2	14.0

*Sahoo et al (2011); #Rakopoulos (2013)

Table 7.3 Fuel properties for Module II.

Properties	Units	Diesel	DEE	VO30	VO30DEE05	VO30DEE10
Calorific value of fuel	kJ/kg	42000*	33900#	40083	39774	39465
Density at 15 °C	kg/m^3	828	713#	869.2	861.4	853.6
Viscosity at 40 °C	mm^2/s	2.036	0.23#	4.996	-	-
Stoichiometric air-fuel ratio	-	15	11.2	14.1	13.9	13.6

7.2 Module I with VO20

7.2.1 Performance Analysis

The BSFC of test fuels at the different engine load is shown in Figure 7.1. The fuels recorded a lower BSFC with the increasing engine load. Diesel mode shows a lower BSFC as compared to VO20. Diesel has higher caloric value in comparison to VO20 that results in lower BSFC of diesel. With the addition of DEE in VO20 blend, the BSFC of engine increases and this increment is greater with larger the amount of DEE added especially at higher loads. It increases by 1.1, 0.6 and 1.7% with the use of 5% DEE to the VO20 blend (VO20DEE05) at the loads of 60, 80 and 100%, respectively. On using 10% DEE to the VO30 blend (VO20DEE10), the BSFC increases by 2.1, 1.9 and 3.1% at the loads of 60, 80 and 100%, respectively. As DEE has a lower calorific than VO20, the increase in the amount of DEE in VO20 blend tends to reduce the overall calorific value of fuel and this results in a higher BSFC to produce a unit power as that of VO20. The BTE of the engine increases with the increasing engine load as revealed in Figure 7.2. The BTE of engine reduces with the use of VO20 as compared to diesel mode. This reduction in BTE is in the range of 2.4 to 4.7% as compared to diesel. With the blending of *Mesua ferrea* Linn oil with diesel, the viscosity of fuel tends to increase and possibly this results in the formation larger fuel droplets. This reduces the surface area of contact between fuel and air which results in the poor combustion and reduction in BTE.

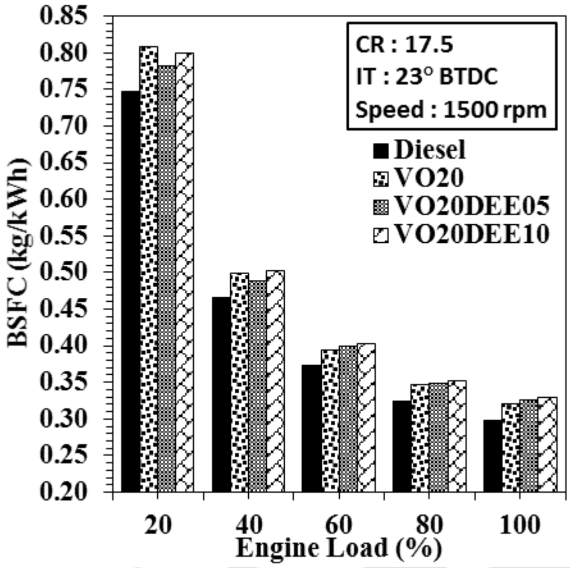


Figure 7.1 BSFC of fuels at different engine load.

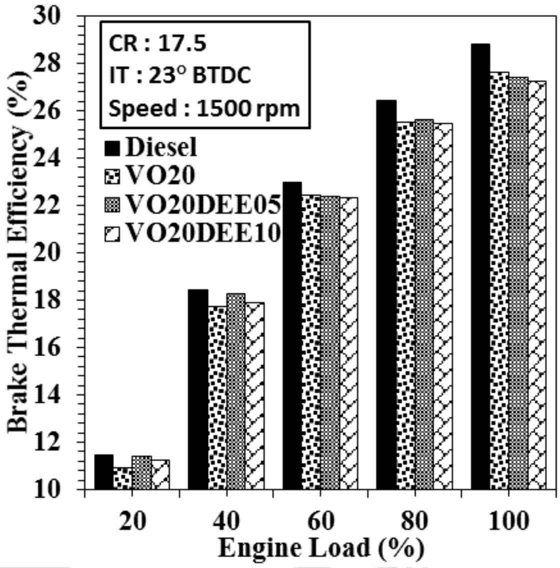


Figure 7.2 BTE of fuels at different engine load.

Moreover, it is evident from Figure 7.3 that the fuel-air mixture is richer with the use of VO20 in comparison to diesel and this results in the poorer combustion of fuel. The equivalence ratio of engine increases by 1.6 to 4.6% with the use of VO20. The use of DEE in VO20 does not show much variation in the BTE at the loads of 60, 80 and 100%. However, at the lower loads of 20 and 40%, there is a little improvement in BTE. At 20% load, the BTE of VO20 increases from 10.9 to 11.4 and 11.2% with the use of 5 and 10% DEE, respectively. While it increases from 17.7 to 18.3 and 17.9% with VO20DEE05 and VO20DEE10, respectively. At lower loads, the addition of DEE leads to the formation of leaner fuel-air mixture as compared to neat VO20 as revealed in Figure 7.3. This results a better combustion and improves the BTE at lower engine loads. However, with the increasing load, it is likely that DEE vaporizes out from the blend due to its high volatility and then autoignites before the actual combustion. The EGT of the engine increases with the blending of VO to the diesel as illustrated in Figure 7.4. This increase is in the range of 3.2 to 4.5%. This has been attributed to the high viscosity and low volatility of VO which affects the atomization and spray formation results in slow combustion which continues to burn at the later phase of the combustion and increase the EGT (Pramanik 2003; Hebbal *et al*. 2006;

Devan and Mahalakshmi 2009; Chauhan *et al*. 2010). An increase in the EGT is observed with the addition of DEE in VO20 blend. This increase is up to 3.4 and 4.4% with the use of 5 and 10% DEE, respectively. The use of DEE as additives to VO20 blend does not seem to have any remarkable impact on the volumetric efficiency of the engine as revealed in Figure 7.5. At 100% load, it is 85, 85, 84.1 and 84.6% respectively for diesel, VO20, VO20DEE05 and VO20DEE10, respectively.

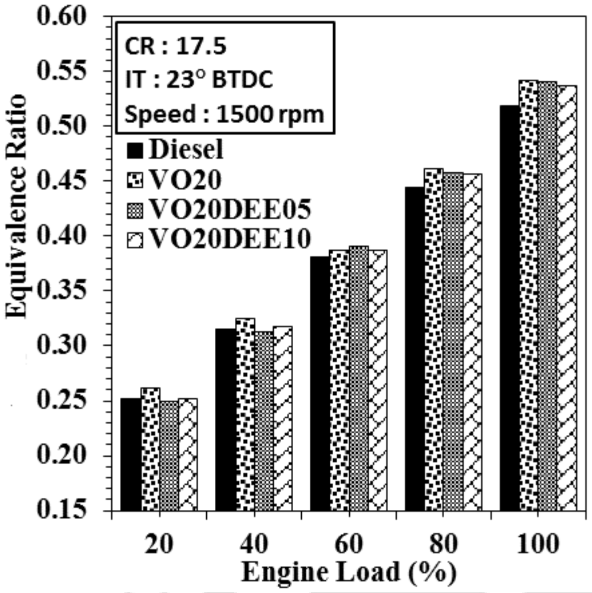


Figure 7.3 Equivalence ratio at different engine load.

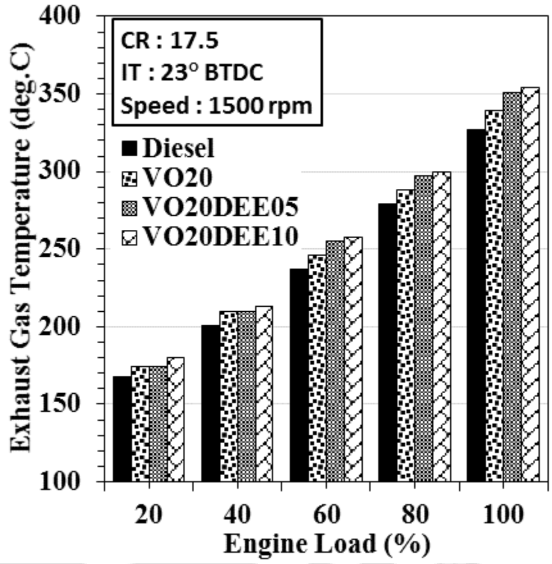


Figure 7.4 EGT of fuels at different engine load.

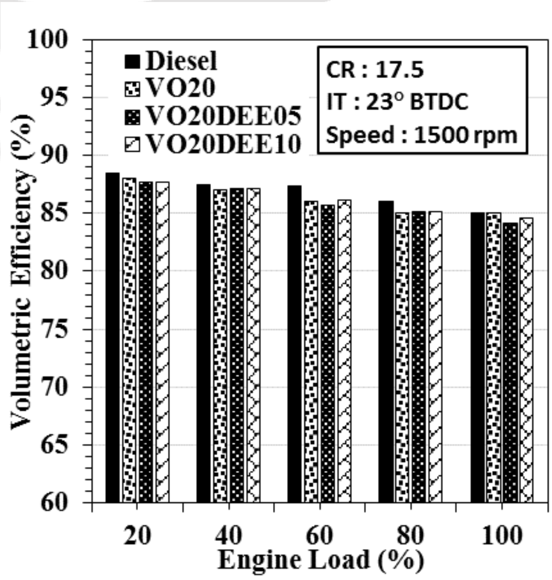


Figure 7.5 Variation volumetric efficiency with engine load.

7.2.2 Combustion Analysis

The variation in the in-cylinder pressure at the different crank angle for tested fuels at the engine loads of 60 and 100% is shown in Figure 7.6. All the test fuels show almost similar variation of pressure profile with little a difference in their magnitude. The blended fuels indicate a little advance in pressure profile as compared to diesel. This is due to the earlier rise in the RoPR of the blended fuels as compared to diesel as shown in Figures 7.7 and 7.8. However, blended fuels recorded lower peak RoPR. This results in the higher PCP with diesel as compared to blended fuels. At 60% load, the peak RoPR is found to be 5.9, 5.4, 5.6 and 5.7 bar/deg.CA for diesel, VO20, VO20DEE05 and VO20DEE10, respectively. While it is 7.3, 6.7, 6.8 and 7 bar/deg.CA respectively for full load.

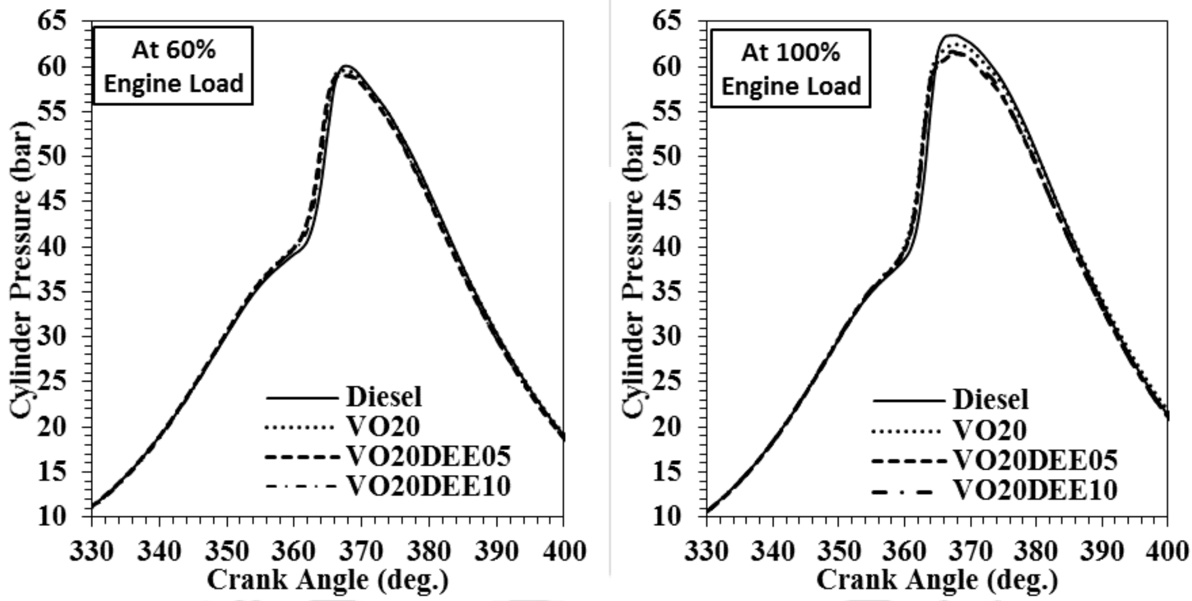


Figure 7.6 Pressure-crank angle diagram of engine at 60 and 100% load for different fuels.

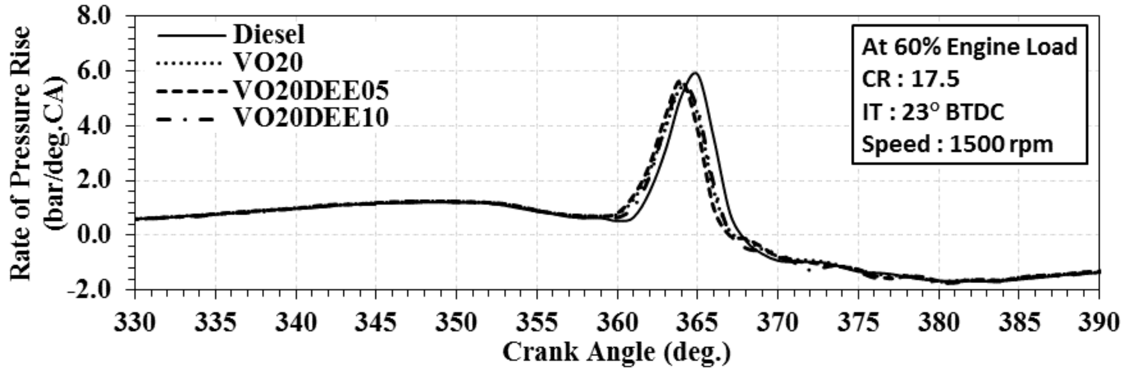


Figure 7.7 Rate of pressure rise diagram of engine at 60% load of engine for different fuels.

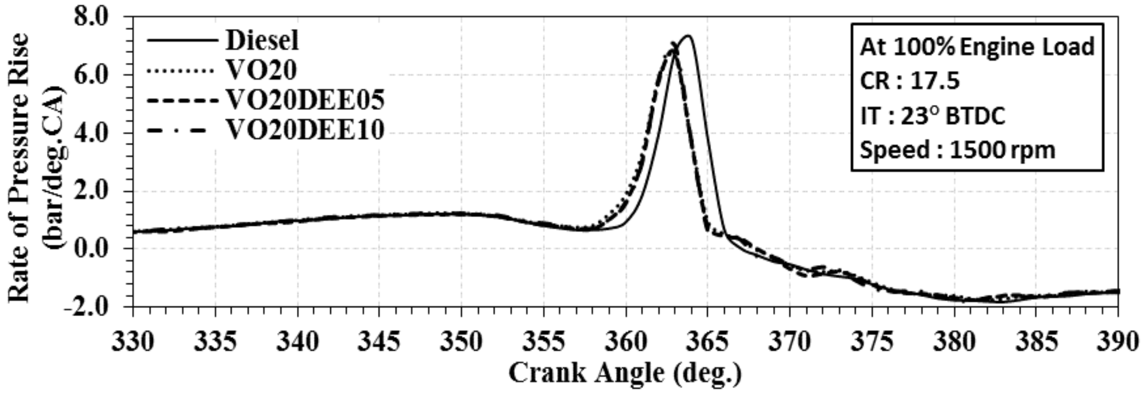


Figure 7.8 Rate of pressure rise diagram at 100% load of engine for different fuels.

There is a little drop in the PCP with the blending of 5% DEE in VO20 which improves upon blending with 10%. At 100% load, the PCP is found to be 63.5, 62.5, 61.4 and 61.7 bar for diesel, VO20, VO20DEE05 and VO20DEE10, respectively. The PCP of the test fuels at other loads are shown in Figure 7.9. The IDs of test fuels at the loads of 20, 40, 60, 80 and 100% is shown in Figure 7.10. It decreases with the increase of engine load. With the increasing load, more fuel is combusted resulting in an increase of temperature in the combustion chamber and cylinder walls. The blended fuels VO20, VO20DEE05 and VO20DEE10 show almost same degree of IDs. The blended fuels have recorded shorter IDs as compared to diesel. Since DEE has a high cetane value, the addition of DEE might have resulted a shorter ID.

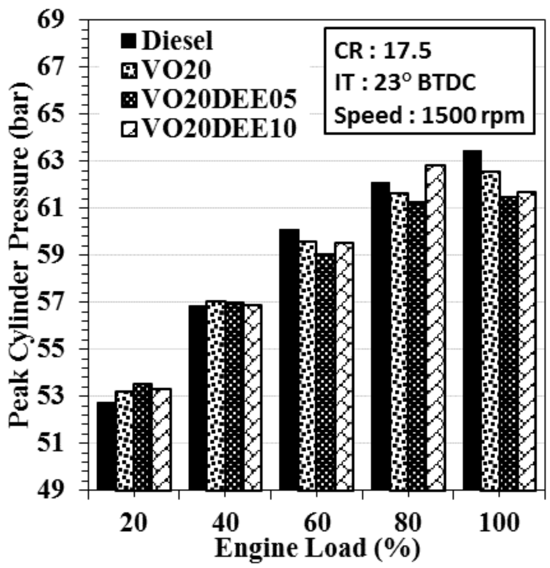


Figure 7.9 Variation of peak cylinder pressure with the engine load.

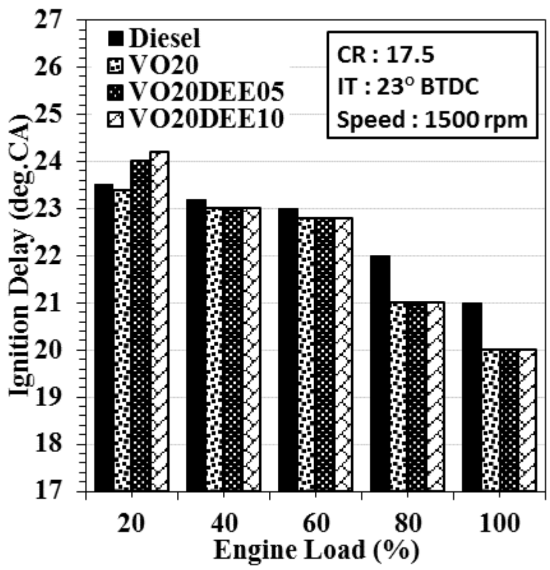


Figure 7.10 Variation of ignition delay with the engine load.

This shorter ID results in the earlier start of combustion of fuel and advancement in the rate of pressure with the blended fuels as shown in Figure 7.7 and 7.8. The RoPR in the engine cylinder depends on the amount of heat released during the combustion of fuel. The heat release rate results in the increase of temperature and pressure. The NHRR of the engine at 60 and 100% loads is shown in Figure 7.11. The blended fuels show a lower NHRR as compared to diesel. At 60% load, the peak NHRR is 83.8, 74.4, 77.1 and 76.2 J/deg.CA for diesel, VO20, VO20DEE05 and VO20DEE10, respectively. While it is found to be 97, 87, 87.6 and 90.5 J/deg.CA respectively at 100% load. This is attributed to the lower heating value of blended fuels as compared to diesel. With the increasing load, the combustion duration increases for all the test fuels as shown in Figure 7.12. The DEE blended VO20 indicates a longer combustion as compared to neat VO20.

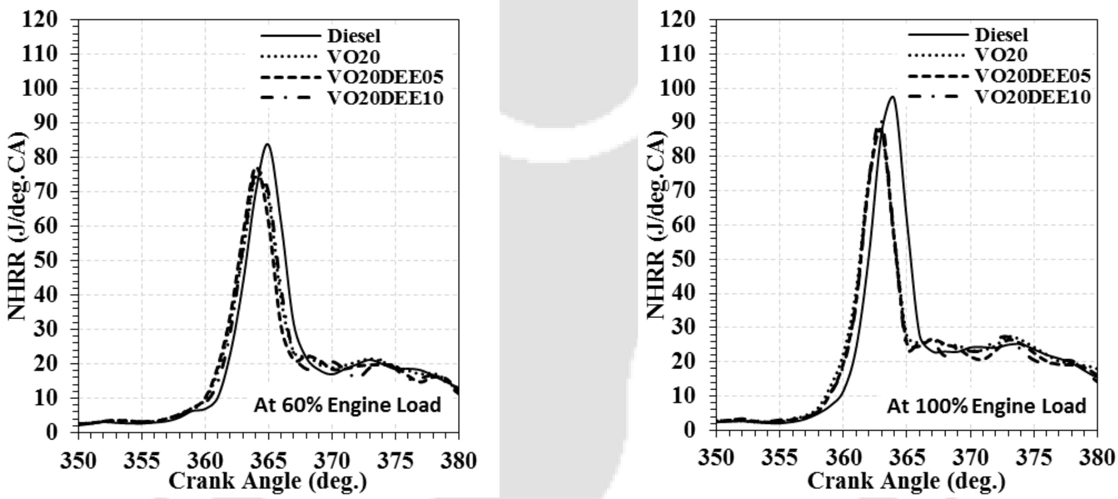


Figure 7.11 NHRR diagram of engine at 60 and 100% engine load for different fuels.

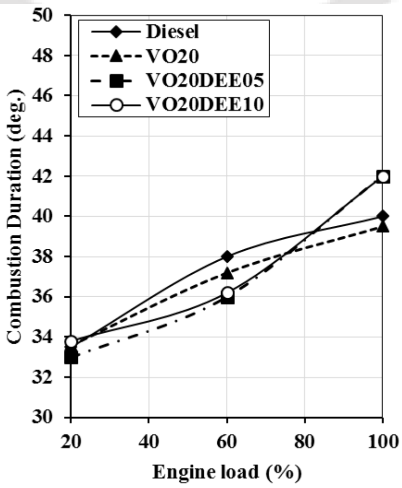


Figure 7.12 Combustion duration of engine for different fuels.

7.2.3 Emissions Analysis

The CO emitted by engine at the various load for the tested fuels have been illustrated in Figure 7.13. Throughout the operating range, the use of VO20 results in the release of higher CO in the exhaust as compared to diesel. This increase is in CO emissions with VO20 is in the range of 1.5 to 33.3%. At the 100% load, the CO emitted by diesel and VO20 are 0.026 and 0.03%, respectively. This increase in CO is possibly due to the poor combustion with the use of VO20. The details have already been discussed in the performance analysis. The use of DEE in VO20 reduces the emissions of CO. It reduces by 2.2 to 8.3% with the use of VO20DEE05 as compared to VO20. The reduction in CO emissions of VO20DEE10 is in the range of 4.3 to 13.6% as compared to VO20. At the 100% load, the CO emitted by VO20, VO20DEE05 and VO20DEE10 are found to be 0.034, 0.032 and 0.03%, respectively. This reduction in CO emissions could be due to the decreasing viscosity of VO20 with the addition of DEE. The lower viscosity of fuel ensures a better atomization of fuel thereby improving the combustion. Sivalakshmi and Balusamy (2013) assumes that the better atomization of fuel and fuel-air mixing reduces rich region in cylinder and reduces the CO emission. Moreover, the DEE blends show a leaner fuel-air ratio as indicated in Figure 7.3.

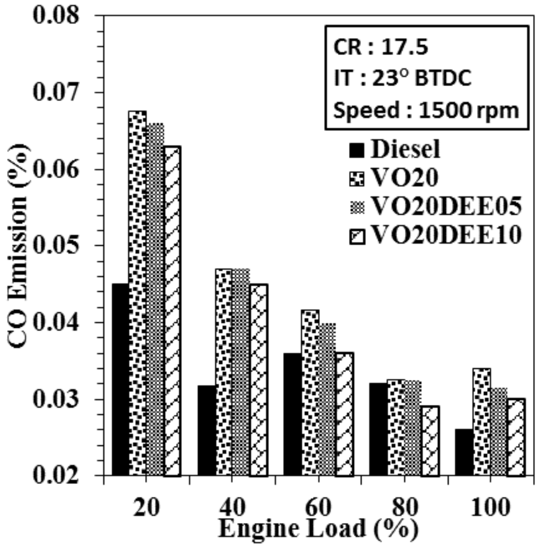


Figure 7.13 Variation of CO emissions with the engine load.

The NO emissions increases with the increasing load for all the test fuels as revealed in Figure 7.14. With the increasing load, a larger quantity of fuel is injected and combusted in the cylinder producing a high combustion temperature and forming higher NOx in the exhaust. The rate of NO_X formation largely depends on combustion temperature, the residence time of nitrogen at that temperature, and the contents of oxygen in the combustion chamber (Huang *et al*. 2009). The VO20 shows 2.1 to 10.2% lower NO emissions as compared to diesel. At full load, the NO emitted by diesel and VO20 are found to be 578 and 519 ppm, respectively. This has been attributed to higher viscosity of VO20. Higher viscosity results in larger fuel droplets and longer combustion duration with substantial energy release during the late burning phase. Because of this, the peak combustion chamber temperature becomes possibly lower due to lower heat release in the pre-mixed combustion and mixing controlled combustion phase. Thus reduces the formation of NO (Agarwal and Rajamanoharan 2009). This energy release during late burning phase may also be responsible for higher EGT with VO20 as shown in Figure 7.4. The addition of DEE in VO20 further reduces the emissions of NO. At the intermediate load of 60%, the NO emissions of VO20 reduces from 336 ppm to 277 and 263 ppm with the use of VO20DEE05 and VO20DEE10, respectively. While at full load, it is 519, 419 and 395 ppm with VO20, VO20DEE05 and VO20DEE10, respectively. Rakopoulos (2013) assumes that the lower in the NOx emissions with DEE is due to the reduction in premixed phase of combustion and associated lower temperatures of the engine cylinder. The formation of NOx mainly takes at the premixed phase of the combustion. It is evident from Figure 7.10, especially at 100% load, the NHRR profile of VO20 envelops over the profiles of VO20DEE05 and VO20DEE10.

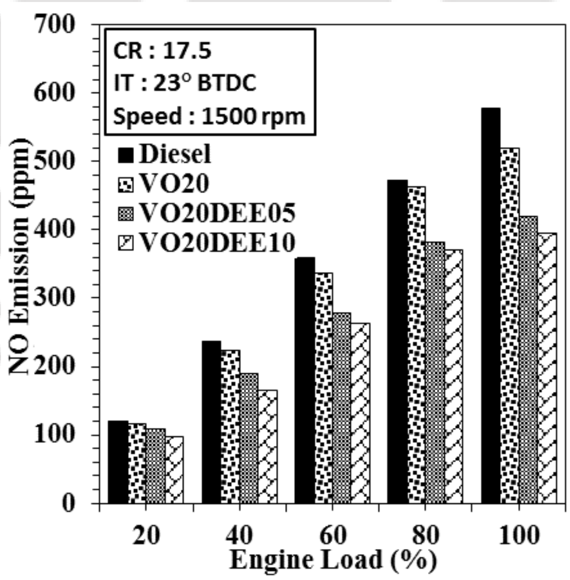


Figure 7.14 Variation of NO emissions with the engine load.

The variation in the emissions of unburned HC at various loads for the test fuels is shown in Figure 7.15. It increases with the increasing load for all the test fuels. The VO20 blend shows a higher HC emissions as compared diesel. It is 12, 18 and 27 ppm for diesel at the loads of

20, 60 and 100%, respectively. On the other hand, the VO20 shows 17, 21 and 30 ppm at the respective loads. The higher viscosity of VO20 might have resulted a poor combustion leading to a larger amount of HC emissions. The blending of DEE with VO20 further accelerates the formation of HC. At 100% load, it is 30, 40 and 49 ppm for VO20, VO20DEE05 and VO20DEE10, respectively. This has been attributed to the high latent heat of vaporization of DEE which results in lowering the combustion temperature especially around the cylinder walls at time of mixture formation. This leads to the formation of higher HC (Sivalakshmi and Balusamy 2013; Venu and Madhavan 2017). As more amount of fuel is burned with the increasing load, the CO_2 emissions also increases as depicted in Figure 7.16. Diesel shows a higher CO_2 emissions as compared to VO20 indicating a better combustion of the former as compared to the later. With the addition of DEE in VO20, the CO_2 emission increases. This could be due to the better combustion with DEE.

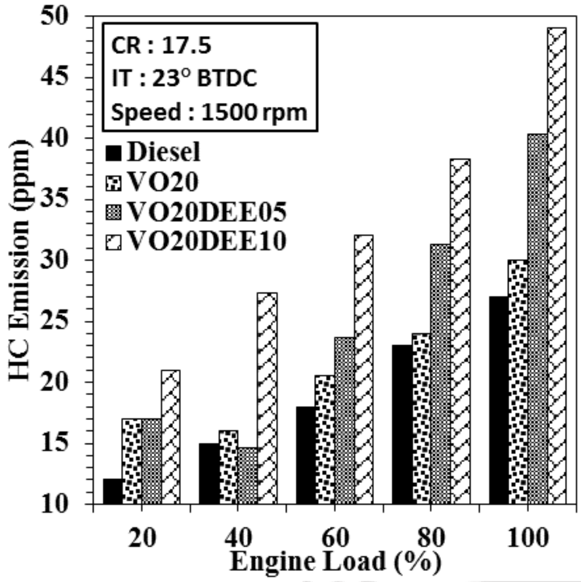


Figure 7.15 Variation of HC emissions with the engine load.

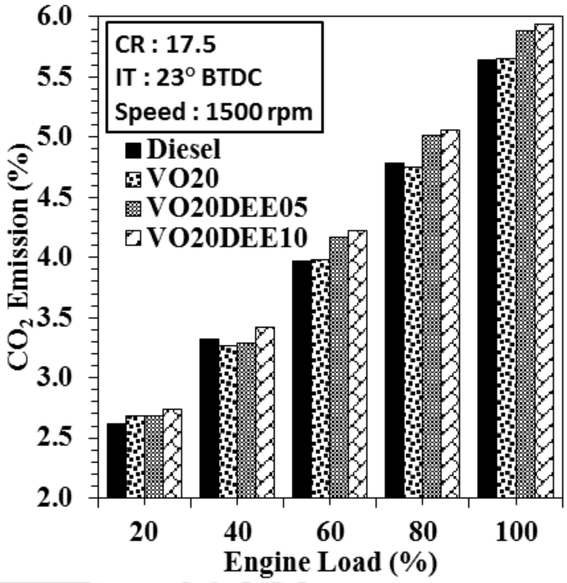


Figure 7.16 Variation of CO_2 emissions with the engine load.

7.2.4 Summary of Module I

- With the addition of DEE in VO20 blend, the BSFC of engine increases which is greater with larger the amount of DEE added especially at higher loads. It increases by 1.1, 0.6 and 1.7% with the use of 5% DEE at the loads of 60, 80 and 100%, respectively. While it increases by 2.1, 1.9 and 3.1% respectively with 10% DEE to VO20.

- The use of DEE in VO20 does not show much variation in the BTE. However, at the lower loads of 20 and 40%, there is a little improvement in BTE. At 20% load, it increases from 10.9% to 11.4 and 11.2% with the use of 5 and 10% DEE respectively. While it increases from 17.7% to 18.3 and 17.9% with VO20DEE05 and VO20DEE10, respectively.
- An increase in the EGT is observed with the addition of DEE in VO20. This increase is up to 3.4 and 4.4% with the use of 5 and 10% DEE respectively.
- The use of DEE in VO20 reduces the emissions of CO. It reduces by 2.2 to 8.3% with the use of VO20DEE05 as compared to VO20. The reduction in CO emissions of VO20DEE10 is in the range of 4.3 to 13.6% as compared to VO20.
- The addition of DEE in VO20 reduces the emissions of NO. At the intermediate load of 60%, the NO emissions of VO20 reduces from 336 ppm to 277 and 263 ppm with the use of 5 and 10% DEE respectively. While at full load, it is 519, 419 and 395 ppm with VO20, VO20DEE05 and VO20DEE10, respectively.
- The blending of DEE with VO20 further accelerates the formation of HC. At 100% load, it is 30, 40 and 49 ppm for VO20, VO20DEE05 and VO20DEE10, respectively. With the addition of DEE in VO20, the CO_2 emissions increases. This could be due to the better combustion of the blend in presence of DEE.

7.3 Module II with VO30

7.3.1 Performance Analysis

The variation in the BSFC of test fuels at the various engine loads is shown in Figure 7.17. It reveals a decreasing trend in the BSFC with the increase of engine load for all the test fuels. Diesel mode records a lowest BSFC among all the test fuels due to the higher calorific value of diesel. Overall, the addition of DEE on VO30 shows a little increase in the BSFC especially at loads of 40, 60, and 100% load. Conversely, at the lower loads of 20 and 40%, the BSFC is higher with the use of DEE. This is possibly due to the lower heating value of DEE. The heating value of VO30 decreases with the addition of diethyl ether in the blend (Table 7.3). This decrease in calorific value results in higher BSFC with the use of 5 and 10% DEE in the VO30. Irrespective of the fuel used, an increase of BTE with increasing load is observed as depicted in Figure 7.18. In comparison to diesel, the VO30 shows a lower BTE throughout the operating loads of the engine. It decreases by 2.1%, 4.3 and 4.6% at the loads of 20, 60 and 100%, respectively. This could be due to the higher viscosity VO30 as compared to diesel which leads to the injection of larger fuel droplets into combustion chamber. Larger fuel droplets result in poor combustion and reduction in BTE.

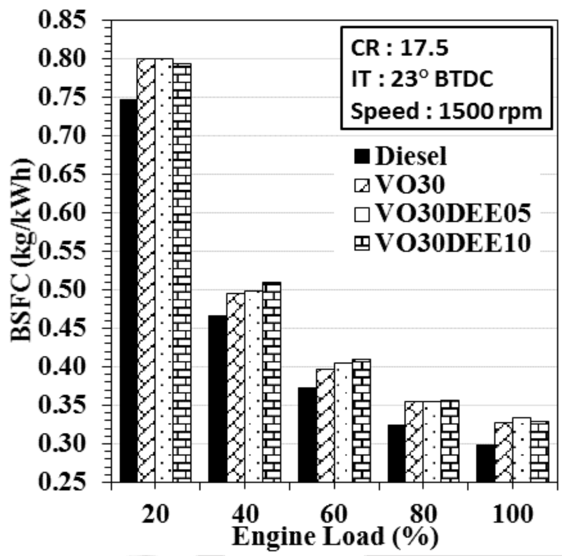


Figure 7.17 BSFC of fuels at different engine load.

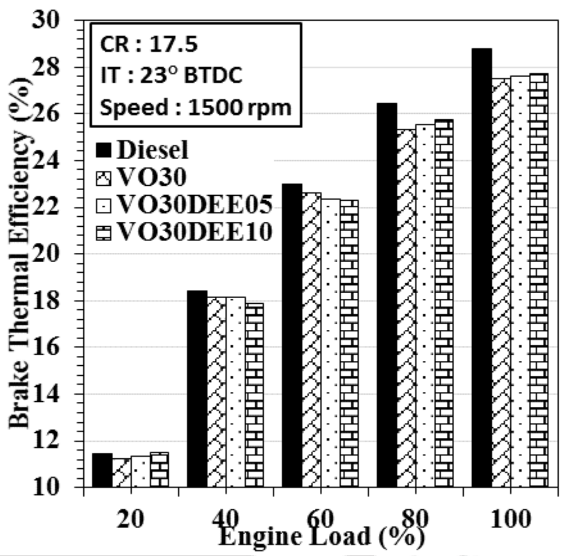


Figure 7.18 BTE of fuels at different engine load.

Moreover, the VO30 being a high density fuel with higher BSFC, the fuel-air mixture becomes richer as revealed in Figure 7.19. This richer mixture results in an incomplete combustion. The equivalence ratio of VO30 is higher than diesel by 1.1 to 4.1%. The addition of DEE to VO30 increases the BTE of the engine as compared to neat VO30. This increase, on an average, is 0.6% and 1.6% with the addition of 5 and 10% DEE respectively. The equivalence ratio of VO30 decreases with the blending of DEE and this results in a smooth combustion thereby improving the BTE.

Figure 7.20 illustrates the variation of EGT of test fuels at various engine loads. It shows an increment trend with the increasing loads as more fuels are being combusted. The VO30 blend shows a little higher EGT as compared to diesel. This increment in the EGT is in the range of 1 to 3.6%. At full load, diesel shows an EGT of 327 °C, while VO30 records an EGT of 333 °C. It is believed that the high viscosity and low volatility of VO affects the atomization and spray formation which results in slow combustion. This slow combustion leads to the continuation of the burning of fuel at the later stages of combustion, and consequently the EGT increases (Pramanik 2003; Hebbal *et al*. 2006; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010). There is not much significant variation in the EGT with the use of DEE. The volumetric efficiency of engine at various load for the test fuels are shown in Figure 7.21. It decreases with the increasing load for all the tested fuels. This could be due to the higher pressure of residual gas at the end of exhaust stroke with increasing load. This high pressure residual gas needs an additional volume of suction stroke for expansion and admit the fresh air. The DEE blend VO30 has a volumetric efficiency slightly higher than that of the neat VO30.

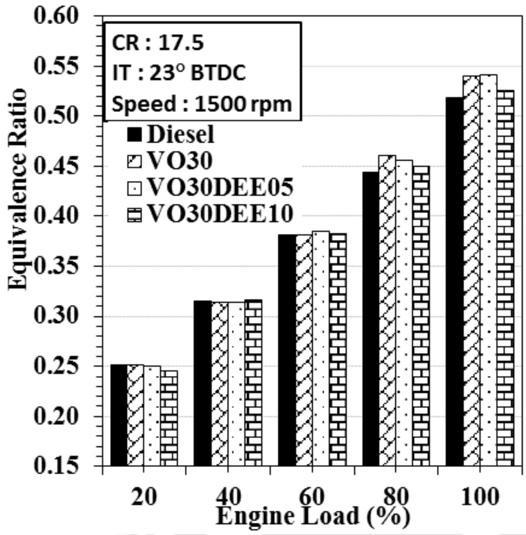


Figure 7.19 Equivalence ratio at different engine load.

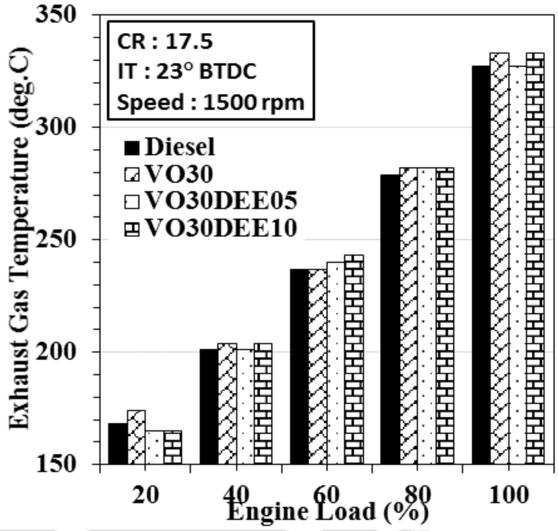


Figure 7.20 EGT of fuels at different engine load.

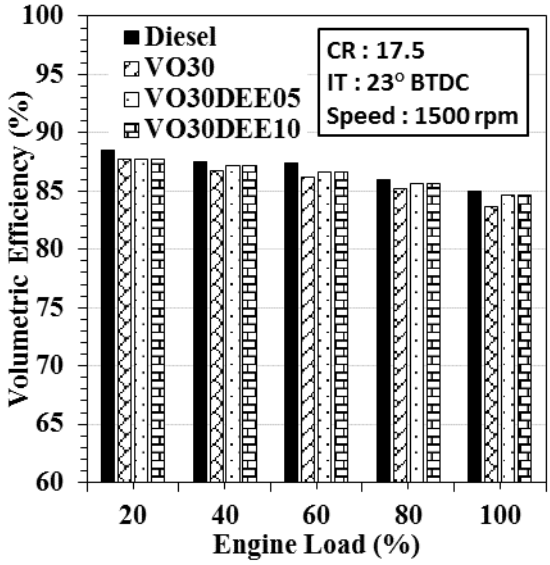


Figure 7.21 Volumetric efficiency of engine at different engine load.

7.3.2 Combustion Analysis

The in-cylinder pressure-crank angle diagram of engine at the 60% (part load) and 100% (full load) of engine load is illustrated in Figure 7.22. All the test fuels exhibit a similar pressure profile trend across the given range of engine crank angle. The VO30 maintains a lower cylinder pressure profile in comparison to the other fuels especially at the full load. This could be due to the higher viscosity of VO30 which results in the formation larger fuel droplets affecting the proper combustion of the fuel. The PCP of the test fuels at various loads is shown in Figure 7.23. In overall, the PCP improves with the addition DEE to the VO30. At 100% load, the PCP is 63.5, 61.4, 63.1 and 62.6 bar, respectively for diesel, VO30, VO30DEE05 and VO30DEE10, respectively.

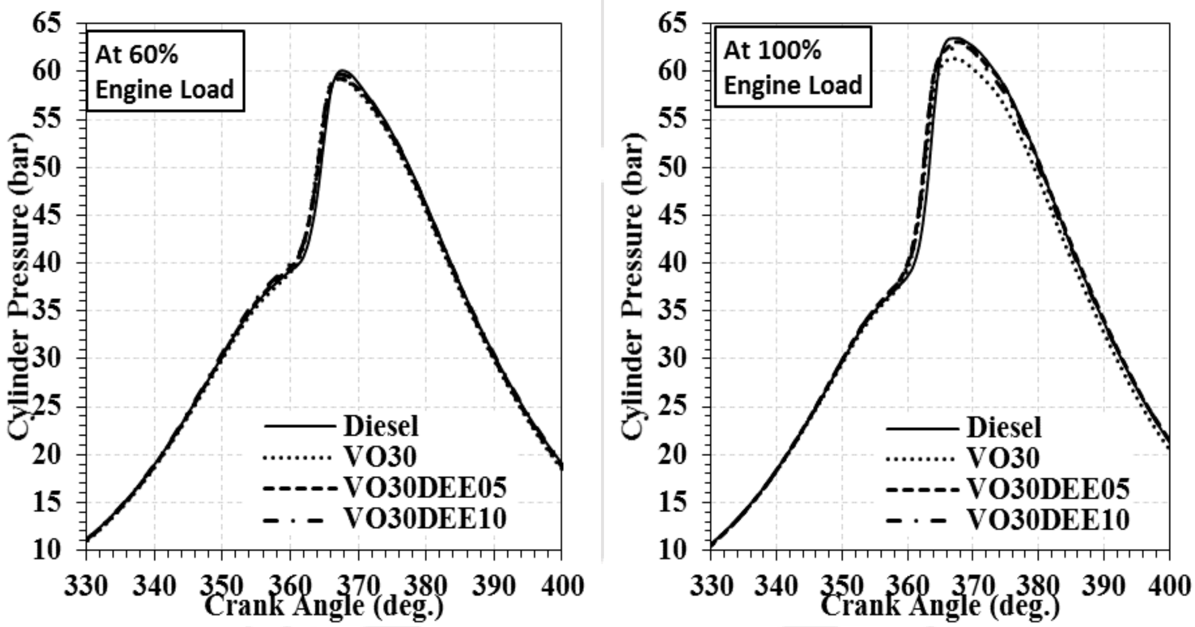


Figure 7.22 Pressure-crank angle diagram of engine at 60 and 100% load for different fuels

Diesel recorded a highest RoPR in both part and full loads of the engine as depicted in Figures 7.24 and 7.25. At 60% load, the peak RoPR is 5.9, 5.5, 5.1 and 5.6 bar/deg.CA for diesel, VO30, VO30DEE05 and VO30E10, respectively. While it is 7.3, 6.3, 6.9 and 7.0 bar/deg.CA respectively at 100% load. The fuel upon injected into the combustion chamber passes through a phase of ID followed by combustion with the subsequent heat release that results in the rise of pressure and temperature.

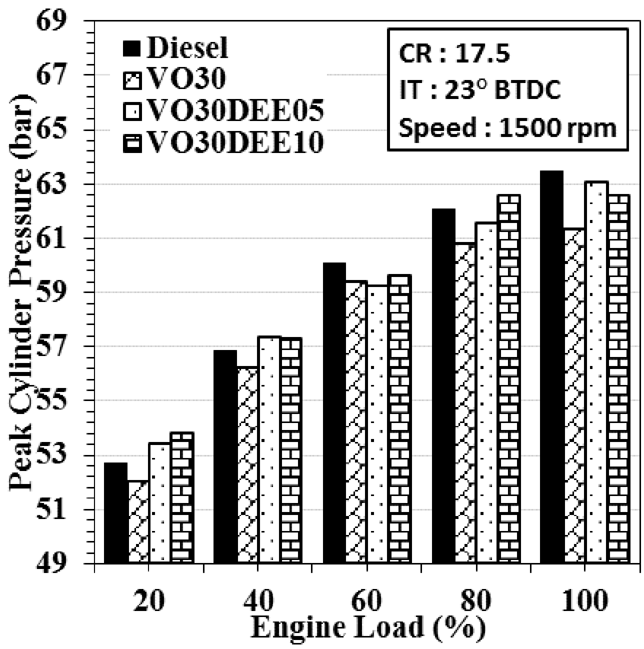


Figure 7.23 Variation of peak cylinder pressure with the engine load.

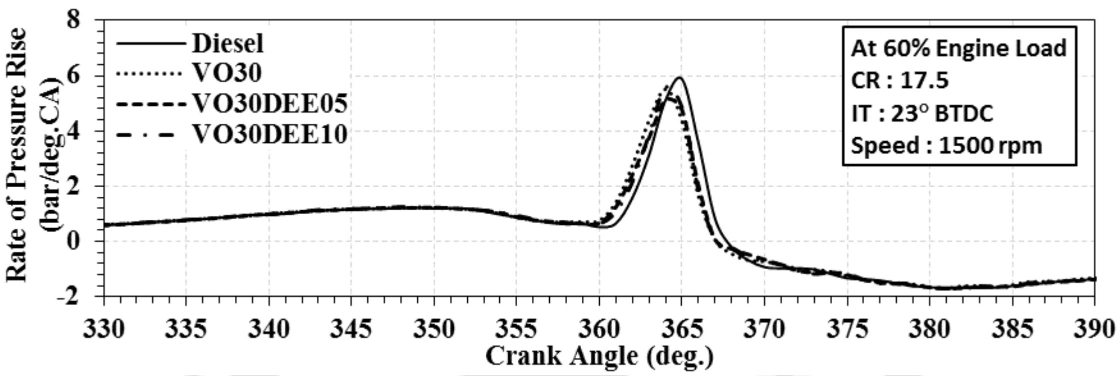


Figure 7.24 Rate of pressure rise diagram of engine at 60% load of engine for different fuels.

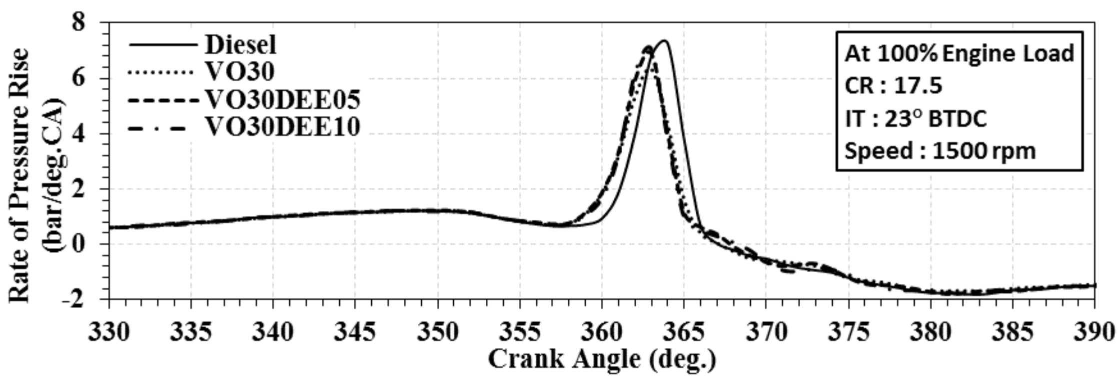


Figure 7.25 Rate of pressure rise diagram of engine at 100% load of engine for different fuels.

The variation in the ignition delay (ID) of the test fuels at the different engine loads is shown in Figure 7.26. The fuels exhibit a decrease in the ID with the increasing engine load. At 20% load, the IDs are found to be 23.5, 23.6, 23 and 23 deg.CA for diesel, VO30, VO30DEE05 and VO30DEE10 respectively. However, at 100% load, the ID decreases to 21, 20, 20 and 20 deg.CA respectively. The combustion chamber temperature increases with increasing load due to burning of larger amount of fuel which results in shorter ID. The blending of DEE to VO30 results in the shorter ID. This could be attributed to the high cetane number of DEE.

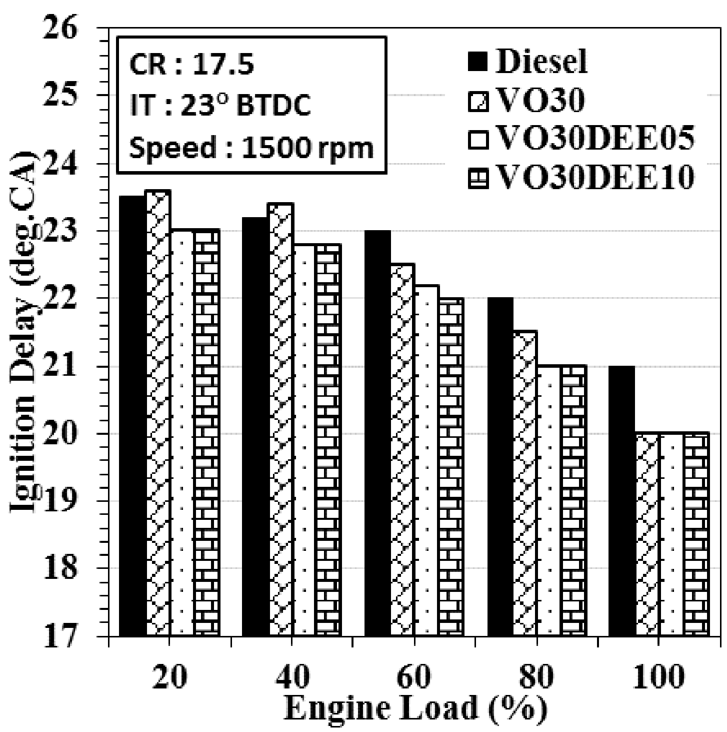


Figure 7.26 Variation of ignition delay with the engine load.

The NHRR of test fuels at the engine loads of 60 and 100% is depicted in Figure 7.27. All the fuels exhibit a similar trend of NHRR across the engine cycle. Figure 7.27 also indicates an earlier combustion of blended fuels as compared to neat diesel as their NHRR profiles outgrow diesel in the early phase of combustion. This results in the earlier peaking up of the NHRR of blended fuels. At 100% load, the peak NHRR is 96.9, 82, 89.2 and 90.8 J/deg.CA for diesel, VO30, VO30DEE05 and VO30DEE10 respectively. While at 60% load, it registers 83.8, 75.2, 70.8 and 76.4 J/deg.CA respectively. The NHRR directly effects the RoPR as illustrated in Figures 7.24 and 25. The variation of combustion duration of the tested fuels at different loads is shown in Figure 7.28. It increases with the increasing load for all the test fuels. The DEE added VO30 records a longer combustion duration as compared to the neat VO30.

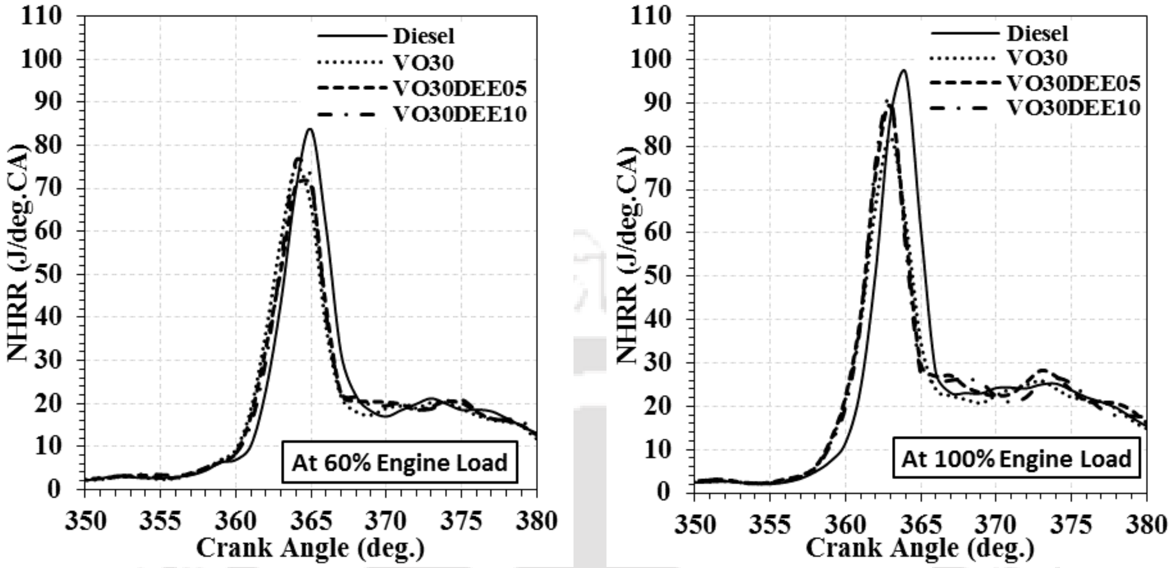


Figure 7.27 NHRR diagram of engine at 60 and 100% engine load for different fuels.

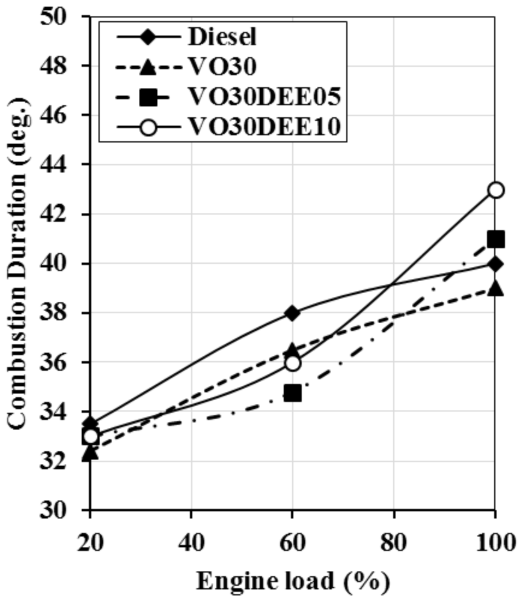


Figure 7.28 Combustion duration of engine at different fuels.

7.3.3 Emissions Analysis

The CO emissions in the exhaust results from the incomplete combustion of the fuel in the engine. The variation in the CO emissions for the test fuels at various engine loads are shown in Figure 7.29. Diesel indicates a lowest CO emissions as compared to the other test fuels.

The CO emissions increases with the use of VO30 in the engine. This increase, on an average, is 62.3% in comparison to the diesel mode. The CO emitted by the engine at full load is 0.026 and 0.044% for diesel and VO30, respectively. This has been attributed to the higher viscosity of *Mesua ferrea* Linn oil. The blending of this VO with diesel results in the higher viscosity. This creates a difficulty in fuel atomization and formation of locally rich mixtures. It is known fact that rich mixtures lead to incomplete combustion due to lack of oxygen and produces CO (Almeida *el al*. 2002; Wang et al. 2006; Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010; Shah and Ganesh 2016). There is a decrease in the CO emissions up to 8 and 13% in the exhaust when VO30 is blended with 5 and 10% DEE, respectively. This reduction in CO emissions has been attributed to better atomization due to reduction in fuel viscosity. Further, the additional oxygen contributed by DEE results in a better combustion (Rakopoulos *et al*. 2011). The improvement in the BTE of engine with DEE in comparison to VO30 as shown in Figure 7.18 signifies that combustion becomes better with the addition of DEE thereby reducing the CO emissions.

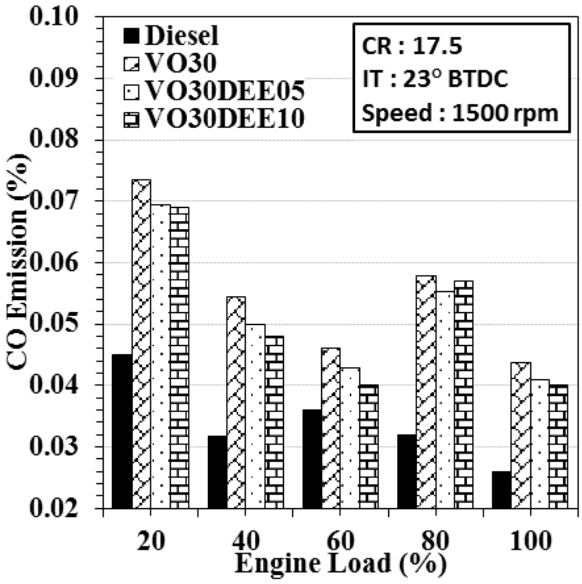


Figure 7.29 Variation of CO emissions with the engine load.

The variation of NO emitted at various loads of the engine run on the test fuels are depicted in Figure 7.30. Generally, there is a growing trend in the amount of NO emissions from engine with the increasing load regardless of fuels used. The NO_X formation is mainly governed by combustion temperature, the residence time of nitrogen at that temperature, and the contents of oxygen in the combustion chamber. The increasing load, therefore, admits more fuels into the combustion chamber leading to higher gas temperature and this makes a favourable for larger NOx formation (Huang *et al*. 2009). The VO30 blend produces lower NO in the exhaust as compared to diesel. At the loads of 20, 60 and 100%, the diesel produces 120, 360 and 578 ppm of NO, while the VO30 produces 104, 309 and 472 ppm respectively. This could be due to the deterioration in combustion with the use of VO30 resulting in lower in-cylinder pressure and temperature which reduces the formation of NO. There is a drop in the NO emissions with the blending of DEE to VO30 as compared to neat VO30. It drops from 309, 417 and 472 ppm of VO30 to 303, 412 and 457 ppm with 5% DEE blending and 283, 409 and 416 ppm with 10% DEE blending at loads of 60, 80 and 100% respectively. This may be attributed to the lower temperature during the period of combustion with the use of DEE, which do not favour the formation of NOx despite DEE being an oxygenated fuel (Rakopoulos *et al*. 2014).

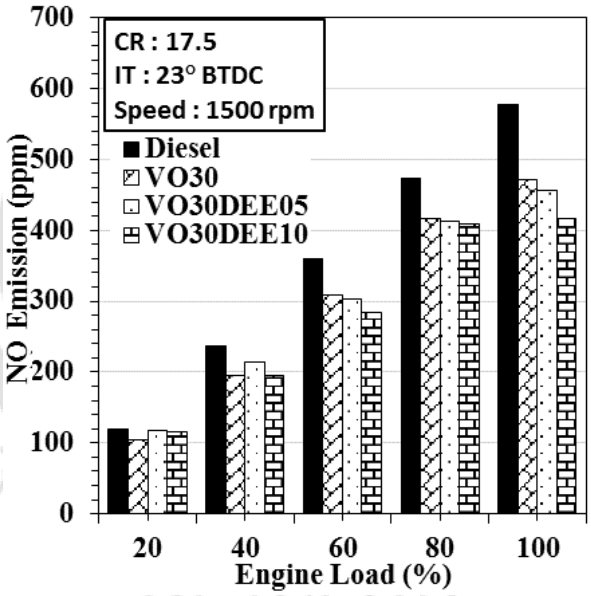


Figure 7.30 Variation of NO emissions with the engine load.

The unburned HC emission of the engine increases with the increasing load as revealed in Figure 7.31. This has been attributed to the increasing fuel-air equivalence ratio with increase of engine load as depicted in Figure 7.19. The increasing fuel-air equivalence ratio results in the relatively higher unavailability of oxygen for the combustion process and increase the formation of HC (Almeida *et al*. 2002; Wang *et al*. 2006; Agarwal and Rajamanoharan 2009; Devan and Mahalakshmi 2009; Chauhan *et al*. 2010). There is an increase in the emissions of

HC with the use of VO30 in the engine. At the loads of 20, 60 and 100%, diesel indicates 12, 18 and 27 ppm of HC in the exhaust, while VO30 shows HC of 30, 24 and 33 ppm, respectively. Higher amount of HC with VO30 is possibly due to the same causes which is responsible for formation higher CO as discussed above. The emission of HC increases, on an average, by 20 and 48% with the use of VO30DEE05 and VO30DEE10 respectively as compared to VO30. Sivalakshmi and Balusamy (2013) and Venu and Madhavan (2017) have cited that due to the high latent heat of vaporization of DEE, it lowers the combustion temperature, particularly around cylinder walls at time of mixture formation resulting in production of more HC from cylinder boundary. While Rakopoulos (2013) sums it up that the increase in spray life due to DEE is causing undesirable fuel impingement on the walls of combustion chamber and this leads to flame quenching and increase in the lean out flame zone which facilitates the production of higher HC. The VO30 blend records the lowest CO_2 emissions among all the test fuels as illustrated in Figure 7.32. This evidently reflects the poor combustion with the use of VO30 in the engine. In comparison to VO30DEE05, the VO30DEE10 has higher CO_2 emission.

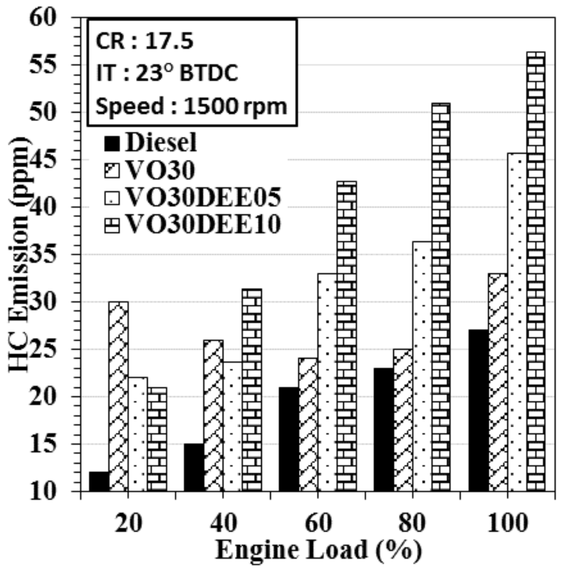


Figure 7.31 Variation of HC emissions with the engine load.

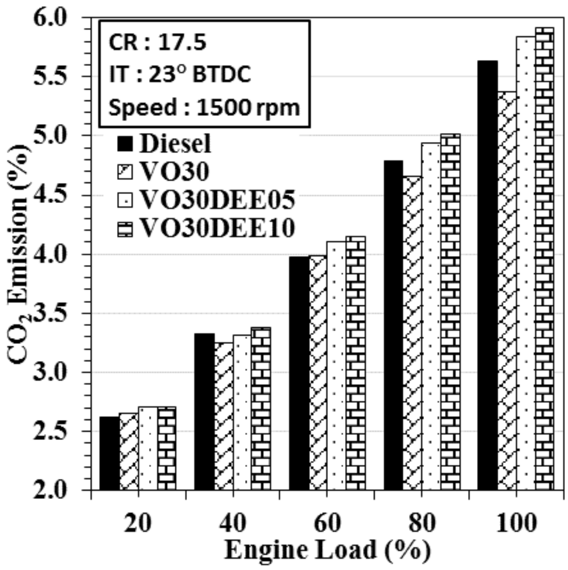


Figure 7.32 Variation of CO_2 emissions with the engine load.

7.3.4 Summary of Module II

- The addition of diethyl ether on VO30 as overall showing a little increase in the BSFC especially at load of 40, 60, and 100% load. Conversely, at the lower load of 20 and 40% BSFC is higher with the use of DEE.

- The addition of diethyl ether to VO30 increases the BTE of the engine as compared to neat VO30. This increase, on an average, is 0.6% and 1.6% with the addition of 5 and 10% diethyl ether respectively.
- There is not much significant variation in the EGT with the use of diethyl ether.
- There is a decrease in the CO emissions up to 8 and 13% in exhaust with the blending 5 and 10% diethyl ether with VO30 respectively.
- There is a drop in the NO emissions with the blending of DEE to VO30 as compared to neat VO30. It drops from 309, 417 and 472 ppm of VO30 to 303, 412 and 457 ppm with 5% DEE blending and 283, 409 and 416 ppm with 10% DEE blending at the engine load of 60, 80 and 100% respectively.
- The emissions of HC increases, on an average, by 20 and 48% with the use of VO30DEE05 and VO30DEE10 respectively as compared to VO30.

7.4 Overall Summary

In this part of the work, the experiments have been conducted on *Mesua ferrea* Linn oil-diesel blend (VO20 and VO30) with the same experimental matrix of the previous investigation (Chapter 6) except the fact that ethanol is replaced by DEE. The major findings of the study using DEE are summarized below:n *Mesua ferrea* Linn oil-diesel blend.

- With the addition of DEE in VO20 blend, the BSFC of engine increases which is greater with larger the amount of DEE added especially at higher loads. For both VO20 and VO30 blends, the BSFC, on an average, increases approximately by 1.1 and 2 % with the use of 5 and 10% DEE respectively.
- DEE in VO20 does not show much variation in the BTE. However, at the lower loads of 20 and 40%, there is a little improvement in BTE. At 20% load, it increases from 10.9% to 11.4 and 11.2% with the use of 5 and 10% DEE respectively. While it increases from 17.7% to 18.3 and 17.9% with VO20DEE05 and VO20DEE10 respectively. The addition of DEE to VO30 increases the BTE of the engine as compared to neat VO30 which, on an average, is 0.6 and 1.6% with the addition of 5 and 10% DEE respectively.
- An increase in the EGT is observed with the addition of DEE in VO20 blend which increases up to 3.4 and 4.4% with the use of 5 and 10% DEE respectively. There is not much significant variation in the EGT with the use of DEE in VO30 blend.

- The use of DEE in VO20 reduces the emissions of CO. It reduces by 2.2 to 8.3% with the use of VO20DEE05 as compared to VO20. The reduction in CO emissions of VO20DEE10 is in the range of 4.3 to 13.6% as compared to VO20. There is also a decrease in the CO emissions up to 8 and 13% in the exhaust with the blending 5 and 10% DEE with VO30 respectively.
- The addition of DEE in VO20 and VO30 blends reduce the NO emissions. However, the HC emission increases with the addition of DEE with VO20 and VO30 blends.

A direct comparison of additives on the performance and emission behaviour of the CI engine run on VO20 and VO30 have been made in Table 7.4 and Table 7.5, respectively.

Table 7.4 Effect of ethanol and DEE on VO20 blend.

	VO20	
Performance		
	5% Ethanol blend	**5% DEE blend**
BTE	Increases up to 3.5%	Increases up to 4.4%
BSFC	Increases in the range of 1.4 to 1.9%	Increases in the range of 0.6 to 1.7%
EGT	1.3 to 4.8%	Increases Up to 3.7%
Emissions		
CO	Reduces up to 34%	Reduces up to 7.4%
NO	Reduces up to 57%	Reduces by 6.9 to 19.3%
HC	Decreases on avg by 19%	Increases on an average by 14.4%
Performance		
	10% Ethanol blend	**10% DEE blend**
BTE	Increases in the range of 1 to 4%	Increases up to 2.8%
BSFC	Increases in the range of 1.8 to 2.9%	Increases in the range of 0.8 to 3.1%
EGT	Increases in the range of 2.1 to 5.4%	Increases in the range of 1.4 to 4.9%
Emissions		
CO	Reduces up to 11%	Reduces up to 13.6%
NO	Increases by 3.8% in intermediate loads	Reduces by 15.5 to 25.6%
HC	Increases on avg by 9%	Increases on an average by 55%

Table 7.5 Effect of ethanol and DEE on VO30 blend.

VO30

Performance

	5% Ethanol blend	**5% DEE blend**
BTE	Increases up to 1.3%	Increases up to 1%
BSFC	Increases in the range of 0.4 to 4.4%	Increases in the range of 2.3%
EGT	Increases up to 3.8%	Does not showing much difference

Emissions

CO	Increases on avg by 17%	Reduces up to 6.9%
NO	Reduces by up to 7.2%	Reduces by 1.2 to 3.2%
HC	Increases on avg by 37%	Increases on an average by 31%

Performance

	10% Ethanol blend	**10% DEE blend**
BTE	Increases in the range of 0.5 to 1.5%	Increases up to 2.3%
BSFC	Increases in the range of 1.9 to 5.2%	Increases up to 3.1%
EGT	Increases up to 3.8%	Does not showing much difference

Emissions

CO	Increases on avg by 28%	Reduces up to 13%
NO	Reduces up to 10.8 %	Reduces by 0.5 to 11.9%
HC	Increases on avg by 51%	Increases on an average by 65%

The use of DEE as an additive in VO20 and VO30 shows a better overall performance as compared to ethanol. The BTE increases with decreases in the emissions of CO and NO.

Chapter 8

Energy and Exergy Analyses

OVERVIEW

This chapter focuses into the energy and exergy analyses of the CI engine run on the test fuels. The chapter begins with a brief literature review on the energy and exergy analyses performed on CI engines by various researchers. This is followed by the energy analysis of the engine run on test fuels to give an insight on how the supplied fuel energy is utilized for shaft work and losses to exhaust, cooling water, and others. The chapter progresses with the exergy analysis which provides an information about the maximum useful work that can be derived from a system. Important findings of the study are summarized at the end.

Chapter Outline:

8.1 Introduction

The increasing gap in the supply and demand of fuel, due rapid urbanization and growing population across the world, demands an absolute need of mechanisms to conserve the energy and minimize waste. This can be accomplished by adopting new policies on energy utilization and by upgrading the technology of existing energy conversion system with focus on improving the effective utilization of the energy and reducing the wastage of energy. The thermodynamic (energy and exergy) analysis serves as a mathematical tool to evaluate the quantity and quality of energy utilized and loss in the system.

Energy analysis is based on the first law of thermodynamics and deals with the quantity of energy and states that energy cannot be created or destroyed. On the other hand, the exergy analysis is based on second law of thermodynamics and deals with quality of energy. Exergy, also termed as availability or available energy, is the maximum theoretical work obtainable from a system at a given state under specified environment. Thus, the exergy represents the work potential of the energy contained in a system at a specified state. Unlike energy, exergy is not conserved and is destroyed by irreversibility (Cengel and Boles 2006; Moran *et al*. 2011). Many studies have been carried out on energy and exergy analyses of CI engines, and a brief literature review is made in the following section.

Flynn *et al.* (1984) used second law analysis on turbocharged diesel engine to evaluate the magnitude of various losses and the effects of operating variables. Al-Najem and Diab (1992) carried out an energy-exergy analysis of turbo-charged, Mercedes-Benz OM422A diesel engine and found that about 50% of the chemical availability of the fuel was destroyed due to uncounted factors while about 15% was lost in the cooling water or exhaust gases. The energy analysis revealed of 25.7, 24.1 and 14.7% of input fuel energy being wasted in the cooling water, exhaust gases and uncounted factors, respectively. In the study of chemical energies and exergies of fuels, Stepanov (1995) opined the energy and exergy analyses reveal the energy potentials of fuels than the calorimetric values more correctly and completely. Canakci and Hosoz (2006) performed energy and exergy analyses of a diesel engine powered by various biodiesels. The tested biodiesels were found to offer almost the same energetic performance to that of fossil diesel, while exergetic performance parameters generally followed the similar trends to that of the corresponding energetic ones. Sahoo *et al.* (2011) carried out second law analysis of a CI engine powered by syngas-diesel under dual fuel mode at the varying load condition and varying amount of hydrogen. The study revealed an increase in the cumulative work availability and a reduction in destroyed availability with increasing hydrogen quantity of syngas. This was attributed to the improved combustion process and increased work output with higher amount of hydrogen. The dual fuel cumulative work availability was found to increase at higher loads resulting in the increase of exergy efficiency.

The second law analysis on diesel engine powered by water emulsified palm biodiesel showed a potential of trapping up to 40% of the fuel availability. With the increase in compression ratio (CR), the shaft availability was found to increase, while the availability destruction remained unaffected. The retarding of injection timing (IT) resulted in higher shaft availability (Debnath *et al*. 2014a). Singh *et al*. (2017) reported that a maximum 30% of fuel input energy can be converted into useful energy for the engine powered by biodiesel. While studying the effect of fuel oxygen on the energetic and exergetic efficiency of a CI engine, Jena and Misra (2014) demonstrated the possibility of better combustion along with a lower irreversibility with higher amount of oxygen content in the fuel.

With the help of energy-exergy analysis, Bora and Saha (2016a) studied the theoretical performance limits of a biogas–diesel run dual fuel diesel engine for different combinations of CR and IT. Energy transferred to the cooling water was found to increase with the use of higher CR and IT advancement. The exergy analysis revealed on an average, 33.44% of the fuel energy being converted to useful energy. The shaft availability increased with the increase of CR up to 18 and IT advancement up to 29° before top dead centre (bTDC). Krishnamoorthi and Malayalamurthi (2017) carried out an exergy analysis of a variable compression ratio engine fueled with diesel-aegle marmelos oil-diethyl ether blends to maximize the work availability and reduce the destroyed availability. Paul *et al.* (2017) studied the effect of biodiesel and ethanol on the exergy characteristics of a diesel engine by blending diesel-ethanol with pongamia pinata methyl ester. The exergetic efficiency of base diesel was found to be poor as compared to the blends. The increase in the percentage of ethanol upto 15% in the diesel-biodiesel blends increased the exergetic efficiency.

While studying on the effect of combustion chamber geometry on exergy and BTE, it was found that toroidal combustion chamber showed an improved exergy and BTE than the hemispherical and trapezoidal combustion chambers at part load condition (Karthickeyan 2019). As reported by Nabi *et al.* (2019), the fuel energy and exergy were found to be slightly lower with the use of biodiesel blends in a diesel engine. The fuel blends were prepared by maintaining fuel oxygen constant with 3.35% by weight. Odibi *et al*. (2019) carried out the exergy and energy analysis of a diesel engine powered by oxygenated fuel based on waste cooking biodiesel and a triacetin. The use of oxygenated fuels was found to improve the BTE with the consequence of lowering the exhaust energy loss. Exhaust temperature is lowered with the used of oxygenated fuel leading to lower exhaust exergy loss and higher exergetic efficiency.

According to Azoumah *et al*. (2009), the thermal efficiencies were not sufficient to judge the performance of an engine. Therefore, the combination of exergy and gas emissions analysis was used as a tool for evaluating the optimal load that can be delivered by engine. Krishnamoorthi and Malayalamurthi (2018) reported the higher input availability of fuel and cooling water availability with the increase in the amount of VO in the blend consisting of diesel, VO and DEE. Das *et al.* (2020) studied the energy-exergy analysis on a DI single cylinder diesel engine powered by pyrolytic waste plastic oil (WPO) diesel blend. The fuel exergy rate of the blended fuels were found to be higher than the neat diesel. The exergetic efficiency of blended fuels were found to increase with an increase in engine load. While the exergetic efficiency decreases with the increase in the concentration of WPO in WPO-diesel blend as compared to diesel. In the energy and exergy analyses of a CI engine fueled by tire pyrolytic oil (TPO)-diesel blends consisting of 10, 30 and 50% TPO by volume, the highest energy and exergy efficiencies were obtained for the blend with 10% TPO by volume (Karagoz *et al*. 2020). With higher amount of TPO in the blend, the exergy and energy efficiencies were found to decrease. Sanli *et al*. (2020) carried out the energy and exergy analyses of a diesel engine at different ambient temperatures. The engine was powered by diesel fuel and microalgae biodiesel (MAB). The energetic and exergetic efficiencies were found be slightly higher with the use of diesel than with the use of MAB. The exergy destruction rate increased with increasing ambient temperature, while the heat transfer exergy rate and the exhaust exergy rate decreased. Sarıkoc *et al*. (2020) studied the effects of biodiesel and butanol properties on energetic-exergetic efficiencies of a diesel engine. It was found that the energetic-exergetic efficiencies of biodiesel were higher than the euro diesel.

Sharma *et al*. (2020) studied the effect of hydroxyl gas (HHO) variation on energy and exergy under dual fuel mode. The BTE was found to increase with the amount of HHO increases. At the optimized condition, the energy losses increased by 6.29% and 8.55% for the exhaust gas and the heat transfer, respectively. The work availability, exhaust gas and heat transfer exergy were found to increase. This was attributed to the higher diffusivity of hydrogen and faster oxidation of fuel species in the cylinder. In the energetic and exergetic analysis of a diesel engine powered by peanut oil biodiesel and conventional diesel fuel, the maximum energy and exergy efficiencies were found at full load (Yesilyurt 2020). Being fueled with biodiesel, the efficiencies were found to be lower as compared to diesel.

From the aforementioned review, it is evident that without the inclusion of energy-exergy analysis, the study of an engine through performance, combustion and emissions is incomplete. Therefore, an attempt has been made to carry out the energy analysis and exergy analyses of the CI engine run on the test fuels. The test fuels consists of diesel, three diesel-VO blends (VO10, VO20, VO30), four diesel-VO-ethanol blends (VO20E05, VO20E10, VO30E05, VO30E10) and four diesel-VO-diethyl ether blends (VO20DEE05, VO20DEE10, VO30DEE05, VO30DEE10). The expressions used in this study are incorporated in Appendix C. The energy analysis includes amount of fuel energy supplied in unit time (Q_{in}), energy converted to shaft power or shaft energy per unit time (Q_S), energy transferred to the cooling water per unit time (Q_W), energy loss through the exhaust per unit time (Q_E) and uncounted energy losses per unit time (Q_U). While the exergy analysis includes input availability of the fuel (A_{in}), shaft availability (A_S), availability input converted to cooling water availability (A_W), availability transferred exhaust or exhaust gas availability (A_E), availability destroyed (A_D) and exergy efficiency (η_{II}). The results of energy and exergy analysis is discussed in the following sections.

8.2 Energy Analysis

The results of energy analysis of all the tested fuels at the engine loads of 20, 60 and 100% is presented in Table 8.1**,** Table 8.2 and Table 8.3, respectively. The energies are expressed in terms of their values in kW and also in the percentage of the supplied fuel energy (Q_{in}). At 20% load, on an average, only 11% of supplied fuel is converted to the shaft work, 41% transferred to cooling water and 34% of supplied fuel is lost through exhaust gas. The uncounted energy loss constitutes of 14%. The shaft power output increases to 22% (on an average) of the supplied fuel as the engine load is increased to 60%, while the energy transferred to cooling water is decreased to 35%. The energy lost to the exhaust remains at 33% and uncounted loss reduces to 9%. The engine load of 100% shows an increase in the shaft energy output which lies in the range of 27.1 to 28.8% of the supplied fuel energy. This resulted in the reduction of uncounted loss to 8% (on an average). Also, the heat loss to the cooling water reduces to 32%, while the average exhaust gas lost remains at 33%. With the increase in the content VO in the VO-diesel blend, the fuel energy input increases as compared to neat diesel. At 60% load, the fuel energy input is 8.94, 8.97, 9.16 and 9.34 kW for diesel, VO10, VO20 and VO30, respectively. While at 100% load, it is 11.88, 12.22, 12.38 and 12.45 kW, respectively. This results in the decrease of energy efficiency with VO blend. At 100% load, the fuel energy input is 11.88, 12.22, 12.38 and 12.45 kW for diesel, VO10, VO20 and VO30, respectively. Overall, the addition of ethanol and DEE in VO-diesel blends improves the energy efficiency which is evident from the lower fuel energy input. The energy distributions of the test fuels as a function of input fuel at the different engine load are shown in Figure 8.1 (for 20 and 60% load) and Figure 8.2 (for 100% load). At loads of 20, 60 and 100%, the fuel energy input lies in the range of 5.97 to 6.26 kW, 8.94 to 9.25 kW and 11.88 to 12.64 kW, respectively. The shaft work is found to increase with the increasing engine load. The shaft work is found to be 0.68, 2.05 and 3.42 kW for the load of 20, 60 and 100%, respectively.

Table 8.1 Energy analysis of tested fuels at 20% engine load.

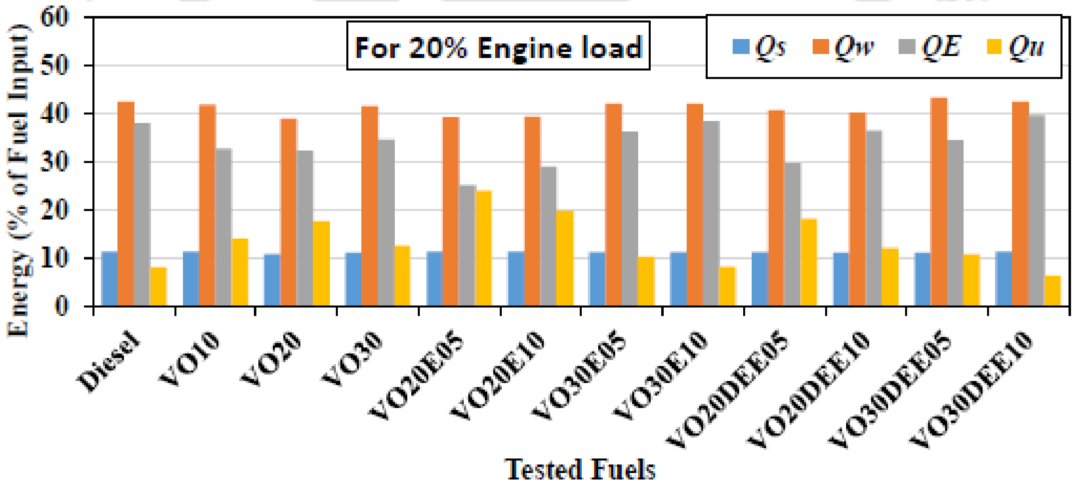
Fuel	Q_{in}		Q_S		Q_W		Q_E		Q_U	
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)
Diesel	5.97	100	0.68	11.4	2.53	42.4	2.27	38.0	0.49	8.2
VO10	5.95	100	0.68	11.4	2.49	41.8	1.94	32.6	0.84	14.1
VO20	6.26	100	0.68	10.9	2.44	39.0	2.03	32.4	1.11	17.7
VO30	6.09	100	0.68	11.2	2.53	41.5	2.11	34.6	0.77	12.6
VO20E05	5.99	100	0.68	11.4	2.35	39.2	1.51	25.2	1.44	24.0
VO20E10	5.97	100	0.68	11.4	2.35	39.4	1.73	29.0	1.19	19.9
VO30E05	6.02	100	0.68	11.3	2.53	42.0	2.18	36.2	0.62	10.3
VO30E10	6.02	100	0.68	11.3	2.53	42.0	2.31	38.4	0.50	8.3
VO20DEE05	6.00	100	0.68	11.3	2.44	40.7	1.78	29.7	1.09	18.2
VO20DEE10	6.09	100	0.68	11.2	2.44	40.1	2.22	36.5	0.74	12.2
VO30DEE05	6.05	100	0.68	11.2	2.62	43.3	2.09	34.5	0.66	10.9
VO30DEE10	5.96	100	0.68	11.4	2.53	42.4	2.36	39.6	0.38	6.4

Table 8.2 Energy analysis of tested fuels at 60% engine load.

Fuel	Q_{in}		Q_S		Q_W		Q_E		Q_U	
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)
Diesel	8.94	100	2.05	22.9	3.31	37.0	2.84	31.8	0.73	8.2
VO10	8.97	100	2.05	22.9	3.27	36.5	2.67	29.8	0.98	10.9
VO20	9.16	100	2.05	22.4	3.31	36.1	3.07	33.5	0.73	8.0
VO30	9.34	100	2.05	21.9	3.31	35.4	3.44	36.8	0.53	5.7
VO20E05	9.22	100	2.05	22.2	3.18	34.5	2.49	27.0	1.49	16.2
VO20E10	9.15	100	2.05	22.4	3.10	33.9	2.66	29.1	1.34	14.6
VO30E05	9.25	100	2.05	22.2	3.14	33.9	3.40	36.8	0.66	7.1
VO30E10	9.20	100	2.05	22.3	3.23	35.1	3.28	35.7	0.64	7.0
VO20DEE05	9.18	100	2.05	22.3	3.05	33.2	3.13	34.1	0.95	10.3
VO20DEE10	9.20	100	2.05	22.3	3.14	34.1	3.31	36.0	0.69	7.5
VO30DEE05	9.19	100	2.05	22.3	3.40	37.0	2.85	31.0	0.88	9.6
VO30DEE10	9.21	100	2.05	22.3	3.23	35.1	3.6	39.1	0.33	3.6

Table 8.3 Energy analysis of tested fuels at 100% engine load.

Fuel	Q_{in}		Q_S		Q_W		Q_E		Q_U	
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)
Diesel	11.88	100	3.42	28.8	3.92	33.0	3.95	33.2	0.59	5.0
VO10	12.22	100	3.42	28.0	4.1	33.6	3.59	29.4	1.11	9.1
VO20	12.38	100	3.42	27.6	3.92	31.7	3.92	31.7	1.12	9.0
VO30	12.45	100	3.42	27.5	4.01	32.2	4.41	35.4	0.61	4.9
VO20E05	12.56	100	3.42	27.2	3.79	30.2	3.48	27.7	1.87	14.9
VO20E10	12.38	100	3.42	27.6	3.88	31.3	3.29	26.6	1.79	14.5
VO30E05	12.49	100	3.42	27.4	4.01	32.1	4.58	36.7	0.48	3.8
VO30E10	12.32	100	3.42	27.8	4.01	32.5	4.51	36.6	0.38	3.1
VO20DEE05	12.48	100	3.42	27.4	3.84	30.8	4.29	34.4	0.93	7.5
VO20DEE10	12.55	100	3.42	27.3	3.84	30.6	4.30	34.3	0.99	7.9
VO30DEE05	12.64	100	3.42	27.1	4.1	32.4	4.20	33.2	0.92	7.3
VO30DEE10	12.34	100	3.42	27.7	4.1	33.2	4.33	35.1	0.49	4.0



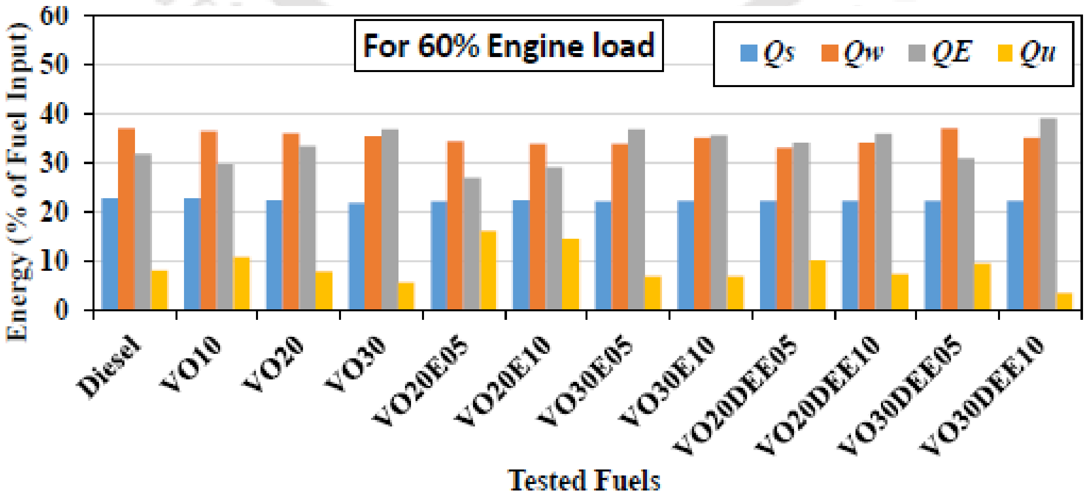


Figure 8.1 Energy distribution with fuel input as function at the engine load of 20 and 60% for tested fuels.

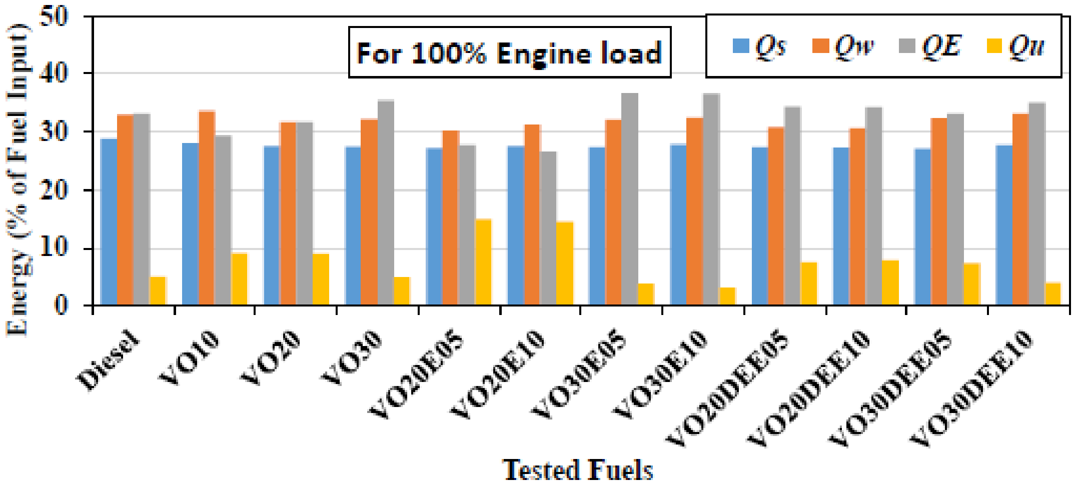


Figure 8.2 Energy distribution with fuel input as function at the engine load of 100% for the tested fuels.

The shaft work lies in the range of 10.9 to 11.4%, 21.9 to 22.9% and 27.1 to 28% of the input fuel energy at loads of 20, 60 and 100%, respectively. The energy transferred to cooling water and uncounted loss decreases with the increasing engine load. The energy transferred to cooling water is in the range of 39 to 43.3%, 33.9 to 37% and 30.2 to 33.6% of the input fuel energy at loads of 20, 60 and 100%, respectively. The uncounted loss, on an average, is 14, 9 and 8% of the input fuel energy at the loads of 20, 60 and 100%, respectively. The energy loss to exhaust, on an average, is 34% of Q_{in} for 20% load. While it is, on an average, is 33% of Q_{in} for both the loads of 60 and 100%.

8.3 Exergy Analysis

The result of exergy analysis of the tested fuels at loads of 20, 60 and 100% is shown in Table 8.4, Table 8.5 and Table 8.6 respectively. The energies are expressed in terms of their values in kW and also in the percentage of the input availability of the fuel with the input availability of the fuel being assigned 100%. The exergy distributions of the tested fuels as a function of input fuel at the engine load of 20, 60 and 100% is also been shown in Figure 8.3. The input availability of the fuel is in the range 6.15 to 6.48 kW, 9.24 to 9.65 kW and 12.28 to 13.06 kW at the loads of 20, 60 and 100%, respectively. The input fuel availability increases with the increase in the amount of VO in the VO-diesel blend. At 100% load, the input fuel availability is 12.28, 12.63, 12.80 and 12.87 kW for diesel, VO10, VO20 and VO30, respectively.

Table 8.4 Exergy analysis of tested fuels at 20% engine load.

Fuel	A_{in}		A_S		A_W		A_E		A_D		η_{II}
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(%)
Diesel	6.17	100	0.68	11.0	0.07	1.1	0.31	5.0	5.10	82.7	17.28
VO10	6.15	100	0.68	11.1	0.09	1.5	0.30	4.9	5.08	82.6	17.38
VO20	6.48	100	0.68	10.5	0.07	1.1	0.30	4.6	5.43	83.8	16.19
VO30	6.30	100	0.68	10.8	0.07	1.1	0.31	4.9	5.24	83.2	16.83
VO20E05	6.19	100	0.68	11.0	0.09	1.5	0.21	3.4	5.20	84.0	15.93
VO20E10	6.17	100	0.68	11.0	0.09	1.5	0.29	4.7	5.11	82.8	17.20
VO30E05	6.22	100	0.68	10.9	0.07	1.1	0.31	5.0	5.15	82.8	17.13
VO30E10	6.23	100	0.68	10.9	0.07	1.1	0.32	5.1	5.16	82.8	17.17
VO20DEE05	6.20	100	0.68	11.0	0.07	1.1	0.26	4.2	5.19	83.7	16.31
VO20DEE10	6.30	100	0.68	10.8	0.06	1.0	0.34	5.4	5.20	82.5	17.34
VO30DEE05	6.26	100	0.68	10.9	0.07	1.1	0.28	4.5	5.22	83.4	16.57
VO30DEE10	6.16	100	0.68	11.0	0.07	1.1	0.36	5.8	5.05	82.0	18.03

Table 8.5 Exergy analysis of tested fuels at 60% engine load.

Fuel	A_{in}		A_S		A_W		A_E		A_D		η_{II}
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(%)
Diesel	9.24	100	2.05	22.2	0.10	1.1	1.16	12.6	5.93	64.2	35.85
VO10	9.28	100	2.05	22.1	0.13	1.4	1.12	12.1	5.98	64.4	35.58
VO20	9.47	100	2.05	21.6	0.11	1.2	1.18	12.5	6.13	64.7	35.26
VO30	9.65	100	2.05	21.2	0.10	1.0	1.32	13.7	6.18	64.0	35.97
VO20E05	9.53	100	2.05	21.5	0.13	1.4	1.06	11.1	6.28	65.9	34.12
VO20E10	9.46	100	2.05	21.7	0.13	1.4	1.05	11.1	6.23	65.9	34.17
VO30E05	9.56	100	2.05	21.4	0.09	0.9	1.37	14.3	6.04	63.2	36.82
VO30E10	9.51	100	2.05	21.6	0.10	1.1	1.42	14.9	5.94	62.5	37.51
VO20DEE05	9.49	100	2.05	21.6	0.09	0.9	1.33	14.0	6.02	63.4	36.57
VO20DEE10	9.51	100	2.05	21.6	0.09	0.9	1.34	14.1	6.02	63.3	36.69
VO30DEE05	9.50	100	2.05	21.6	0.10	1.1	1.23	12.9	6.11	64.3	35.66
VO30DEE10	9.52	100	2.05	21.5	0.10	1.1	1.28	13.4	6.08	63.9	36.08

Table 8.6 Exergy analysis of tested fuels at 100% engine load.

Fuel	A_{in}		A_S		A_W		A_E		A_D		η_{II}
	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(kW)	(%)	(%)
Diesel	12.28	100	3.42	27.9	0.13	1.1	1.16	9.4	7.57	61.6	38.36
VO10	12.63	100	3.42	27.1	0.18	1.4	1.12	8.9	7.91	62.6	37.36
VO20	12.80	100	3.42	26.7	0.14	1.1	1.18	9.2	8.06	63.0	37.03
VO30	12.87	100	3.42	26.6	0.14	1.1	1.32	10.3	8.00	62.2	37.87
VO20E05	12.99	100	3.42	26.3	0.17	1.3	1.06	8.2	8.33	64.1	35.85
VO20E10	12.80	100	3.42	26.7	0.17	1.3	1.05	8.2	8.15	63.7	36.31
VO30E05	12.91	100	3.42	26.5	0.14	1.1	1.37	10.6	7.98	61.8	38.20
VO30E10	12.74	100	3.42	26.8	0.13	1.0	1.42	11.1	7.77	61.0	39.05
VO20DEE05	12.91	100	3.42	26.5	0.13	1.0	1.33	10.3	8.03	62.2	37.80
VO20DEE10	12.97	100	3.42	26.4	0.13	1.0	1.34	10.3	8.08	62.3	37.70
VO30DEE05	13.06	100	3.42	26.2	0.14	1.1	1.23	9.4	8.28	63.4	36.66
VO30DEE10	12.75	100	3.42	26.8	0.14	1.1	1.28	10.0	7.91	62.0	37.99

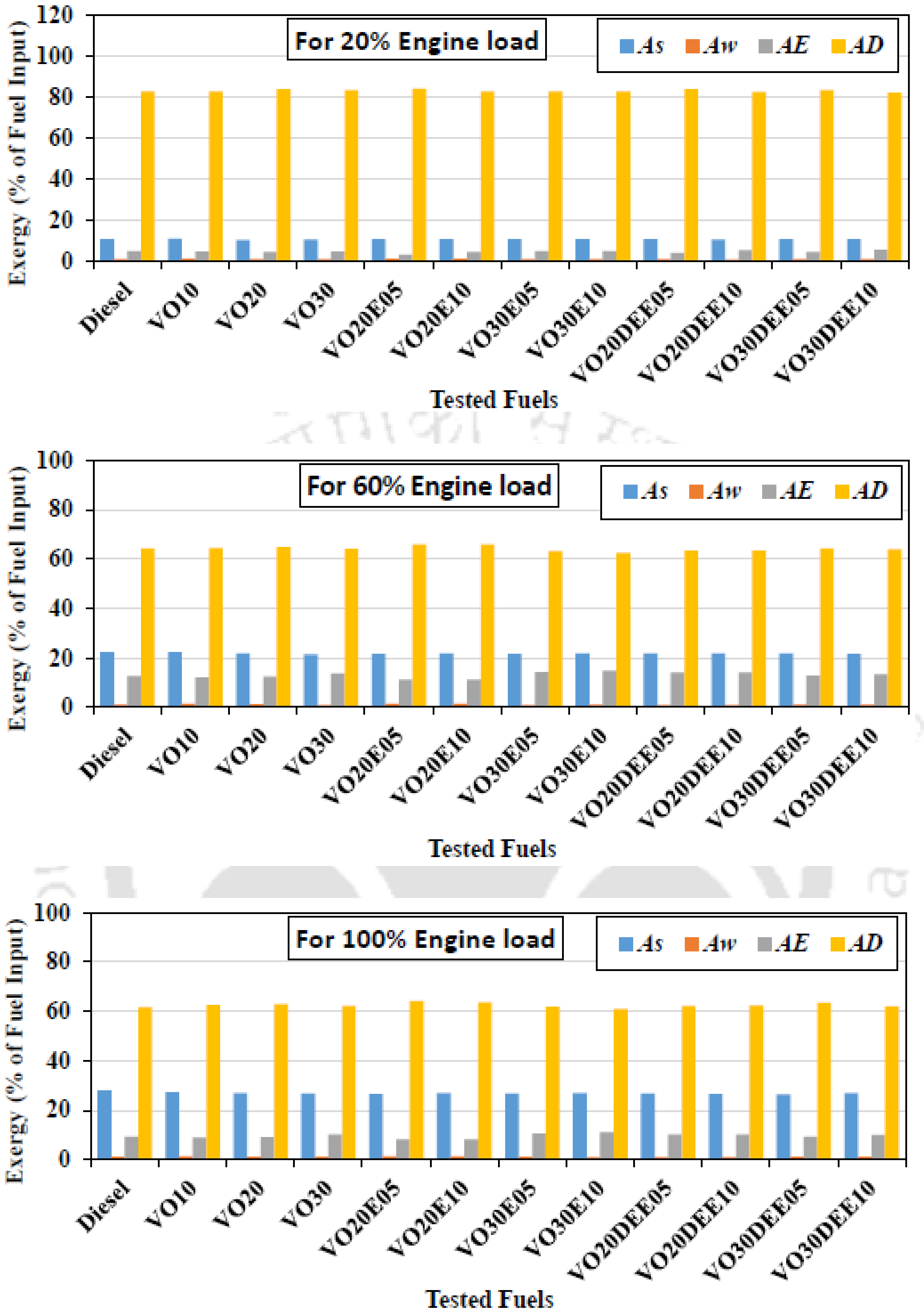


Figure 8.3 Exergy distribution with fuel input as function at different loads for the tested fuels.

The availability destroyed decreases and the work or shaft availability increases with the increasing load. In comparison to diesel, the blended fuels recorded a higher destructed availability and a lower shaft availability against the input fuel availability. On an average, the availability destroyed is 83, 64 and 62% of input availability of fuel at the loads of 20, 60 and 100%, respectively. While the shaft availability is 11, 22 and 27% of the input availability of fuel at the respective loads. This resulted in the increase of exergy efficiency with the increasing load. The exergy efficiency is in the range of 16.19 to18.03%, 34.12 to 37.51% and 36.66 to 39.05% at the loads of 20, 60 and 100%, respectively.

The exergy efficiency decreases with the increase in the content of VO in the VO-diesel blend. The addition of ethanol to VO30 blend shows a significant improvement in the exergy efficiency. However, the addition of ethanol to VO20 blend does not show a significant improvement in the exergy efficiency. The addition of DEE to VO20 and VO30 blends improves the exergy efficiency. The exhaust gas availability increases till the 60% of the engine load, and then decreases at end of 100% load. At the loads of 20, 60 and 100%, the exhaust availability, on an average, is 5, 13 and 10% of input availability of the fuel, respectively. The engine BTE, on an average is 11, 22 and 28% at the load of 20, 60 and 100% respectively. While the exergy efficiency average, is 17, 36 and 38%, respectively. This difference in the energy and exergy efficiencies reflects the effective utility of fuel in the engine.

8.4 Summary

The energy and the exergy analyses of the CI engine powered by a VO (*Mesua ferrea* Linn oil) blended with diesel, ethanol and DEE have been carried out. A total of twelve fuels/fuel blends have been tested in the engine to evaluate the various energy and exergy parameters. The key findings of the study are summarized below:

- At the engine load of 20, 60 and 100%, fuel energy input is in the range of 5.97 to 6.26 kW, 8.94 to 9.25 kW and 11.88 to 12.64 kW, respectively.
- The shaft work increases with the increasing engine load. The shaft work is 0.68, 2.05 and 3.42 kW for the load of 20, 60 and 100% respectively. It is in the range of 10.9 to 11.4%, 21.9 to 22.9% and 27.1 to 28% of the input fuel energy at the engine load of 20, 60 and 100% respectively.
- At 20% load, on an average, only 11% of supplied fuel is converted to the shaft work, 41% transferred to cooling water and 34% of supplied fuel lost through exhaust gas. The uncounted energy loss constitutes of 14%.

- The shaft power output increases to on an average to 22% of the supplied fuel as the engine load increase to 60% while energy transferred to cooling water decreases to 35%. The energy lost to exhaust remains at 33% and uncounted loss reduces to 9%.
- The 100% engine load shows an increase in the shaft energy output which is in the range of 27.1 to 28.8% of the supplied fuel energy. This resulted in the reduction of uncounted loss to 8% on an average. Also the heat loss to the cooling water reduces to 32% while the average exhaust gas lost remains at 33%.
- The energy transferred to cooling water and uncounted loss decreases with the increasing engine load. The energy transferred to cooling water is in the range of 39 to 43.3%, 33.9 to 37% and 30.2 to 33.6% of the input fuel energy at the engine load of 20, 60 and 100% respectively. The energy loss to exhaust also decreases with the increasing load.
- Fuel energy input increases with increase in the content VO in the VO-diesel blend as compared to neat diesel. Overall, the addition of ethanol and DEE in VO-diesel blends improves the energy efficiency.
- The input availability of the fuel is in the range 6.15 to 6.48 kW, 9.24 to 9.65 kW and 12.28 to 13.06 kW at the engine load of 20, 60 and 100% respectively.
- The availability destroyed decreases and work or shaft availability increases with the increasing load. On an average the availability destroyed it is 83, 64 and 62% of input availability of the fuel at the engine load of 20, 60 and 100% respectively.
- The shaft availability is 11, 22 and 27% of the input availability of the fuel at the engine load of 20, 60 and 100%, respectively.
- The exergy efficiency increases with the increasing load. The exergy efficiency is in the range of 16.19 to18.03%, 34.12 to 37.51% and 36.66 to 39.05% at the engine load of 20, 60 and 100%, respectively.
- The exhaust gas availability increases till the 60% of the engine load and then decrease at end of 100% load. At the engine load of 20, 60 and 100%, the exhaust availability, on an average, is 5, 13 and 10% of input availability of the fuel respectively.

- The energy efficiency or BTE of the engine, on an average, is 11, 22 and 28% at the load of 20, 60 and 100% respectively. While the exergy efficiency, on an average, is 17, 36 and 38%, respectively.
- Blended fuels recorded a higher destructed availability and a lower shaft availability against the input fuel availability as compared to diesel. Exergy efficiency decreases with the increase in the content of VO in the VO-diesel blend.
- The addition of ethanol to VO30 blend shows a significant improvement in the exergy efficiency. However, with VO20 blend ethanol does not show a significant improvement in the exergy efficiency.
- The addition of DEE to VO20 and VO30 blends improves the exergy efficiency.

Chapter 9

Conclusion and Future Scopes

OVERVIEW

This experimental investigation is carried out to study the applicability of Mesua ferrea Linn oil (a type of VO) in stationary CI engine. The VO extracted from the seeds is processed for its use in the engine. Various combinations of the VO without and with additives have been experimented at different loads of the engine. In each case, the performance, combustion and emissions characteristics of the engine are examined. Furthermore, the energy and exergy analyses of the engine under all the test fuels have been done. This chapter highlights all the key findings of the investigation together with its application potential and future scopes.

Chapter Outline:

9.1 Contribution of the Present Work

The present investigation is carried out to study the performance, combustion and emissions characteristics of a single-cylinder, four-stroke, water cooled, naturally aspirated, constant speed, direct injection, 3.5 kW CI engine run on VO derived from the *Mesua ferrea* Linn seeds. The VO is extracted from the decorticated and dried seeds using hot screw press oil expeller. The filtered and de-moisturized oil is then used for the experiments. The VO, derived from the *Mesua ferrea* Linn seeds, is characterized and is found have lower heating value together with higher viscosity and density as compared to diesel. Therefore, to use this VO in the engine, it has been blended with the diesel. Two additives viz., ethanol and DEE are used separately in the blend of diesel and *Mesua ferrea* Linn oil. The blending reduces the viscosity and density. The key findings of this study are summarized below.

9.1.1 Characteristics of *Mesua ferrea* Linn Oil

- The oil yield was in the range of 33.19 to 53.33% with the hand operated oil expeller while the yield of power operated was in range of 44 to 65%.
- The fatty acid constituting the oil is mainly composed of oleic acid which constitutes 55.93% of the total composition. This is followed by 14.19 and 13.68% stearic and linoleic acid respectively.
- The oil has higher and heating value of 38,280 and 35,611 kJ/kg respectively. The density is 927.8 kg/m^3 at the temperature of 15 °C while kinematic viscosity was found to be 56.107 mm^2/s at 40 °C.

9.1.2 VO-diesel Blends

- Three blends viz., VO10, VO20 and VO30 consisting of 10, 20 and 30% VO by volume have been considered in the experimental investigation.
- The engine suffers slight drop in the BTE with the use of *Mesua ferrea* Linn oil which is up to 1.78, 3.94 and 5.47% respectively with the use of VO10, VO20 and VO30.
- The BSFC increases in the range of 1 to 3%, 3 to 7% and 5 to 11% respectively with 10, 20 and 30% blending.
- The VO blend produces higher CO and HC in the combustion. The CO emissions increases, on an average, by 23 and 39% respectively with the blending of VO by 20

and 30%; while the increase in the HC emissions lies in the range of 3 to 14%, 7 to 46% and 17 to 95% with VO10, VO20 and VO30, respectively.

- The NO emissions decreases with increasing content of VO in the blend.
- The 10% blend show almost similar performance to that of diesel. However, with the higher blends of VO such as VO20 and VO30, the performance of the engine decreases. This is mainly attributed to the higher viscosity of the oil blend.

9.1.3 VO-diesel-ethanol Blends

- To overcome drop in the performance of with the V20 and VO30 blend, ethanol is used as an additive.
- With 10% ethanol blend to VO20 and VO30, the BTE improves by approximately 1 to 4% and 1 to 1.5%, respectively. However, 5% ethanol blend does not show any significant change as compared to neat VO20 and VO30.
- The BSFC of engine increases with the use of the ethanol in both V2O and VO30 blends and is found to be higher with higher in amount of ethanol in the blend. On an average, the increment is 1.7 and 2.5% respectively with the use of 5 and 10% ethanol in the VO20 blend. While it increases, on an average, by 1.8 and 3% with the blending of 5 and 10% ethanol respectively in VO30. This is due to the lower calorific value of the ethanol.
- The EGT of engine records a slight increment with use of ethanol in VO20 blend. This is in the range of 7 to 12 $^{\circ}$C and 8 to 14 $^{\circ}$C, respectively with the use of 5 and 10% ethanol. There is not much significant changes in the EGT with ethanol in VO30 blend.
- The CO emissions of VO20 decreases and NOx increases with the use of ethanol. However, HC emissions decreases with 5% ethanol addition and increases with the use of 10% ethanol in the VO20 blend. The use of ethanol in VO30 results in increase in the emissions CO and HC. While, it reduces the NO emissions up to 7 and 11% with the addition 5 and 10% respectively.

9.1.4 VO-diesel-diethyl ether Blends

- To overcome drop in the performance of the engine run on V20 and VO30 blends, DEE has also been used as an additive.

- With the addition of DEE in VO20 blend, the BSFC of engine increases which is greater with larger the amount of DEE added especially at higher loads. It increases, on an average, approximately by 1.1 and 2 % with the use of 5 and 10% DEE respectively both in VO20 and VO30 blends.
- DEE in VO20 does not show much variation in the BTE. However, at the lower loads of 20 and 40%, there is a little improvement in BTE. At 20% load, it increases from 10.9% to 11.4 and 11.2% with the use of 5 and 10% DEE respectively. While it increases from 17.7% to 18.3 and 17.9% with VO20DEE05 and VO20DEE10 respectively. The addition of DEE to VO30 increases the BTE of engine as compared to neat VO30 which, on an average, is 0.6 and 1.6% with the addition of 5 and 10% DEE respectively.
- An increase in the EGT is observed with the addition of DEE in VO20 blend which increases up to 3.4 and 4.4% with the use of 5 and 10% DEE respectively. There is not much significant variation in the EGT with the use of DEE in VO30 blend.
- The use of DEE in VO20 reduces the emissions of CO. It reduces by 2.2 to 8.3% with the use of VO20DEE05 as compared to VO20. The reduction in CO emissions of VO20DEE10 is in the range of 4.3 to 13.6% as compared to VO20. There is also a decrease in the CO emissions up to 8 and 13% in exhaust with the blending 5 and 10% DEE with VO30 respectively.
- The addition of DEE in both VO20 and VO30 blends reduces the emissions of NO. However, the HC emissions increases with the blending of DEE with VO20 and VO30 blends.

9.1.5 Energy and Exergy Analyses

- At the engine loads of 20, 60 and 100%, the fuel energy input is in the range of 5.97 to 6.26 kW, 8.94 to 9.25 kW and 11.88 to 12.64 kW, respectively.
- The shaft work increases with the increasing engine load. The shaft work is 0.68, 2.05 and 3.42 kW for the load of 20, 60 and 100% respectively. It is in the range of 10.9 to 11.4%, 21.9 to 22.9% and 27.1 to 28% of the input fuel energy at the engine load of 20, 60 and 100% respectively.
- At 20% load, on an average, only 11% of supplied fuel is converted to the shaft work, 41% transferred to cooling water and 34% of supplied fuel lost through exhaust gas. The uncounted energy loss constitutes of 14%.

- The shaft power output increases, on an average, to 22% of the supplied fuel as the engine load increase to 60%, while the energy transferred to cooling water decreases to 35%. The energy loss to exhaust remains at 33% and uncounted loss reduces to 9%.
- The engine at 100% load shows an increase in the shaft energy output which is in the range of 27.1 to 28.8% of the supplied fuel energy. This resulted in the reduction of uncounted loss to 8% on an average. Also the heat loss to the cooling water reduces to 32% while the average exhaust gas lost remains at 33%.
- The energy transferred to cooling water and uncounted loss decreases with the increasing engine load. The energy transferred to cooling water is in the range of 39 to 43.3%, 33.9 to 37% and 30.2 to 33.6% of the input fuel energy at the engine load of 20, 60 and 100%, respectively. The energy loss to exhaust also decreases with the increasing load.
- The fuel energy input increases with increase in the content VO in the VO-diesel blend as compared to neat diesel. Overall, the addition of ethanol and DEE in VO-diesel blends improves the energy efficiency.
- The input availability of the fuel is in the range 6.15 to 6.48 kW, 9.24 to 9.65 kW and 12.28 to 13.06 kW at the engine load of 20, 60 and 100%, respectively.
- The availability destroyed decreases and work or shaft availability increases with the increasing load. On an average the availability destroyed it is 83, 64 and 62% of input availability of the fuel at the engine load of 20, 60 and 100%, respectively.
- The shaft availability is 11, 22 and 27% of the input availability of the fuel at the engine load of 20, 60 and 100%, respectively.
- The exergy efficiency increases with the increasing load. The exergy efficiency is in the range of 16.19 to18.03%, 34.12 to 37.51% and 36.66 to 39.05% at the engine load of 20, 60 and 100%, respectively.
- The exhaust gas availability increases till the 60% of the engine load and then decrease at end of 100% load. At the engine load of 20, 60 and 100%, the exhaust availability, on an average, is 5, 13 and 10% of input availability of the fuel respectively.
- The energy or BTE of the engine, on an average, is 11, 22 and 28% at the load of 20, 60 and 100%, respectively. While the exergy efficiency, on an average, is found to be 17, 36 and 38%, respectively.

- Blended fuels recorded a higher destructed availability and a lower shaft availability against the input fuel availability as compared to diesel. Exergy efficiency decreases with the increase in the content of VO in the VO-diesel blend.
- The addition of ethanol to VO30 blend shows a significant improvement in the exergy efficiency. However, with VO20 blend ethanol does not show a significant improvement in the exergy efficiency.
- The addition of DEE to VO20 and VO30 blends improves the exergy efficiency

9.2 Application Potential of the Work

The objective of this study is to explore the applicability of VO derived from *Mesua ferrea* Linn seeds in a stationary CI engine. The study reveals the potential of VO to supplement diesel through blending. The blending of 10% *Mesua ferrea* Linn oil with diesel shows almost no deterioration in the engine. The blending up to 20% can be run the engine without any modification of engine or the fuel. This work can effectively be used in the rural sectors for irrigation and power generation. *Mesua ferrea* Linn trees can be generally grown abundantly in rural areas around the residential campus for their aesthetic look and excellent sun shade because of their thickly distributed leaves.

9.3 Future Scopes

In this work, *Mesua ferrea* Linn oil is blended with diesel, ethanol and DEE to use in the CI engine. However, for better and effective utilization of *Mesua ferrea* Linn oil in the engine, the following experimental investigations may be carried in future.

- *Preheating of oil blend*: The high viscosity of vegetable oil possess the main hindrance for blending higher amount of oil with the diesel. To overcome this limitation, the preheating of oil blend before injecting it to the engine cylinder may be adopted for vegetable oil-diesel blend with greater fraction of vegetable oil in the blend. The preheating will reduce the viscosity of the oil blend. Moreover, as reported literature, the use of preheated oil is found to improve the performance of the engine in comparison the neat VO.
- *Engine durability test*: Throughout the experimental runs, the engine does not show any sign of abnormality with the *Mesua ferrea* Linn oil blend with diesel, ethanol and

DEE under all the loading conditions. However, it must be ensured that these fuels do not affect the engine during long run. Therefore, the durability test needs to be carried out to study the effect of these fuels on the diesel engine. This can be performed by running the engine continuously for a certain period and analyzing the wear and tear of the moving components and quality of lubricants before and after the running of the engine.

- *Injection timing and compression ratio*: Since the engine is designed and optimized to run with diesel as a fuel at CR = 17.5 and IT = 23° bTDC, it is likely that the same may not hold true for *Mesua ferrea* Linn oil blend. Therefore, the experimental investigations may be carried out at CRs by means of a variable CR engine, to optimize the CR. Similarly, the IT is also required to be optimized for obtaining the best performance of the engine. The IT can be optimized through retarding and advancing the fuel injection.
- *Test under overload condition*: In this work, the experiments have been conducted at the various engine loads of 20, 40, 60, 80, and 100% where an increase in BTE is observed with increasing load. While the BSFC decreases with the increasing load. The maximum BTE and minimum BSFC are found at 100% load. However, it is to be understood that if the engine load goes beyond the full load, how these parameters would vary. Thus, there is a need to test the engine under overload conditions to get the inflection point of the parameters like BTE, BSFC and others.

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Appendix A

Expressions for Performance and Combustion Analysis

I. Performance analysis

The following expressions are used to calculate the performance parameters for all the tested fuels such as diesel, VO-diesel blend and VO-diesel-additive blends.

1. Brake Power (BP)

It is the power available at the crankshaft of the engine expressed as

$$BP = \frac{2 \times \pi \times N \times w \times R}{60000}, \text{ kW} \qquad \text{.....A1}$$

where N, w and R are the speed of engine (rpm), engine load (kg.m/s^2) and dynamometer arm radius (m) respectively.

2. Brake Thermal Efficiency (BTE)

It is the ratio brake power to the energy supplied by the fuel or input fuel energy which is expressed as

$$BTE = \frac{BP \times 3600}{\dot{m}_f \times LHV_f} \times 100, \% \qquad \text{.....A2}$$

where $\dot{m}_f$ and LHV_f are the mass flow rate (kg/s) and lower heating value (kJ/kg) of liquid fuel respectively. The mass flow rate of fuel is determined using the following equation.

$$\dot{m}_f = \frac{\rho_f \times V}{\tau \times 10^6}, \text{ kg/s} \qquad \text{.....A3}$$

where ρ_f and V are the density and volume of fuel (ml) consumed in τ time (second).

3. Brake Specific Fuel Consumption (BSFC)

It is the amount of fuel consumed for each unit of brake power developed per hour.

$$BSFC = \frac{\dot{m}_f \times 3600}{BP}, \text{ kg/kWh} \qquad \text{.....A4}$$

4. Air Flow Rate

It is the rate at which the air is inducted into the engine during the suction stroke. The air flow rate is determined using the following expression where C_d is coefficient of discharge, d is orifice diameter (m), g is acceleration due to gravity (9.81 m/s^2) and h is the manometer reading across orifice meter (m of water). ρ_w and ρ_a are the water and air density respectively (kg/m^3).

$$\dot{m}_a = C_d \times \frac{\pi}{4} \times d^2 \times \rho_a \times \sqrt{\frac{2gh \times \rho_w}{\rho_a}}\text{, kg/s} \qquad \text{.....A5}$$

5. Stoichiometric Reaction

The following equation were used for the calculations of the stoichiometric fuel-air or air-fuel ratio.

For diesel (Sahoo 2010; Sarkar and Saha 2018a)

$$C_{12}H_{26} + 18.5 \times (O_2 + 3.76 \times N_2) = 12 \times CO_2 + 13 \times H_2O + 18.5 \times (3.76 \times N_2) \qquad \text{.....A6}$$

For *Mesua ferrea* Linn oil

$$\begin{aligned} C_{17.71}H_{33.75}O_2 + 25.1475 \times (O_2 + 3.76 \times N_2) = 17.71 \times CO_2 + 16.875 \times H_2O \\ + 25.1475 \times (3.76 \times N_2) \end{aligned} \qquad \text{.....A7}$$

For ethanol

$$C_2H_5OH + 3 \times (O_2 + 3.76 \times N_2) = 2 \times CO_2 + 3 \times H_2O + 3 \times (3.76 \times N_2) \qquad \text{.....A8}$$

For diethyl ether

$$(C_2H_5)_2O + 6 \times (O_2 + 3.76 \times N_2) = 4 \times CO_2 + 5 \times H_2O + 6 \times (3.76 \times N_2) \qquad \text{.....A9}$$

The stoichiometric air-fuel ratio of diesel, *Mesua ferrea* Linn oil, ethanol and diethyl ether are 15, 12.5, 9 and 11.2 respectively.

II. Combustion Analysis

1. Net Heat Release Rate

Net heat release rate is the rate at which work is done on the piston plus the rate of change of sensible internal energy of the cylinder contents. It equals to the difference between the apparent gross heat release rate and the heat transfer rate to the walls. The instantaneous pressure and volume recorded during the experiments were used for determining net heat release rate. For each complete engine cycle, data acquisition device records 720 pressure and volume data which is at the rate of one pressure and volume data per one degree of rotation of crank. In this analysis, 20 cycles were considered for determination of heat release at particular load. The equation for the calculation of net heat release rate is on the basis of first law of thermodynamics and ideal gas law which is expressed as (Heywood 2011; Debnath *et al* 2014; Bora and Saha 2015; Bora and Saha 2016a; Sarkar and Saha 2018b).

$$\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}, \text{ J/°CA}$$ **.....A10**

where $dQ_n / d\theta$, θ, γ, P and V are net heat release rate, crank angle, ratio of specific heat, instantaneous cylinder pressure (N/m^2) and volume (m^3) respectively. The value of for γ is considered as 1.35 as reported in (Heywood 2011; Debnath *et al*. 2014; Bora and Saha 2015; Bora and Saha 2016).

2. Smoothing Pressure

It has been observed that on differentiating the raw pressure data, recorded by the data acquisition, shows a noisy trend with the successive values (Debnath *et al*. 2014; Bora and Saha 2015; Bora and Saha 2016; Sarkar and Saha 2018b). Therefore, the smoothing of pressure data was necessary for better result which has been done using the following smoothing algorithm for (2b+1) values (Stone 1992).

$$P_n = \frac{1}{b^2}[P_{n-(b-1)} + 2P_{n-(b-2)} + 3P_{n-(b-3)} + \ldots + bP_n + \ldots + 3P_{n+(b-3)} + 2P_{n+(b-2)} + P_{n+(b-1)}]$$ **.....A11**

The terms in the above equation are only evaluated when the part of the subscript is brackets is not negative. Considering b = 2 the smoothing equation reduces to

$$P_n = \frac{P_{n-1} + 2(P_n) + P_{n+1}}{4}$$ **.....A12**

This equation is used for smoothing pressure where *P* is the recorded instantaneous pressure.

3. Rate of Pressure Rise

The change in pressure per unit change in crank angle ($dP/d\theta$) was determined by using first-order finite difference equation with fourth-order accuracy (Debnath *et al*. 2014; Bora and Saha 2015; Bora and Saha 2016b; Stone 1992) which is expressed as

$$\frac{dP}{d\theta}=\frac{P_{n-2}-8(P_{n-1})+8(P_{n+1})-(\mathrm{P}_{n+2})}{12(\Delta\theta)} \quad \textbf{.....A13}$$

where p is the instantaneous pressure data and $\Delta\theta$ is the successive change in crank angle.

4. Ignition Delay (ID)

Ignition delay is the period between fuel injection and start of combustion of fuel. It was calculated similarly as one used by (Debnath *et al.* 2014; Bora and Saha 2015; Bora and Saha 2016) using the equation

$$ID=\theta_{soc}-\theta_{soi} \quad \textbf{.....A14}$$

where θ_{soc} and θ_{soi} are the crank angle at which combustion starts and fuel injected respectively. θ_{soc} is obtained from $dP/d\theta$ diagram as it changes its concavity up on the starts of the combustion and θ_{soi} is the standard injection timing specified by manufacturers which is 23° bTDC.

Appendix B

Experimental Uncertainties

The uncertainties in the measurement of the parameters is calculated based on the Kline and McClintock (1953) and Moffat (1985). To estimate the uncertainty in the calculated experimental result on the basis of the uncertainties in the primary measurements, the result N is a given function of the independent variables x_1, x_2, x_3, …, x_n. Hence,

$$N = N(x_1,\ x_2, x_3,\ .....,\ x_n) \qquad \textbf{.....B1}$$

If ΔN is the uncertainty in the result created due to the individual uncertainties of the independent variables ΔN_1, ΔN_2, ΔN_3, ..., ΔN_n. Then the uncertainty of the result can be written as

$$\Delta N = \left[\left(\frac{\partial N}{\partial x_1}\Delta N_1\right)^2 + \left(\frac{\partial N}{\partial x_2}\Delta N_2\right)^2 + \left(\frac{\partial N}{\partial x_3}\Delta N_3\right)^2 + \cdots + \left(\frac{\partial N}{\partial x_n}\Delta N_n\right)^2\right]^{\frac{1}{2}} \qquad \textbf{.....B2}$$

The calculated uncertainties of the measured independent parameters are presented in Table B1. The uncertainties of the dependent performance parameters are shown in Table B2. The calculation of the some of the dependent performance parameters are illustrated below.

Brake Power

From the Equation A1, it can be concluded that brake power is the function of N, w and R where N and w are independent variable and R is constant. The uncertainties of N and w is 0.5%. Therefore, the uncertainty of brake power (ΔBP) is

$$\Delta BP = \left[(0.005)^2 + (0.005)^2\right]^{\frac{1}{2}} = 0.007071 \approx 0.7\%$$

Brake Specific Fuel Consumption

From the Equation A4, brake specific fuel consumption is the function of BP and $\dot{m}_f$ which having uncertainties of 0.7 and 1% respectively. Therefore, the uncertainty of brake specific fuel consumption ($\Delta BSFC$) is

$$\Delta BSFC = \left[(0.007)^2 + (0.01)^2 \right]^{\frac{1}{2}} = 0.0122 \approx 1.2\%$$

Brake Thermal Efficiency

From the Equation A2, brake thermal efficiency is the function of *BP*, $\dot{m}_f$ and LHV_f having uncertainties of 0.7, 1 and 1% respectively. Therefore, the uncertainty of brake thermal efficiency (ΔBTE) is

$$\Delta BTE = \left[(0.007)^2 + (0.01)^2 + (0.01)^2 \right]^{\frac{1}{2}} = 0.0158 \approx 1.6\%$$

Similarly the uncertainties other performance parameters were calculated and presented in Table B2.

Table B1 Uncertainties of independent variables

Sl. No.	Independent variable	Relative error (%)
1	Engine speed	0.5
2	Engine load	0.5
3	Fuel flow rate	1.0
4	Calorific value of fuel	1.0
5	Height of manometer column	1.0

Table B2 Uncertainties of performance parameters

Sl. No.	Performance Parameters	Uncertainty (%)
1	Brake power	0.7
2	Brake specific fuel consumption	1.2
3	Brake thermal efficiency	1.6
4	Mass flow rate of air	1.0
5	Air fuel ratio	1.4

Appendix C

Expressions for Energy and Exergy Analyses

The energy analysis includes amount of fuel energy supplied in unit time (Q_{in}), energy converted to shaft power or shaft energy per unit time (Q_S), energy transferred to the cooling water per unit time (Q_W), energy loss through the exhaust per unit time (Q_E) and uncounted energy losses per unit time (Q_U). While the exergy analysis includes input availability of the fuel (A_{in}), shaft availability (A_S), availability input converted to cooling water availability (A_W), availability transferred exhaust or exhaust gas availability (A_E), availability destroyed (A_D) and exergy efficiency (η_{II}). The expressions for the evaluation of the components of the energy and exergy analysis is based on the one given by Flynn *et al.* (1984); Al-Najem and Diab (1992); Debnath *et al*. (2012); Debnath (2013) and Bora (2015).

I. Energy Analysis

1. Fuel Energy supplied per unit time

$$Q_{in} = \dot{m}_f \times LHV_f \text{, kW} \qquad \textbf{.....C1}$$

where $\dot{m}_f$ and LHV_f are the mass flow rate (kg/s) and lower heating value (kJ/kg) of liquid fuel respectively.

2. Shaft power or shaft energy per unit time

$$Q_S = \text{Brake power of the engine} = BP \text{, kW} \qquad \textbf{.....C2}$$

3. Energy transferred to the cooling water per unit time

$$Q_W = \dot{m}_w \times C_{pw} \times (T_2 - T_1) \text{, kW} \qquad \textbf{.....C3}$$

where $\dot{m}_w$ is the mass flow rate of cooling water (kg/s) passing through engine jacket and C_{pw} is the specific heat of water (kJ/kg-K). T_2 and T_1 are the outlet and inlet temperature of cooling water passing through engine jacket respectively.

4. Energy in the exhaust per unit time energy loss through the exhaust per unit time

$$Q_E = [(\dot{m}_a + \dot{m}_f) \times C_{pe} \times (T_5 - T_{amb})], \text{ kW} \qquad \text{.....C4}$$

where $\dot{m}_a$ and $\dot{m}_f$ are the mass flow rate of air and fuel (kg/s) respectively. T_5 is the exhaust gas temperature from engine and T_{amb} is ambient temperature. C_{pe} is the specific heat of the exhaust gas which is evaluated from the energy balance of the exhaust gas calorimeter using the following equation.

$$C_{pe} = \frac{\dot{m}_{wc} \times C_{pw} \times (T_4 - T_3)}{(\dot{m}_a + \dot{m}_f) \times (T_5 - T_6)}, \ \frac{\text{kJ}}{\text{kgK}} \qquad \text{.....C5}$$

where $\dot{m}_{wc}$ is the mass flow rate of cooling water (kg/s) passing through calorimeter, T_4 and T_3 are the outlet and inlet temperature of the cooling water passing through the calorimeter respectively. T_5 and T_6 are the inlet and outlet temperatures of exhaust gas passing through calorimeter respectively.

5. Uncounted energy losses per unit time

$$Q_U = [Q_{in} - (Q_S + Q_W + Q_E)], \text{ kW} \qquad \text{.....C6}$$

II. Exergy Analysis

1. Input availability of the fuel

$$A_{in} = \frac{1.0338 \times \dot{m}_f \times LHV_f}{3600}, \text{ kW} \qquad \text{.....C7}$$

2. Shaft availability

$$A_S = \text{Brake power of the engine} = BP, \text{ kW} \qquad \text{.....C8}$$

3. Cooling water availability

$$A_C = Q_C - \dot{m}_w \times C_{pw} \times T_{amb} \times \ln\frac{T_2}{T_1}, \text{ kW} \qquad \text{.....C9}$$

4. Exhaust gas availability

$$A_E = Q_E + [(\dot{m}_a + \dot{m}_f) \times T_{amb} \times \{C_{pe} \times \ln \frac{T_{amb}}{T_5} - R_e \times \ln \frac{P_{amb}}{P_e}\}]\text{, kW} \qquad \textbf{.....C10}$$

where P_{amb} is ambient gas pressure, P_e is the exhaust gas pressure out from engine and R_e is the specific gas constant of the exhaust gas (kJ/kgK). It is calculated from the thermodynamic relation expressed as

$$R_e = \frac{\text{Universal Gas Constant}}{\text{Molecular Weight of Combustion Products}}\text{, kJ/kgK} \qquad \textbf{.....C11}$$

The molecular weight of the combustion products is calculated considering complete combustion.

5. Destructed availability

$$A_D = [A_{in} - (A_S + A_C + A_E)]\text{, kW} \qquad \textbf{.....C12}$$

6. Exergy efficiency

$$\eta_{II} = [1 - \frac{A_D}{A_{in}}] \times 100\text{, \%} \qquad \textbf{.....C13}$$

List of Publications

Book Chapter

1. Sarkar A, **Dabi M**, and Saha UK, (2018) Supplementing the energy need of diesel engines in Indian transport and power sectors, in: Gautam, A., De, S., Dhar, A., Gupta, J.G., Pandey, A. (Eds.), Sustainable Energy and Transportation: Technologies and Policy, *Springer*, pp. 61–86.

Journals

2. **Dabi M**, and Saha UK, (2019) Application potential of vegetable oils as alternative to diesel fuels in compression ignition engines: A review, *Journal of the Energy Institute,* Vol. 92, pp. 1710 – 1726.
3. **Dabi M**, and Saha UK, (2020), Implications of blended Mesua Ferrea Linn oil on performance, combustion and emissions of compression ignition diesel engines, *Thermal Science and Engineering Progress,* Vol. 19, 100579.
4. **Dabi M**, and Saha UK, (2020), Influence of ethanol on the performance, combustion and emission characteristics of a stationary diesel engine run on diesel-Mesua ferrea Linn oil blend, *Sadhana - Academy Proceedings in Engineering Science, India,* Vol. 45, pp. 285.

Conferences

5. **Dabi M**, and Saha UK, (2018) Performance and emission characteristics of a compression ignition engine run on vegetable oil-diesel blends and diethyl ether, *7th International and 45th National Fluid Mechanics and Fluid Power Conference (FMFP-2018)*, December 10 -12, IIT Bombay, Mumbai.

Under Review

6. **Dabi M**, and Saha UK, (2020), Effect of ethanol additive on the behavior of a single-cylinder compression ignition engine fueled by *Mesua ferrea* Linn vegetable oil-diesel blend, *Applied Thermal Engineering*.
7. **Dabi M**, Sahoo BB, and Saha UK, (2020), Increase of efficiency and reduction of CO and nox emissions in a stationary compression ignition engine run on *Mesua ferrea* linn oil-diesel and diethyl ether, *Thermal Science and Engineering Progress.*
8. **Dabi M**, Sahoo BB and Saha UK, (2020), Improvement of performance and emission characteristics in a compression ignition engine powered by *Mesua ferrea* linn oil–diesel and diethyl ether, *Sadhana - Academy Proceedings in Engineering Science, India*.
9. **Dabi M**, and Saha UK, (2020), Thermodynamic performance of a compression ignition diesel engine powered by *Mesua Ferrea* Linn oil blends, *Energy*.