

EVALUATION OF ENGINE CHARACTERISTICS FOR A VARIABLE COMPRESSION IGNITION ENGINE FUELED WITH PREHEATED BIODIESEL BLENDS

A Thesis

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Dedicated to my Parents.....

whose endless faith and blessings always inspired me to move forward





Declaration

I hereby certify that the work compiled in this dissertation is the outcome of the research work, performed by myself, else stated, under the guidance of Professor Niranjan Sahoo.

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 contained in the thesis entitled Evaluation of Engine Characteristics for a Variabl Compression Ignition Engine Fueled with Preheated Biodiesel Blends, by Menelik Walle Mekonen (Registration No. 146103038), a student of 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 and this work has not been submitted elsewhere.

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The depletion of fossil fuel resources and environmental issues are encouraging the use of renewable fuels. Biodiesel is the most used renewable fuel in compression ignition (CI) engine, and became more and more attractive in the recent years due to its environmental advantages. They can be defined as “the mono alkyl esters of long chain fatty acids derived from lipid feed stock, such as vegetable oils, waste cooking oils and animal fats for its use in diesel engines”. They are produced through the reaction of a vegetable oil with methyl alcohol or ethyl alcohol in the presence of a catalyst. Chemically biodiesel is called a methyl ester if the alcohol used is methanol. If ethanol is used, it is called an ethyl ester. The majority of literature agrees that the use of biodiesel in a conventional diesel engine results in substantial reduction of unburned hydrocarbons (HC), carbon monoxide (CO) and particulate matter than that of diesel fuel. However, biodiesel emits higher nitrogen oxide (NO_x) when burnt in a diesel engine due to the oxygen content of the biodiesel. This induces a more complete and cleaner combustion process. By employing exhaust gas recirculation (EGR) in a diesel engine, it is possible to reduce/shrink the NO_x emissions. Also, biodiesel offered lower engine performance parameters compared to diesel fuel, mainly due to high density and viscosity of biodiesel, which leads to poor atomization, bigger droplets, longer jet spray penetration, and improper homogeneity in the charge and low volatilities in combustion process of a diesel engine. This reduces the overall efficiency of the engine. Its poor cold flow problems can adversely affect the performance of the fuel system as well as clog the fuel filter, fuel lines, and injector. Thus there is a need to modify the behavior of the fuel property of biodiesel.

The present work focused only on edible (palm oil methyl ester-POME) and non-edible oil (castor oil methyl ester-COME) as fuel for VCR DI compression ignition engine. There are relatively more studies on non-edible biodiesel for investigations of engine performance, combustion and emission characteristics of diesel engines using biodiesel and/ biodiesel–diesel blends. The past works revealed that uses of non-edible vegetable oils biodiesel for compression ignition engines in place of diesel were promising. However, edible oils, being consumable oil, it is not famous in the compression ignition engine application in India. Hence, author purposefully used POME in a diesel engine to indicate its opportunities for substituting diesel fuel in future diesel engine applications. Additionally, in India, there are no enough studies on POME in a diesel engine. Hence, the major emphasis has been given on POME as a potential competitive fuel with respect to diesel in this present study. The study includes performance parameters, combustion and emission characteristics analysis considering different operating and design conditions in a diesel engine.

The present investigations are planned after a thorough review of literature in this area. Experiments are carried out in a popular variable compression ratio (VCR) direct injection (DI) single cylinder, water cooled compression ignition engine. Major problems associated with biodiesels are higher viscosities and densities, lower heating values, which leads to poor fuel spray characteristics and mixing rate, improper homogeneity in charge and fuel atomization in combustion process of a diesel engine. Many efforts have been done and introduced the possible methods of reducing the viscosity and density of biodiesel, and improvement of engine characteristics running with biodiesel. These are fuel modifications like blending, pyrolysis and micro emulsion and engine modifications like preheating, dual fueling, modification of operating parameters such as CR, IP, IT. The author has focused on utilization of two methyl ester biodiesels (POME and COME) in a VCR DI compression ignition engines. In this study, the author attempted to improve the fuel properties of biodiesels by using blending, preheating, preheating biodiesel and blending with diesel techniques. Blending is the most convenient method for the formulation biodiesel property to reduce viscosity and density as a fuel in a diesel engine, at the cost of reduction in calorific value. In this regard, the POME/COME biodiesel is blended to diesel in different volume percentages to improve certain properties. This would help in having a good understanding of the dependence of the diesel properties on the biodiesel proportion. In this analysis, the fuel properties of the blends are determined and compared to blended biodiesel standards (ASTM D7467 and BIS) and diesel fuel. Moreover, some of mixtures of the blends were tested in a diesel engine to investigate the effect of blend ratios on engine performance and emissions parameters. Then, the preheating is applied to reduce the viscosity and density of inlet fuel and is implemented in a diesel engine. In this analysis, the author attempts to use a helical coiled heat exchanger for preheating of biodiesel. While executing each test, the neat biodiesel was heated to different fuel inlet temperatures (54–138 °C) in 12 °C increments before entering into combustion chamber to enhances better fuel injection and there by better fuel atomization. The aim of this study is to investigate the effects of preheating on fuel properties of biodiesel, and performance and emissions of diesel engine under full load conditions. Here also, the fuel properties of the preheated biodiesel (POME/COME) at different fuel inlet temperatures were determined and compared to biodiesel standards (ASTM 6751, EN14214 and IS15607) and diesel fuel.

Keeping in view these facts, combined preheating and blending can be a viable option to enhance important fuel properties biodiesel and to increase the fraction of POME biodiesel in blends used in a CI engine. Thus, in this work, experiments were designed to study the effects of reducing POME biodiesel viscosity and density by blending along with preheating, thereby eliminating its effect on blend fuel properties, performance, combustion and emission characteristics of the engine. Experiments have been conducted to obtain performance, combustion and emission characteristics with various blend ratios of preheated POME biodiesel/diesel blends. Tests were executed at varied engine loads (0%–100%) with

20% increments. Then, similar experiments were studied under four EGR rates of 10–40% with 10% increments with an intention to reduce the high nitrogen oxides (NO_x) that were prevalent at full engine loads using these blends of fuel. The results are then compared with No-EGR rate (0% of EGR) operation.

In this experimental analysis, first the fuel properties of the preheated biodiesel/diesel blends were determined and compared to blended biodiesel standards (ASTM D7467 and BIS) and diesel fuel. This is followed by the investigation focused on the effect of preheating of intake air to improve the operating range of the engine in a different preheated biodiesel(114 °C)/diesel blends mode of combustion. This investigation mainly emphasizes on the enhancement of combustion efficiency and the overall performance of the engine with meaningful reductions of CO and HC emissions. In this regard, an intake air preheating attachment, a shell and tube counter flow type heat exchanger was used to transfer heat from exhaust gases to intake air. The parameters were evaluated in the engine at four elevated intake air temperatures at 33 °C (ambient air), 41 °C, 49 °C and 61 °C, respectively at 90% of engine load. Then, the preheated biodiesel/diesel blends are tested in a variable compression ratio (VCR) diesel engine for variation of operating parameters of engine (CR, IP and IT). Being a fuel of different origin, the standard design parameters of a diesel engine may not be suitable for preheated biodiesel(114 °C)/diesel blends. Thus engine operating parameters play a key role in tuning the engine conforming to the better performance and emission standards. Therefore, the initial part of the present investigation targets at finding the performance characteristics of preheated blends of biodiesel in a VCR DI diesel engine at various combinations of compression ratios (CR = 15, 16, 17.5 and 18), fuel injection pressure (IP = 188, 200, 212 and 224 bar) and injection timings (IT = 19°, 23°, 27° bTDC). During this study, the engine load is fixed at 90% (10.8 kg) and EGR rate (EGR30%) for each CR-IP-IT combination tested. Later on, a thermodynamic and thermo-economic potential study for the aforementioned test results were done. In recent years, many studies have been carried out on the effect of biodiesel and its blends on thermodynamic potentials. But in those studies, no attention has been paid to the thermo-economic analysis of engines while using biodiesel blends. For this purpose, various preheated blends of POME biodiesel with diesel fuel were tested in the VCR DI compression ignition engine. The experiments were carried out at three engine loads (low–10%, medium–50% and high–90%) for five blends. The combustion reactions were determined using experimental data and, finally, the thermo-economic analysis was performed for steady-state control volume of the engine by using and solving energy, exergy and economic balances.

From the experimentation, blending helped in pulling down the density, viscosity, fire point and flash point of the biodiesel. It is observed that 20% of neat POME mixed with 80% of diesel, whereas 10% of neat COME mixed with 90% of diesel is the best suited blend, without heating and without any modification of the engine. It meets blended biodiesel standards (ASTM D7467 and BIS). Methyl ester of POME biodiesel is the better performing fuel due to better performance and lower emissions compared to other chosen

COME. As expected, the preheating of the biodiesel caused a considerable decrease in its viscosity and density with increasing temperature, thus causing them to approach the value of diesel fuel. Thus as per the result, it is suggested that POME/COME biodiesel up to 114 °C can be used as a substitute of diesel fuel without any significant modification at expense of increased NO_x emissions. However, for fuel inlet temperature above 114 °C, the performance was observed to be marginally inferior. Preheating biodiesel/diesel blends caused improvement in some properties such as kinematic viscosity, density, calorific value. However, flash point decrease as the percentage of diesel increases. Thus, the blended fuel viscosity and density meets the blended fuel standards requirements ASTM D7467 and BIS for up to 60% preheated blended ratio (PPBD60). Significant improvement in engine performance with reduction in CO and HC emissions is observed with preheated biodiesel//diesel blends mixtures compared to diesel fuel. From the engine test results it has been established that up to 60% of preheated POME (114 °C)/diesel blends (up to PPBD60) are optimum preheated mixtures and can be substituted for diesel. In addition, preheating the intake air there is a considerable decrease in CO, HC and with an increase NO_x emission. It was found that combined effect of intake air preheating (61 °C) and preheating biodiesel(114 °C) along with blending with diesel provided better result on engine performance (BSFC and BTHE), combustion parameters (PCP and ID) and there is a considerable decrease in CO, HC and with an increase NO_x emission than individual effect. Finally, the variation of engine operating parameters significantly improved engine characteristics fueled with different preheated biodiesel/diesel blends (PPBD20, PPBD40 and PPBD60) with intake air preheated at 61 °C under rated 90% of engine load with EGR30% rate operating conditions. The superior combinations of CR–IP–IT was found to be CR = 18, IP = 212 bar, IT = 27° bTDC for both preheated biodiesel/diesel blends and engine characteristics became at par with diesel fuel at this near optimal modified operating parameter. The results of economic analysis showed that all preheated biodiesel/diesel blends were more expensive than diesel fuel.

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PhD is although a small part of life; still it teaches us how to live the life in a large scale. It is equally a magnificent and a productions training. Apart from acquiring some knowledge about a tiny, part of scientific fraternity, PhD also teaches one, how to talk, to work in a group, to know people and also to make one understandable to others. In this regard, I have also interacted with a number of personalities, without the support of whom, I may not come this far.

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Nomenclature

Notations		Subscripts	
C_p	Specific Heat (kJ/kg.K)	a	Air
Ex	Exergy	amb	Ambient
g	Gravitational Acceleration (m/sec ²)	cw	Cooling water
H	Energy (kW)	$cwic$	Cooling water inlet to calorimeter
\dot{m}	Mass Flow Rate (kg/hr.)	$cwie$	Cooling water inlet to engine
P	Pressure (bar)	$cwoc$	Cooling water outlet from calorimeter
R	Gas Constant (kJ/kg.K)	$cwoe$	Cooling water outlet from engine
T	Temperature (°C)	e	Exist
T_0	Ambient Temperature (°C)	exh	Exhaust gas
T_1	Engine water inlet Temperature (°C)	exh,ic	Exhaust gas inlet to calorimeter
T_2	Engine water outlet Temperature (°C)	exh,oc	Exhaust gas outlet from calorimeter
T_3	Calorimeter water inlet Temperature (°C)	$engc$	Engine cost
T_4	Calorimeter water outlet Temperature (°C)	f	Fuel
T_5	Engine exhaust gas outlet Temperature (°C)	hr	hour
T_6	Calorimeter exhaust gas outlet Temperature (°C)	i	Inlet
V	Volume (cc)	o	Ambient condition
W	Dynamometer Load (kg)	omc	Operating maintenance cost
Z	Capital/Investment	th	Thermal
Greek Symbols		System Constants	
θ	Crank Angle (degree)	C_d	Coefficient of Discharge (0.61)
$^\circ$	Degree	C_{pw}	Specific Heat of Water (4.187 kJ/kg.K)
$\%$	Percentage	d	Orifice Diameter (0.02 m)
μ	Dynamic Viscosity, Ns/m ²	B	Cylinder Bore (0.0875 m)
ε	Chemical exergy factor	l	Engine Stroke Length (0.11 m)
ρ	Density, kg/m ³	k	Number of Cylinder
γ	Ratio of Specific Heats	r	Dynamometer Arm Radius (0.185 m)

λ	excess Air Ratio	n	Number of Revolutions per Cycle
s	Entropy	ρ_a	Density of air (1.174 kg/m ³)
Abbreviations			
ASTM	America Society of Testing Materials	IC	Internal Combustion
aTDC	After Top Dead Centre (degree)	ICE	Internal combustion engine
BD	Biodiesel	ID	Ignition Delay
BIS	Bureau of Indian Standards	IP	Injection Pressure (bar)
BP	Brake power (kW)	IS	Indian Standards
BSFC	Brake Specific Fuel Consumption (kg/kW-hr)	IT	Injection Timing (degree)
BSEC	Brake Specific energy Consumption (kJ/kW-S)	J	Joule
bTDC	Before Top Dead Centre (degree)	JOME	Jatropha Oil Methyl Ester
BTHE	Brake Thermal Efficiency (%)	K	Kelvin
C	Centigrade	kW	Kilo Watt
CA	Crank Angle (degree)	kJ	Kilo Joule
CBD	Castor Biodiesel	LHV	Lower Heating Value (kJ/kg)
Cc	Cubic Centimeter	lph	Liters per hour
COME	Castor Oil Methyl Ester	MHRR	Maximum Heat Release Rate (kJ/°CA)
CI	Compression Ignition	NHRR	Net Heat Release Rate (J/deg. CA)
CN	Cetane Number	min	Minute
CO	Carbon Monoxide (%)	NO	Nitric Oxide (ppm)
CO ₂	Carbon Dioxide (%)	NO _x	Oxides of Nitrogen (ppm)
CR	Compression Ratio	O ₂	Oxygen (%)
deg.	Degree	PBD	Palm Biodiesel
DI	Direct Injection	PCBD	Preheated castor biodiesel diesel
EGR	Exhaust Gas Recirculation	PCP	Peak Cylinder Pressure bar)
EGT	Exhaust Gas Temperature (°C)	PM	Particulate Matter
EN	European committee standardization	POME	Palm Oil Methyl Ester
FPT	Fuel Preheating Temperature (°C)	PPBD	Preheated Palm Biodiesel
HC	Hydrocarbon (ppm)	ppm	Part per million
HRR	Heat release rate (kJ/°CA)	SI	Sustainability index
IAPT	Intake Air Preheating Temperature (°C)	VCR	Variable Compression Ratio

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CHAPTER -1

Introduction

Overview

A vital segment of the growth of any civilization rests on the use of petroleum diesel fuels, in transport sectors. However, the current trend for use of petroleum fuels needs to be reduced. This is mainly because they have limited potential and produce serious ecological impact. One of the feasible solutions that has got popularity during last few decades is the use of the fatty acid methyl esters, termed chemically as biodiesels, as alternative fuels of diesel. While burnt in diesel engines, biodiesels offer drop of unburnt hydrocarbons (HC) and carbon monoxide emissions (CO), and no sulfurized emission as compared to diesel fuel. However, being oxidized fuels, biodiesels emit nitrogen oxide (NO_x) emissions when burnt in diesel engine, which is due to oxygen content of the biodiesel. The exhaust gas recirculation (EGR) can be an effective solution of this problem even if a minor effect of engine performance and emission parameters. However, it is wise to consider many technical problems with the use of biodiesel. The causes of these problems arising from the use of high density, high viscosity of biodiesel, which leads to poor atomization, bigger droplets, longer jet spray penetration, and low volatilities attains improper homogeneity in charge and fuel atomization in combustion process of a diesel engine. This reduces the overall efficiency of the engine. Its poor cold flow problems can adversely affect the performance of the fuel system as well as clog the fuel filter, fuel lines, and injector. Thus, there is a need to modify the behavior of the fuel property of biodiesel. Many efforts have been done and introduced the possible methods of reducing the viscosity and density of biodiesel are fuel modification and engine alteration of operating parameters such as CR, IP, IT). Therefore, it is necessary to study and optimize the engine characteristics of preheated biodiesel/diesel blends in a VCR-DI diesel engine at varied operating parameters (load, CR, fuel IP and IT). In order to perform this study, a non-edible vegetable origin of alternative fuels (biodiesels: POME & COME), are selected and tested at various CR, fuel IP and IT settings. This chapter briefly describes about alternative fuels, biodiesel, fuel modification technique, engine modification technique and emission control norms. Finally, the chapter concludes with the objectives of the present investigation and the layout of the thesis.

1.1 Preface

In today's world, diesel engine is widely used in the transportation sector and fuel prices are growing as the effect of its escalating demand and lower availability. The transportation sector among all is the second largest consumer of energy which is playing an important role in the projections of energy demand (Dublely, 2013). The annual energy outlook (2016) has reported that fuel consumption of heavy and medium duty vehicles in the transportation sector is 28% of total energy consumption and 60% of the total liquid fuel consumption in 2015 which will increase to 72% in 2035 (Outlook, 2016). More than half of increase attributed to non-OECD Asia (including China and India) of the two most important developing countries in the world, where strong economic growth drives increasing demand for energy. The energy consumption in non-OECD countries increases 41% between 2015 and 2040 in contrast to a 9% increase in OECD countries (**Figure 1.1**). The day is not far, when both the countries will alone consume almost half of the projected rise of world energy use. In the world consumption of liquid fuels rises from 95 million barrels per day (b/d) in 2015 to 113 million b/d in 2040. Most of the growth in world liquid fuels consumption from 2015 to 2040 comes from non-OECD countries, where strong economic and population growth increase the demand for liquid fuels by 39%. More than 80% of the total increase in liquid fuels consumption is in non-OECD (including China and India) experience rapid industrial growth and increased demand for transportation. India's use of liquid fuels for transportation is projected to increase by 142% from 2015 to 2040 (Conti et al., 2016). The petroleum and other liquid fuels are the largest energy source world-wide which presently counts at almost 35% (Book, 2012).

Diesel fuel is highly viable because it can provide good capacity along with fuel conversion efficiency to the engine. But, consumption of petroleum product produces harmful gases which are highly toxic and affects human life and eventually contribute to the global warming and other environmental issues (Cardone et al., 2002; Lee et al., 2013). So, there is an emergent need to focus on two major issues for the large applicability of compression ignition (CI) engine. One is to search cost effective fuel and other is efficient alternative fuel. Many researchers had introduced a variety alternative to fossil fuels (Permpool and Gheewala, 2017), however most of them affects the efficiency of engine and adversely affected the life of moving engine parts. Along with this, alternate fuels are even costly than the fossil fuels. So, it is necessary to find out an alternative

which is not only cost effective but also yields more or same energy as petroleum products and have low emission profile.

Diesel engine emits regulated emissions (CO_2 , NO_x , HC, CO), and it is produced due to incomplete combustion of fuel because of less oxygen presence in the diesel while NO_x is emitted due to higher combustion temperature (Man et al., 2016). Many organizations and agencies were established all over the world and worked to hold off the pollution and climate change in terms of reduction in emission levels along with an increase in its efficiency. The automobile industries are facing large pressure to meet the CI engine emissions within the approved level. Modification of engine parameters, recirculation of exhaust, adding lubricants, additives, alternative fuels etc. were tried to use for an increase in the efficiency in CI engine. Using oxygenated fuels, biodiesel (BD) is an alternative to use in CI engine because it has some valuable properties like higher oxygen contents; fewer sulphur contents etc.,(Chauhan et al., 2016). Many types of methyl ester of vegetable oil biodiesel and its blends with diesel have been used as fuel in CI engine with and without modification in engine.

Several challenges like higher viscosity, low calorific value, carbon deposition and higher NO_x emission, etc were faced by using biodiesel and its blends used in place of diesel (Patel and Sankhavara, 2017). Some excellent reviews on the performance, combustion, emission and tribological characteristics of biodiesel and its blends has been carried out by various investigators (Fazal et al., 2011). The biodiesel has a higher viscosity than diesel, which leads to the poor atomization of air-fuel mixture by which soot particles and carbon monoxide emissions increases. Also for the same rotation of crank shaft, more amount of fuel entered because of high density of biodiesel thereby it increases the BSFC. As biodiesel has higher oxygen content which helps in better combustion in the combustion chamber due to which increase in the temperature of cylinder takes place and availability of oxygen increases the NO_x emissions (Singh et al., 2016). Diesel fuel properties depend upon the number of hydrocarbon present in the mixture. Due to the higher oxygen atom present in the biodiesel, it is accepted as oxidized hydrocarbon. Biodiesel has low calorific value than diesel fuel; so for obtaining same power, biodiesel consumption is more in comparison with diesel fuel. Many researchers have reported that the reduction in toxic emissions for biodiesel along with alike performance with diesel fuel (Shameer et al., 2017).

Fundamentally, there are two options to improve the efficiency and reduce emissions in a diesel engine. First is the enhancement in mechanical and indicated efficiency by reducing the losses

such as heat loss from the cylinder, wall, cooling water and friction losses, and other is to use an alternative compatible fuel in the engine (Abedin et al., 2016). Biodiesel is having different but nearly same chemical and physical properties as compared to diesel so the engine parameters i.e., compression ratio (CR), fuel injection timing (IT), fuel injection pressure (IP) etc. may not be optimal for the usage of biodiesel and its blends because the engine had manufactured for diesel fuel (Dhar and Agarwal, 2014). Fuel IP and CR mainly affect the exhaust emissions and combustion rate while fuel IT effects mainly particulate emissions. With varying fuel IP and CR, many researchers were found enhanced results for performance parameters in the engine (Celikten et al., 2010). Advanced and retarded fuel IT also affects the performance of diesel engine. Emissions also deteriorated at these modified parameters. So, they have a significant effect on the performance of CI engine (Qi et al., 2011).

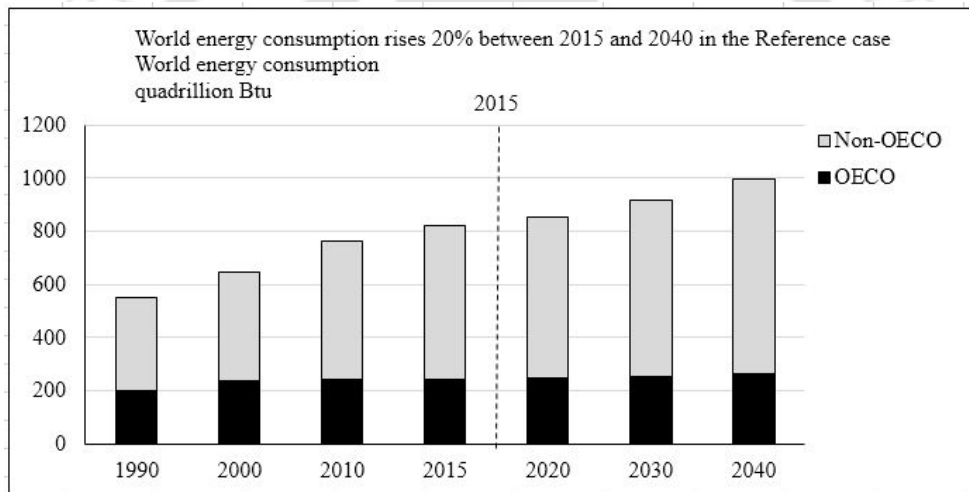


Figure 1.1: International Energy Outlook 2017.

1.2 Alternative Fuels

The primary expectations from the alternative fuels are generation of power with economical fuel consumption and minor emission of pollutants. Alongside they should be safe enough to handle and store for both stationary and mobile application. An alternative fuel turns out to be pretty attractive to use, when it provides provision for local production and easy distribution through marketing networks. Finally, an alternative fuel is acceptable when any combination of engine technology and fuel meets the prescribed vehicular emission norms as far as environmental prospective is concerned (Thipse, 2010).

Alternative fuel also known as non-conventional or advanced fuel can be defined as the fuel that can be utilized in the internal combustion (IC) engines with the exception of diesel or petrol. Principally, the alternative fuels are classified into three categories. The first type takes account of petroleum fuels with non-petroleum additives namely ethers. The synthetic liquids that comprise properties comparable to that of traditional petroleum fuels but found subsequent to treating of gaseous, solid or liquid fuels falls into second type. The last category non-petroleum fuels, namely alcohol, bio-fuels, biogas, hydrogen etc., are obvious choices, (Krylov and Tonkonogov, 2005). In the midst of these the biofuels and biogas have an important role to be present-day scenario of energy crisis. This is because, biofuels, types of liquid alternative fuel, can be produced locally, from renewable sources. This helps to generate employment, thereby supporting domestic economy. This is helpful to shrink down the huge drainage of national currency in the form of petroleum import.

Diesel fuel has been widely used in industry and in automobiles for over a century. As the petroleum prices continue to rise, the diesel supply is becoming scarce and unreliable. There is a growing concern for a cleaner environment; hence scientists have invested considerable effort in searching for renewable sources for substitutes of diesel fuel in a diesel engine. In 2005, studies carried out predicted that the present reserves of fuels used in internal combustion (IC) engines, including diesel, would be exhausted within 40 years if consumed at an increasing rate, which is estimated to be of the order of 3% per annum, (Nabi et al., 2009). In today's automotive industry improving engine performance, and the reduction of NO_x and PM emission, and ensure for long-term energy security is the most important task. National governments are imposing stringent emissions on automotive sector to reduce NO_x and PM emissions. Also global house warming gases (GHG) are going to receive more focus from future auto-sector. Hence, researchers are forced to find the substitute for the petroleum products as a new source of energy which is always recommended to replace with foreign exchange through the import of petroleum products. To improve the environment, decrease foreign imports and enhance the use of renewable fuels, biodiesel is considered an alternative fuel for IC Engines because of abundant availability and biodegradable, (Knothe, 2006).

The concept of using biodiesel, obtained from vegetable oil, as an alternative to diesel is not new. The “diesel engine”, developed by Dr. Rudolf Diesel in 1895 was intended to run on a variety of fuels. In fact, at the World Exhibition in Paris in 1900, his engine was run by ‘peanut

oil' as fuel. Since then, the diesel engine has been periodically modified to run on petroleum based fuel, more specifically diesel obtained from petroleum. Historically, it was the cheapest fuel available (Clean Alternative Fuels: Biodiesel, 2012). At present, the dual crisis of modern era, the depletion of fossil fuel and its environmental constraints mandated people to interchange or alternate the use of fossil diesel. Biodiesel is a substitute fuel that derived from edible and non-edible oils, and animal fats of bio-originated feedstocks through the transesterification process. Each fuel possesses its physicochemical properties that can be used to identify the fuel type. In general, Biodiesel refers to a non-petroleum-based diesel fuel consisting of short chain mono-alkyl (methyl or ethyl) esters originated from different sustainable bio-feedstocks, chemically produced using the transesterification process with an alcohol, usually methanol or ethanol as a chemical catalyst,(Adewale et al., 2015). **Figure 1.2** presents the biodiesel production process. The transesterification process is the most credible method in biodiesel production which requires an alkaline liquid catalyst, usually sodium or potassium methoxide for the chemical reaction between triglycerides and alcohol. Biodiesel and glycerol formation during transesterification process is the result of the chemical reaction between an alcohol and fatty acids, (Pragya et al., 2013).In general, most of the edible and nonedible oils from various bio-feedstocks are qualified to be transesterified into biodiesel fuel, (Bhuiya et al., 2016).

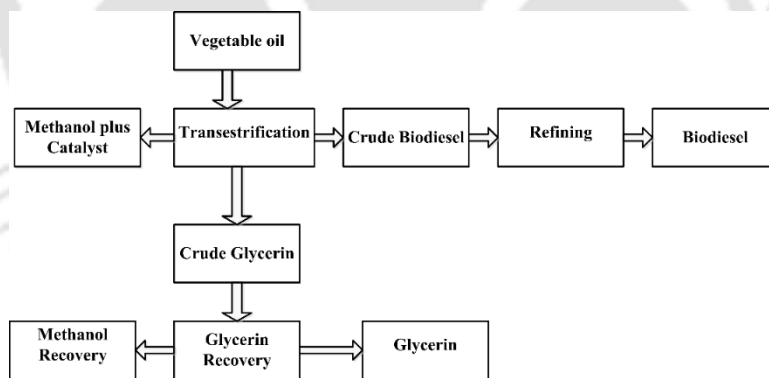


Figure 1.2: Flow Chart of Biodiesel Production (Adewale, Dumont et al. 2015).

Biodiesel is a safe alternative fuel to replace traditional petroleum diesel. It has high-lubricity, is a clean-burning fuel and can be a fuel component for use in existing, unmodified diesel engines. Biodiesel acts like petroleum diesel, but produces less air pollution, comes from renewable sources, is biodegradable and is safer for the environment. Producing biodiesel fuels can help create local economic revitalization and local environmental benefits. Biodiesel is infeasible

compared to conventional fuels as there is a large disproportion between edible oils and fuels. Therefore, non-edible oils come into play for the commercial production of biodiesel and significance research has been done increased to improve the performance characteristics considering available oils (Gui et al., 2008). Biodiesel, an environmental friendly diesel fuel similar to petro-diesel in combustion properties, has presently the most widely accepted alternative fuel for diesel engines due to its technical, environmental and strategic advantages over conventional diesel (Aworanti et al., 2013). It is a methyl or ethyl ester made from renewable biological resources such as vegetable oil and animal fats (Parawira, 2010). It is known as a carbon neutral fuel because the carbon present in the exhaust is originally fixed in the atmosphere. Moreover, different vegetable oils can be used for biodiesel production like soybean oil, Jatropha oil, rapeseed oil, palm seed oil, castor oil, sunflower oil, corn oil, peanut oil and cottonseed oil in addition to the possibility for use of waste cooking oil, and algal species (micro and macroalgae). Biodiesel is better than conventional diesel in terms of its availability, ignition quality, energy content, negligible sulfur content, higher flash point, reduced emissions and high cetane number (Moka et al., 2014). However, the viscosity and density of many biodiesels are found to be slightly more than that of mineral diesel, which is in general not preferred. The cold flow properties of biodiesel are also poor when compared to diesel (Isioma et al., 2013). The kinematic viscosity of methyl ester (biodiesel) is mostly higher than diesel fuel nearly a factor of two (Hoekman et al., 2012).

Biodiesel has different physical and chemical properties compared to petroleum based diesel fuel and it impossible to use this alternate fuel in the existing engine without any modification techniques (Nabi et al., 2006). It can affect the engine performance and emissions parameters (Çelikten et al., 2012). Viscosity is the property of fluid that offers resistance between one layer of fluid over its adjacent layer against relative motion between them (Demirbas, 2008). A fuel which is more viscous has poor atomization characteristics and narrow injection spray angle, incomplete combustion and injector clogging (Borhanipour et al., 2014). One with poor viscosity, leads to excessive wear and poor lubrication. Thus, it is desired for a fuel to possess optimum value of kinematic viscosity. It is therefore an important parameter to compare the atomization and spray characteristics of biodiesel and diesel. The minimum temperature at which a fuel produces vapors sufficient to ignite in the presence of an ignition source is known as the flash point. Therefore, it is important to understand flash point and fire point of a fuel as they

characterize its fire hazards and also its volatile nature. A highly viscous and less volatile fuel could result in misfire, ignition delay and thus poor cold starting. Because a fuel is metered in volume basis in a fuel injection system, a fuel with higher density results in a higher mass of it being injected (Anand et al., 2010). Thus, fuel density affects its consumption. The important fuel properties of biodiesel have to be modified to bring their combustion related properties closer to diesel, and improvement for utilization of biodiesel in the existing diesel engines without any engine modifications (Deepanraj et al., 2011). This fuel modification of biodiesel can be done when several methods are adopted in reducing their high density and viscosity including blending, pyrolysis, microemulsification (co-solvent blending), heating, dual fueling with gaseous and liquid fuels, and use of additives, etc. It is reported that direct implementation all of these methods to biodiesel are not fully promising and practical.

India is one of the largest petroleum consuming and importing countries. India imports about 70 % of its petroleum demands. The current yearly consumption of diesel oil in India is approximately 40 million tones constituting about 40% of the total petro-product consumption. Indian approach towards the development of biodiesel program is different than the other parts of world. The concepts of biodiesel development in India introduced by the former president of India, Dr. A.P.J. Abdul Kalam, which was centers mainly the cultivation and processing of non-edible plant seeds for production of biodiesel (Poonia and Jethoo, 2012). The drivers for three developments are, historic, functional, economic, environmental, moral and political. Initially development was focused on the jatropha (*jatropha curcas*) plant as the most suitable non-edible oilseed for biodiesel production. The market for biodiesel remains at an early stage in India with the country achieving a minimal blend rate with diesel of 0.001% in 2016. The development of biodiesel from *Jatropha* has met a number of agronomic and economic restraints and attention is now moving towards other feedstock's technologies which utilize used cooking oils, other unusable oil fractions, animal fat and non-edible feedstocks such as Karanja, Neem, Mahua, algae etc. The market for biodiesel is nascent and will continue to grow if there is a strong commercially viable strategy for building a sustainable biodiesel industry. Presently, India has five to six plants with capacity to produce 10,000 metric tons to 250,000 metric tons (MT) of biodiesel per year. Biodiesel is produced through multiple feedstock technology. India will produce upwards of 150 million liters of biodiesel in 2017 and will add another 10 million liters through 2018. The Producers utilize multiple feed-stocks such as 'used cooking oil, animal fats, tallow's and 'other

vegetable oils' such as palm, castor oils etc. to produce biodiesel and sell it locally after meeting prescribed BIS standards, thereby utilizing close to 30 percent of the installed capacity.

1.3 Fuel Modification Technique

Compression ignition engines are considered as prime movers in light, medium and heavy duty applications such as automobiles, power plants, marine and industrial sectors due to their reliable operation i.e., lower fuel consumption, and better power performance. The fossil fuels are very limited and may exhaust in coming few decades. In addition, the utilization of fossil fuels is increasing at faster rate due to the population growth and living advancements. Furthermore, the scientific community searches for alternative fuels those are renewable, safe and non-polluting. The renewable fuels such as vegetable oils of methyl ester biodiesel oils are an alternative to petroleum based fuels. However, the problems like high viscosity and density, and poor volatility of the biodiesel oils put obstacle on the end users. Fuel modification technique is being used by various researchers to gain specific fuel properties so as to improve performance and reduce the exhaust emissions of diesel engine. The impact of blending biodiesel with the diesel and preheating biodiesel and blended with diesel are one of the current scopes of research with regards to the fuel modification techniques. Intensive research is underway to separately to utilize the methyl ester of biodiesel judiciously without affecting our ecological environment. So far, much attention has been paid to biodiesel blend with diesel for many years. The majority of blends found in different countries are BD5– BD20. whereas, on preheating only few studies are available.

Initially, it is required to investigate the fuel properties of diesel, JOME, COME and POME, including the cetane number, density, kinematic viscosity, flashpoint, calorific value, cetane number, etc., and compared with international standards of ASTM D6751, EN14214 and IS, and then, characterize the fuel properties various blend ratios of biodiesel. Later, experimentation was carried out to examine the effect of various blend ratios of fuel run in a diesel engine for performance and emission parameters analysis to recommend the best the ranges of blend ratios of biodiesel that strongly offered better performance and emission parameters. Similarly, the effects of heating on the fuel properties of neat biodiesel and different blend ratios of biodiesel at different fuel inlet temperature was studied to utilized more percentage volume fraction of biodiesel as a fuel in a diesel engine with enhanced its engine performance and emission parameters.

1.3.1 Blending Technique

The method used to blend the fuel is the most important factor contributing to blend accuracy. The two major blending techniques used are splash blending and in-line (injection) blending. Currently, the most widely implemented technique is splash blending. This blending process involves adding biodiesel to a fuel storage tank that is partially filled with diesel fuel. The blending occurs as the magnetic stir rotates and the fuel splashes around in the tank. Unfortunately, in many cases, the truck does not drive far enough for the two fuels to blend uniformly. In addition, environmental factors such as temperature and humidity can affect the speed at which the fuels blend. A second, more accurate blending method is in-line blending. This type of blending occurs at a fuel rack, where dedicated blending equipment delivers a metered amount of fuel into a waiting fuel line. With in-line blending, the correct ratio of biodiesel is metered with automated control valves into the diesel fuel before it is dispensed into a fuel injection pump. Since the resulting fuel is blended prior to entering the fuel tank, the mixing problem associated with splash blending is eliminated. Although in-line blending offers a more accurate blending method than splash blending, any mechanical system is subject to wear and/or failures. The need to test the biodiesel blend ratio after final mixing is necessary regardless of the blending method. An accurate method to determine the biodiesel blend is just as important as an accurate blending method. In this study in-line blending technique was applied.

Blending of biodiesel with mineral diesel is a widely used methods for enhancing the biodiesel property (such as kinematic viscosity and density) and overcoming the fuel cold flow problems. Biodiesel from different feedstock's can blend with mineral diesel and used as a fuel for diesel engine under the ASTM blended fuel standard (Atabani et al., 2012). Diesel engines can be fuelled directly or blended with various diesel proportions with biodiesel on some modifications or no modification at all (Abu-Hamdeh and Alnefaie, 2015). Biodiesel and diesel being advantageous and disadvantageous in their respective terms, blending them would yield a fuel with intermediate properties which may improve the combustion and emission characteristics of neat biodiesel and thus its usage as a fuel in a diesel engine. The measurement and evaluation of blended fuel property is an important indicator for the maximum blending ratio of biodiesel from different sources (feedstock) that can meet the fuel specification requirements. Hence, the determination of right blend ratio is the most important criteria for utilizing biodiesel in a diesel engine. Previous investigations were conducted to study the effect of different volume fractions of biodiesel ratios

in a blended fuel for enhancing density and viscosity the fuel mixture operated in a diesel engine (Lahane and Subramanian, 2015). The studies indicated that, the maximum blend ratio of biodiesel currently limited to blends of 20 % or less as commercial fuels for many countries in existing diesel engine operate without engine modification (Smith et al., 2010). This blend is approved as a fuel for the existing diesel engines at low blending ratio up to 20% biodiesel (B20) according to the ASTM D7467 fuel standard specification. Accordingly, it is used as a commercial fuel in many countries. Conversely, at high blending ratios problems related to fuel properties are worsening (Ali et al., 2016). Furthermore, the engine performance and emission parameters results also showed worsen with increasing biodiesel ratio beyond B20 in the blend, when the key properties of the used fuel (biodiesel fuels properties) differ from those of mineral diesel fuel, which means different combustion characteristics.

1.3.2 Preheating Technique

As an alternative fuel for diesel engines, biodiesels are the principal renewable and carbon neutral sources. The causes of technical problems arising from the use of various biodiesel are the high surface tension, high density, and the high viscosity. Transesterification is the processes generally performed in order to reduce the viscosity of biodiesel but still it is higher to that of the diesel. To increase the fraction of biodiesel in blends, it is required to reduce the viscosity by preheating. Fuel preheating technique offers the advantage of easy conversion of the normal diesel engine to work on heavy fuels. It needs no modifications in the engine. Engine with fuel preheating has indeed in principle superior characteristics to that of normal fuel operation (Nwafor, 2003). This technique is economical, feasible and real for improved fuel performance of biodiesel run as a fuel in a diesel engine, and engine overall efficacy without modification existing diesel engine or fuel injection system (Martin et al., 2017). The preheating of the biodiesel improves the injection characteristics by enhancing the fuel properties of biodiesel (surface tension, density, kinematic viscosity and poor flow characteristics). The preheating of biodiesel at different temperatures reduces the viscosity and surface tension which enhances better fuel injection and atomization, improves mixture formation, and it will influence the fuel-air mixing due to the changes of spray evaporation and consequently influence the combustion, performance and emissions of diesel engine. From the fuel properties, viscosity can affect fuel flow rate and cause poor fuel atomization during the combustion process (Khalid et al., 2017). The preheating of biodiesel results in complete combustion of the biodiesel or fuel that results in decreased in amount of carbon dioxide, carbon

monoxide and particulate exhaust emission is also complete combustion of biodiesel. The cleaner exhaust can be obtained while elevated temperature of the fuel increases NO_x emissions. Thereby, minimize the troubles due to poor fuel droplet formation and atomization that results in lots of carbon deposit formation on the valves and injector choking (Mustaffa et al., 2014b). Moreover, increasing fuel temperature or heating also will ease the problem of injection process because it results in a decrease of the arithmetic diameter of the fuel droplets due to the effect of surface tension and viscosity changes with temperature (Augustine et al., 2012). Thus, it gives better spray formation and combustion process.

1.3.3 Preheating and Blending Techniques

Biodiesel is an alternative fuel similar characteristic to diesel fuel. It can be produced from vegetable oil, animal fat and waste cooking oil. The reduction of fossil fuel causes the increasing using the biodiesel fuel. However, use of biodiesel fuel can affect on engine performance and exhausts emission. Biodiesel are not efficiently in cold weather and it is the biodiesel major problem. It influences the fuel spray characteristics during the combustion process. The aim of this study was to determine the effects of biodiesel to temperature and is carried out using of room temperature, 40°C and 60°C. It is required to recommend the biodiesel blending ratio and biodiesel temperature that optimizes the engine performance and lower exhaust emissions. There are three types of biodiesel oil is used to carry out this study (crude palm oil, waste cooking oil and jatropa oil). While there are 3 blending ratios that have been made towards biodiesel (5%, 10% and 15%) except for crude palm oil biodiesel for which there is additional another type of blending ratios provided for carrying out this study (20%). The study about biodiesel is important for investigation as it gives some solution for reduction of fossil fuel. Performance of engine and emissions exhausted from biodiesel fuels is a measure of the results by using the biodiesel in diesel engines. Research and development of biodiesel fuels and its blends are very important to study and investigate in reducing problem in diesel engine.

In this sense, research and focus on preheat biodiesel fuels on these three types of biodiesel sources i.e. POME and COME are very important to be performed in promising alternative to conventional diesel fuel in India and for further comprehensive improvements as well. Increased of load condition and preheated biodiesel blends temperature promotes more rapid engine performance but exhibit relatively small variations in emissions production (Amir et al., 2014). From the fuel properties, the use of biodiesel or its blends effects on fuel droplet formation,

vaporization and air fuel mixing process due to its higher viscosity (Ma and Hanna, 1999). These effects cause important engine failures such as fuel filter clogging, piston ring sticking, injector choking and carbon formation deposits (Jazair et al., 2011). High viscosity fuel also leads to high smoke, HC and CO emissions. The high viscosity and the major chemically bound oxygen component in the biodiesel fuel play as a key element in combustion process especially during the fuel-air premixing. Chemical and physical properties of biodiesel were determined using standard ASTM and American Oil Chemists Society (AOCS). For example, jatropha oil kinematic viscosity is high at 35.98 cSt compare to the mineral diesel at 2.44 cSt. Fuels with high viscosity tend to form larger droplets on injection which can cause poor combustion. While biodiesel is cleaner than standard diesel fuel in many other ways, it's still dirtier (more air polluting) than gasoline. Bio fuels in general "result in more atmospheric CO₂ pollutants than burning an energy equivalent amount of oil when considering the entire production and consumption cycle. Therefore, the major reduction of CO₂ emission should be achieved in road transportation (Shahid et al., 2014). Further studies on the effects of preheat biodiesel blends fuel from crude palm oil, jatropha oil and waste cooking oil on the performance and emissions characteristic was conducted. Preheat is one of the effective method to reduce the viscosity of biodiesel fuels and its blends and viscosity will gradually decrease as the temperature increase.

1.3.4 Intake Air Preheating

The intake temperature also plays a vital role in increasing of peak pressure and brake thermal efficiency. The heating of the intake air helps to decrease engine warm-up times, improving fuel economy and emissions. When intake temperature increases the ignition delay changes and causes a change in the occurrence of peak pressure. Hence, the optimum intake temperature that gives maximum performance is to be identified. An air preheater is positioned in communication with the inlet of the engine. Preheating ensures proper combustion of the fuel mixture and rise in inlet temperature. This higher inlet temperature leads to increase in efficiency, torque, horse power and evaporation of fuel. Air preheating decreases ignition delay and improves the combustion rate. The effect of preheated air on standard diesel fuel engine indicated a good result on emission control. Higher inlet air temperature causes lower ignition delay, which is responsible for lower NO_x formation. Uniform or better combustion is occurred due to pre-heating of inlet air, which also causes lower engine noise. Easy vaporization and better mixing of air and fuel occur due to warm up of inlet air, which causes lower CO emission. Lower temperature intake air leads to inadequate

final compression temperature, increase in emission delay, and longer time between the injection of the fuel to ignition, local over-enrichment, incomplete combustion and high pressure gradients due to abrupt mixture conversion in the cylinder. These factors lead to knocking of the engine, increase in emission of hydrocarbons in the exhaust leading to severe loading of the environment.

1.4 Engine Parameter Modification Techniques

Worldwide pressure to reduce automotive fuel consumption and CO, HC and CO₂ emissions are leading to the introduction of various new technologies for the compression ignition (CI) engine as it fights for market share with the diesel fuel. The performance, combustion and emission characteristics of diesel engines depends on various factors like compression ratio, fuel injection timing, fuel injection pressure, fuel quantity injected, shape of combustion chamber, position and size of injection nozzle geometry, fuel spray pattern, air swirl etc.

1.4.1 Variable Compression Ratio

So far, variable compression ratio (VCR) engines have not reached the market, despite patents and experiments dating back over decades. VCR technology could provide the key to enable exceptional efficiency at light loads without loss of full load performance. Variable compression ratio (VCR) technology has long been recognized as a method for improving the automobile engine performance, efficiency, fuel economy with reduced emission. The main feature of the VCR engine is to operate at different compression ratio, by changing the combustion chamber volume, depending on the vehicle performance needs. The concept of variable compression ratio (VCR) promises improved engine performance, efficiency and reduced emissions. The higher cylinder pressures and temperatures during the early part of combustion and small residual gas fraction owing to higher compression ratio give faster laminar flame speed. Therefore, the ignition delay period is shorter. As a result, at low loads, the greater the compression ratio, the shorter is the combustion time. Time loss is subsequently reduced. Therefore, it seems reasonable that fuel consumption rate is lower with high compression ratios at part load. The increase in compression ratio increases the power output and efficiency. This is due to higher pressure and complete combustion of fuel air mixture in the cylinder. The compression ratio may vary for different fuels, load conditions and speed. So it is challenge to find the optimum compression ratio which gives maximum power, efficiency, lower BSFC and emission. There is a maximum compression ratio limit beyond which the knock may occur (Crookes, 2006).

1.4.2 Fuel Injection Pressure

Fuel injection pressures (IPs) in diesel engine plays an important role for engine performance obtaining treatment of combustion. The fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization for better penetration of fuel in order to utilize the full air charge and to promote the evaporation in a very short time and to achieve higher combustion efficiency. When fuel IP is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. This situation leads to inefficient combustion in the engine and causes decrease the engine performance with increase in CO emissions since combustion process goes to a bad condition. When the IP is increased, the fuel particle diameters will become small. The mixing of fuel and air becomes better during ignition delay period which causes engine performance will increase with low smoke level and CO emission. But, if the IP is too high, ignition delay becomes shorter. So, possibilities of homogeneous mixing decrease and combustion efficiency falls down. Therefore, smoke is formed at exhaust of engine.

1.4.3 Fuel Injection Timing

In diesel engine, the fuel injection timing causes the variation of delay period, rate of evaporation of fuel, rate of combustion, etc. and subsequently it affects the engine performance, combustion and emission characteristics. Variation in injection timing has a strong effect on brake thermal efficiency, brake specific fuel consumption, smoke and NO_x emissions because of the change in maximum pressure and temperature in engine cylinder. Fuel injection timing essentially controls the crank angle at which combustion starts. While the state of the air into which the fuel is injected changes as injection timing is varied and thus ignition delay would vary, these effects are predictable. If injection starts earlier (advanced injection timing), initial air temperature and pressure are lower, the ignition delay would increase. If injection starts later (retarded injection timing), air temperature and pressure are slightly higher initially, and a decrease in ignition delay would occur. This means that injection timing variation has a strong effect on ignition delay, and consequently on engine combustion and emissions because of the changing maximum pressure and temperature in engine cylinder (Sayin et al., 2008).

1.5 Emission Control Norms

Diesel engines are widely used in a variety of applications, namely, power generation, transportation, agriculture, marine, military, telecommunication generator sets, etc. They are more

proficient and resilient than gasoline engines. However, they suffer awful nuisance due to their higher emission of wrongdoer pollutants like CO, HC, SO_x, NO_x and soot along with the traditional greenhouse gas CO₂. In order to trim down this hazardous emission, India has adopted stringent emission norms, which are also updated. The execution program of European Union emission standards in India is demonstrated in **Table 1.1**. Emission standards in Euro norms for Diesel Truck and Bus Engines are showed in **Table 1.2**.

Table 1.1: Indian emission standards for four wheelers (Sood 2012).

Standard	Reference	Date	Region
India 2000	Euro 1	2000	Nation wide
Bharat stage II	Euro 2	2001	NCR [*] , Mumbai, Kolkata, Chennai
		April, 2003	NCR [*] , 11 Cities [#]
		April, 2005	Nation wide
Bharat stage III	Euro 3	April, 2005	NCR [*] , 11 Cities [#]
		April, 2010	Nation wide
Bharat stage IV	Euro 4	April, 2010	NCR [*] , 11 Cities [#]
Bharat stage V	Euro 5	November 2015	NCR [*] , 11 Cities [#]
Bharat stage VI	Euro 6	Proposed 2020	NCR [*] , 11 Cities [#]

^{*} National Capital Region, NCR (Delhi), [#] Mumbai, Kolkata, Chennai, Bangalore, Hyderabad, Ahmedabad, Pune, Surat, Kanpur and Agra

Table 1.2: Emission standards for diesel truck and bus engines, g/kW.hr (Indian Emission Standards, 2012).

Stage	Reference	Year	CO	HC	NO _x	PM
Bharat stage-I	Euro 1	2000	4.5	1.1	8.0	0.36 ^a
Bharat stage-II	Euro 2	2005 ⁺	4.0	1.1	7.0	0.15
Bharat stage-III	Euro 3	2010 [#]	2.1–5.45	0.66–0.78	5.0	0.10–0.16
Bharat stage-IV	Euro 4	2010 [#]	1.5–4.0	0.46–0.55	3.5	0.02–0.03
Bharat stage-V	Euro 5	n/a ^b	1.5–4.0	0.46–0.55	2.0	0.02–0.03
Bharat stage-VI	Euro 6	2020 ^c	1.5–4.0	0.13–0.16	0.4–0.46	0.01

+ Earlier introduction in selected regions (Mumbai, Kolkata, Chennai, Bangalore, Hyderabad, Ahmedabad, Pune, Surat, Kanpur and Agra), [#] only in selected regions (Above cities plus Solapur and Lucknow), ^a 0.612 for engines below 85 Kw, ^b Initially proposed in 2015, ^c Proposed schedule and limits

1.5.1 Excess Air Supply

Oxygen enriched combustion is one of the attractive combustion technologies to control pollution and improve combustion in diesel engines. Heywood (1988) in his text clearly indicated that, ensuring complete combustion even with latest technologies, the engines must operate in excess air. That is, more air carrying 21% O₂ by volume is passed through the intake valve than the

chemically required (stoichiometric) and this process ensures that nearly all fuel molecules receive required oxygen for complete combustion. Excess air speeds up the mixing of fuel with air and ensures complete combustion of fuel but at the same time excess air wastes heat energy by carrying it in the exhaust gases. If sufficient oxygen is not provided to the engine during combustion process, complete conversion of carbon and hydrogen is impossible to attain and that leads to particulates and carbon monoxide resulting in increased exhaust emissions.

1.5.2 Exhaust Gas Recirculation

Exhaust Gas Recirculation (EGR) is a useful technique for reducing NO_x formation in the combustion chamber. Exhaust consists of CO₂, N₂ and moisture contents (mainly water vapours). EGR uses the exhaust gases recirculated through the intake manifold to engine combustion chamber as shown in **Figure 1.3**. When a part of this exhaust gas is re-circulated to the cylinder, it increases the CO₂ concentration in combustion chamber and it acts as diluent to the combusting mixture. This also reduces the oxygen concentration in the combustion chamber, increases the heat capacity (specific heat) of the intake charge. With increase in contents of exhaust gases in the combustion chamber, it affects mixture formation. Thus there is a reduction in the cylinder peak temperature rise and subsequently, the formation of NO_x reduces because of the lower combustion temperature. Increase in EGR ratio suppresses the rapid burning during premixed combustion and consequently reduces NO_x formation.

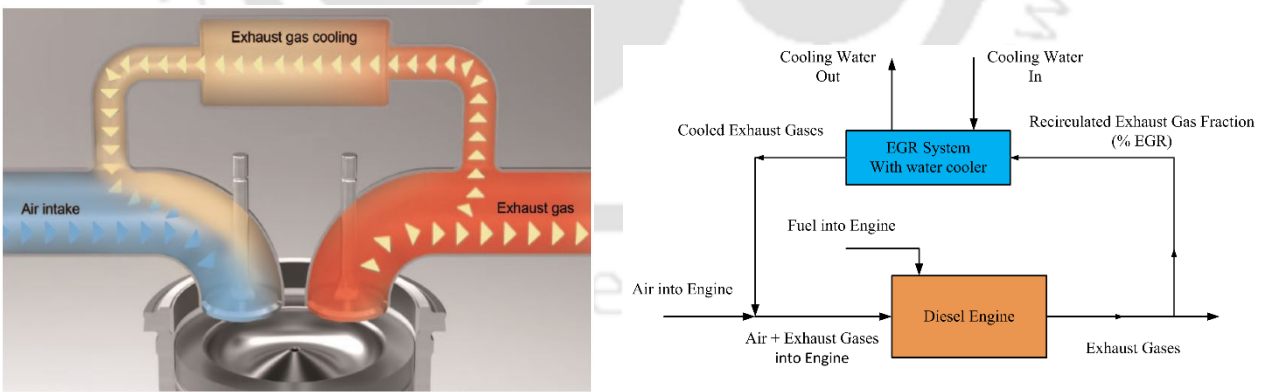


Figure 1.3: Exhaust gas recirculation system (Kech, Schmidt et al. 2011).

1.6 Objective of the Dissertation

In the present investigation, vegetable feed stocks, castor biodiesel and palm biodiesel, which are

a methyl Ester of castor oil (COME) and methyl ester of palm oil (POME), are selected to study in blending, preheating and preheating-blending form. COME and POME had recently gained global interest because of their greatest oil yield per unit land on earth. There are few studies available, regarding its fuel-related properties, most of which denote only part of the specifications. Among these studies, there is an agreement in regards to several of the physical properties of COME and POME biodiesels including, favorably high flash point and improved lubricity, so that biodiesel can be reliable as compared to diesel. Largely, they have extremely high kinematic viscosity and density in comparison to the International Standard Specifications (ASTM methods). This may result in improper spray characteristics. Thus, it limits its use and commercialization as an alternative fuel. However, the COME ($C_{18.25}H_{36.23}O_2$) and POME ($C_{18.07}H_{34.93}O_2$) have sizable amount of oxygen bonded in its molecular structure that makes it burn more intensely. The methyl ester from castor oil and palm oil biodiesel, have low engine emissions of carbon monoxide (CO) and unburnt hydrocarbon (HC), high oxidation stability apart from the high nitrogen oxide emission which is higher. The increase of NO_x emission, which can be reduced by implementation of exhaust gas recirculation system. A number of researchers have worked with COME and POME in various types of compression ignition engines. However, the focus with combined effects of biodiesel preheating and blending under intake air preheating mode as an alternative fuel to diesel is rare. Hence, there is a need to have a thorough and systematic study on preheated biodiesel/diesel blends in various engine settings in order to optimize its performance, combustion, and emission characteristics. These types of studies are very rare in the literature. In view of this, the present investigation is aimed at optimizing the compression ratio (CR), fuel injection pressure (IP) and the fuel injection timing (IT) of preheated POME biodiesel/blends run engine to achieve lower emission levels together with an improved or equivalent performance trends than a standard diesel engine. Besides, to locate and estimate the thermodynamics and thermo-economic parameters for the preheated POME biodiesel/diesel blends run diesel engine, an engine thermodynamic route, has also been considered. This study can play a significant role to understand the enhanced available energy management, of a diesel engine run by preheated and blended palm biodiesel. After extensive literature survey, *the theme of this thesis is framed to evaluate the optimum blend ratios, intake air temperature and engine operating parameters when the preheated blends of biodiesels are used in a CI engine.* In summary, the following set of studies, have been performed in order to arrive at the intended objectives:

- Development of an experimental set-up along with accessories and necessary instruments for a VCR engine.
- Design and fabrication of a novel helical-coiled heat exchanger for biodiesel preheating and integrating with baseline setup for proposed experiments.
- Design and fabrication of a novel shell and tube intake air heat exchanger, modifying the intake manifold system and integrating with baseline setup for proposed experiments.
- Baseline experiments in VCR engine using neat diesel.
- Preparation of various blends of biodiesel and characterize the test fuel properties of the biodiesel blends to determine the optimal blending composition.
- Investigate the performance and emission parameters of a diesel engine fueled with neat biodiesel (POME & COME), subjected to preheating at different temperatures in order to lower its viscosity and density and recommend for optimal fuel inlet temperature, which is enhances engine performance.
- Usage of preheated & blended test fuels in a diesel engine to investigate performance and emission characteristics and subsequently, incorporate the effect of EGR on reduction NOx emission.
- To study the combined effect of intake air preheating and preheated blends of biodiesel on performance, combustion and emission characteristics of CI engine.
- Investigation of performance, combustion and emission characteristics of preheated blends of biodiesel in a VCR-CI engine at varied CRs, fuel IPs and ITs.
- Thermodynamics and thermos-economic analysis of diesel engines while using various preheated blends of biodiesel as a fuel.

1.7 Organization of the Thesis

Chapter 1 introduces the importance and limitation of diesel fuel in a diesel engine, the use of alternative fuel specially biodiesel (castor oil and palm oil), fuel modification technique, engine performance and emission improvement methods, emission control norms. **Chapter 2** discuss about extensive literatures on blended biodiesel, preheated biodiesel, preheated and biodiesel–diesel blend, effect of exhaust gas recirculation on NOx emission reduction, effect of intake air preheating, effect of modified engine operating parameters, recent research progresses made in a compression ratio engines and the scopes of work. After extensive literature survey and

considering limitations/gaps, the objective of this dissertation was framed. **Chapter 3** refers to the experimentation (VCR engine setup and its different measurement and instrumentation devices), fuel supply system, engine conversion methodology, experimental design methodology, procedure along with uncertainty analysis. In the present investigation, an helical coiled heat exchanger and shell and tube heat exchanger was indigenously designed and fabricated locally at Indian institute of Technology Guwahati. The theme of this design is to use waste heat from engine exhaust and its effective utilization to preheat the inlet fuel and intake air, thus making it more sustainable without any external energy. The established experimental setup is planned for executing different experiments as per the specific objective requirements the study. **Chapter 4** highlights the experimental estimation of optimal ratios of POME and COME based on biodiesel blend with respect to fuel properties to study engine performance and emission parameters. In this study, the feasibility of biodiesel produced from palm oil and castor oil biodiesels were investigated. Characterization of blended fuel properties with increasing biodiesel ratios (by volume) were conducted to evaluate the engine performance and emission test results. The qualified blend fuel properties were used to indicate the maximum blending ratio suitable for use in the existing CI engine according to the blended biodiesel standard (ASTM D7467 and BIS). **Chapter 5** represents the effect of preheated biodiesel run engine on fuel properties and performance as well as emissions. The optimum fuel inlet temperature can boost the overall performance and slow up the emission associated with engine running with biodiesel partially or fully. **Chapter 6** discusses about the estimation of the best preferred fuel sample among the preheated (114°C) POME biodiesel blend samples by comparing their performance in terms of engine performance and emission parameter after thorough investigations of varying engine loads and EGR rates (10%, 20%, 30% and 40%) at full engine load. Preheating of optimized POME biodiesel blend results improvement in engine performance parameter (BSFC and BTHE), combustion characteristics (higher cylinder pressure and shorter ignition delay). It also gives reduced CO and HC emissions with significant increased NO_x emissions. All these parameters are expressed/measured in different units and their values are also widely apart. **Chapter 7** explains the effect of preheated biodiesel/diesel blends with intake air preheating mode run engine. In this study, the influences of intake air preheating on engine performance parameters and exhaust emissions were investigated. The parameters were evaluated in the engine at four elevated intake air temperatures at 33°C (engine at ambient air temperature), 41°C, 49°C and 61°C, respectively. **Chapter 8** emphasizes the engine analysis at

modified engine parameter (CR-IP-IT) operating conditions. The results of the experiments help to optimize the aforementioned design parameters to achieve optimum performance, combustion and emission characteristics of preheated and blended biodiesel blend ratios with preheating intake air at elevated temperature with diesel in the diesel engine. The modification in engine had been carried out by varying compression ratio (CR 15 to 18), fuel injection pressure (IP 188 to 224 bar) and fuel injection timing (IT 19 to 27°bTDC). Tests were performed with five different preheated (114°C) biodiesel/diesel blends at 90% engine load under EGR 30% rate. All the above parameters were evaluated in the engine at intake air temperatures at 61°C. **Chapter 9** emphasized about the thermodynamic and thermos-economic aspects of preheated biodiesel/diesel blends covering fundamental laws of thermodynamics. The combustion reactions were determined using experimental data and, finally, the thermo-economic analysis was performed for steady-state control volume of the engine through energy, exergy and economic balances. Finally, a concluding remarks with key findings of the present investigation with future research scopes have been summarized in **Chapter 10**.

Overview

The depletion of fossil fuels is one of the issues that pose a major threat to the future generation due to the escalating energy demands over the years and the limited availability of fossil fuel reserves. The burning of fossil fuels is also detrimental to the environment, being a major contributor of the greenhouse effect, global warming, air pollution, smog, and acid rain. Hence, much effort has been made to develop cleaner alternative fuels in order to fulfill the escalating energy demands while simultaneously reduce harmful exhaust emissions to the environment. Biodiesels are one of the alternative fuels to fulfill this need, and appealing substitutes for diesel because they can be used directly in diesel engines without modifications. However, its detrimental issues, namely, higher viscosity, and density, lower calorific value and higher nitrogen oxide emission have to be resolved. In this regard, this chapter is dedicated towards the detailed review of the literature performed in the fields of biodiesel blending, preheating, and preheating and blending diesel fuel in order to improve the physicochemical properties of the fuel, thus enhancing engine performance with reduced exhaust emissions of diesel engine. It is then, followed by a detailed analyses of intake air preheating for enhancing engine combustion parameters of diesel engine while run in a biodiesel fuel. A brief discussion on the effects of variable compression ratio (VCR) direct injection (DI) diesel engine are also revisited in view of the variation of some engine operating parameters namely, CR, fuel IP and fuel IT. The exhaust gas recirculation (EGR) systems of diesel engine section is focused on the key studies related to the use of exhaust emission treatment in order to reduce the NO_x exhaust emissions of diesel engines. Finally, the objectives drawn in the earlier chapter are justified by identifying the key scopes of works from the comprehensive review of respective literature.

2.1 Preface

In this chapter, a brief review is presented on the key studies pertaining to biodiesel, the engine performance, combustion and exhaust emission characteristics of diesel engines fueled with various types of biodiesel blends, preheated biodiesel, preheated biodiesel blend with diesel and intake air preheating of diesel engine. A critical reviews of the effects of CR, fuel IP and IT on the engine performance, combustion and exhaust emissions characteristics of diesel engines fueled with biodiesels and biodiesel blends is also given in the section. Finally, the key studies related to the use of exhaust gas recirculation (EGR) in order to reduce the exhaust NO_x emissions of diesel engines are also presented.

2.2 Biodiesel

Some excellent reviews on the performance, combustion, emission and property characteristics of biodiesel has been carried out by various investigators (Fazal et al., 2011; Singh et al., 2012). The biodiesel has a higher viscosity than diesel, which leads to the poor atomization of air-fuel mixture by which soot particles and carbon monoxide emissions increases. Also for the same power output, more amount of fuel entered because of high density of biodiesel thereby increases the BSFC. As biodiesel has higher oxygen content which helps in better combustion in the combustion chamber due to which increase in the temperature of cylinder takes place and availability of oxygen increase the NO_x emissions (Buyukkaya, 2010; Dhar et al., 2012; Khond and Kriplani, 2016; Panneerselvam et al., 2015; Singh et al., 2016). Diesel fuel properties depend upon the number of hydrocarbon present in the mixture while due to the higher oxygen atom present in the biodiesel, it is accepted as oxidized hydrocarbon. Biodiesel has low calorific value than diesel fuel; so for obtaining same power, biodiesel consumption is more in comparison with diesel fuel. Many researchers have reported that the reduction in toxic emissions for biodiesel along with alike performance with diesel fuel (Maghbouli et al., 2015; Rajasekar et al., 2010; Shameer et al., 2017). Fundamentally, there are two options to improve the efficiency and reduce emissions in a diesel engine. First is the mechanical and indicated efficiency by reducing the losses (like heat loss from the cylinder, wall, cooling water and friction losses) and other is to use an alternative compatible fuel in the engine (Abedin et al., 2016; Kumar and Chauhan, 2015; Kumar and Chauhan, 2013b; Tsolakis et al., 2007). Because of gifted properties of biodiesel, it shows advantageous results in terms of emission but negative impact on BTHE and BSFC as compared to diesel in CI engine.

So, it is imperative to increase the performance of biodiesel in CI engines by modifying the engine parameters i.e., compression ratio (CR), fuel injection timing (IT), fuel injection pressure (IP) etc. Since biodiesel is having higher kinematic viscosity and density, and lower calorific value properties as compared to diesel so the default engine parameters (CR, IP and IT) may not be optimal for the usage of neat biodiesel and its blends because the engine had manufactured for operating condition of diesel fuel (Dhar and Agarwal, 2014). Fuel IP and CR mainly affect the exhaust emissions and combustion rate while IT affects mainly particulate emissions. With varying IP and CR, many researchers were found enhanced results for performance parameters in the engine (Celikten et al., 2010; Gong et al., 2016). Advanced IT (Hountalas, 2000) or retarded IT (Qi et al., 2011) also affects the performance of diesel engine. Emissions also deteriorated at these modified parameters (Agarwal and Dhar, 2009; Flynn et al., 2000). So, they have a significant effect on the performance of CI engine.

In general, the present compression ignition engines, adapted for mineral diesel, cannot replicate diesel like performance run by 'biodiesel', because of the diesel unfamiliar properties. Thus, the existing engine can be considerably improving its performance, combustion and emission characteristics, and in turn will lead to better engine efficiencies with low environmental impacts while run in a biodiesel partially or fully. There are two possible methods, by which, diesel like performance or better than that may be achieved; firstly, by modifying the fuel itself; secondly, by modifying the fundamental operating characteristics of the engine, namely, CRs, fuel IPs and ITs, etc. In this regard, the remaining part of the literature survey is organized considering these modifying methods in diesel engine.

2.3 Fuel Modification Technique

2.3.1 Diesel engines fueled with Biodiesel Blends

Numerous works in the utilization of biodiesel as well as its blends in engines have been done. However, most of the literatures focused on single biodiesel and its blends. From previous studies, it is evident that single biodiesel offers acceptable engine performance and emissions for diesel engine operation. Lawrence et al. (2011) revealed that prickly poppy methyl ester (PPME) blended with diesel could be conveniently used as a diesel substitute in a diesel engine. The test showed increase in BTHE, BP and reduction of SFC for PPME and its blends with diesel. The most appropriate blend ratio (80% diesel/20% PPME by volume) produced maximum values of the

brake power, brake thermal efficiency and minimum values of the specific fuel consumption and also yielded minimum values of NO_x, CO and HC emission. Deepanraj et al. (2011) studied biodiesel (palm oil methyl ester) blends with diesel was investigated in a direct injection stationary diesel engine. Engine tests were conducted at constant speed using neat diesel fuel and various proportions of biodiesel blends. It is indicated that the lower blends of biodiesel increased the brake thermal efficiency and reduced the fuel consumption. Biodiesel blends produce lower engine emissions than diesel. From the result, it has been established that 20-40% of palm oil biodiesel can be used as a substitute for diesel without any engine modifications. Godiganur et al. (2009) studied the engine performance of mahua biodiesel in a six cylinder diesel engine. The volumetric blending ratios of biodiesel with conventional diesel fuel were set at 0, 20, 40, 60, and 100. The results indicate that with the increase of biodiesel in the blends CO, HC reduces significantly, fuel consumption and NO_x emission of biodiesel increases slightly compared with diesel. Brake specific energy consumption decreases and thermal efficiency of engine slightly increases when operating on 20% biodiesel than that operating on diesel. It is recommended that mahua biodiesel can be blended up to 20% without affecting the engine performance and emission. Mofijur et al. (2013), investigated Physico-chemical properties of Jatropha biodiesel and its blends with diesel followed by engine performance and emissions characteristics of blends. Compared to B0, the average reduction in brake power (BP) is 4.67% for B10 and 8.86% for B20. It was observed that brake specific fuel consumption (BSFC) increases as the percentage of biodiesel increase. Compared to B0, a reduction in hydrocarbon (HC) emission of 3.84% and 10.25% and carbon monoxide (CO) emission of 16% and 25% was reported using B10 and B20. However, the blends give higher nitrogen oxides (NO_x) emission of 3% and 6% using B10 and B20. As a conclusion, it is recommended that both B10 and B20 can be used in a diesel engine without any modifications. Chen et al. (2013) studied the fuel properties and combustion characteristics of Jatropha oil biodiesel blend and established a correlation between different fuel properties. As a result, a blending ratio of the JMEs with diesel was recommended up to 40 vol.% in comparison with the relevant specifications for biodiesel–diesel blends. The combustion tests of the JMEs–diesel blends were performed in a diesel generator. Higher brake thermal efficiency and lower brake specific fuel consumption were clearly observed with higher output loading. The concentration of carbon dioxide and nitrogen monoxide in the exhaust gas increased with higher output loading while the concentration of oxygen and carbon monoxide decreased. The

concentration of oxide of nitrogen (NO_x) decreased with the addition of pyrogallol. Sureshkumar et al. (2008) have studied the performance and emission characteristics of pongamia biodiesel and its blends with diesel in an unmodified diesel engine. Engine tests have been conducted to get the comparative measures of brake specific fuel consumption (BSFC), brake specific energy consumption (BSEC) and emissions such as CO, CO₂, HC, NO_x to evaluate the behaviour of PPME and diesel in varying proportions. The results reveal that blends of PPME with diesel up to 40% by volume (B40) provide better engine performance (BSFC and BSEC) and improved emission characteristics of low NO_x and HC emissions. Buyukkaya (2010) evaluated the performance and emission of diesel engine using neat rapeseed oil and its blends of 5%, 20% and 70%, and standard diesel fuel separately. The results indicate that the use of biodiesel produces lower smoke opacity (up to 60%), and higher brake specific fuel consumption (BSFC) (up to 11%) compared to diesel fuel. The measured CO emissions of B5 and B100 fuels were found to be 9% and 32% lower than that of the diesel fuel, respectively. The BSFC of biodiesel at the maximum torque and rated power conditions were found to be 8.5% and 8% higher than that of the diesel fuel, respectively. From the combustion analysis, it was found that ignition delay was shorter for neat rapeseed oil and its blends tested compared to that of standard diesel. The combustion characteristics of rapeseed oil and its diesel blends closely followed those of standard diesel. Dhar et al. (2012) characterized for its physical, chemical and thermal properties. Performance, emission and combustion characteristics of this biodiesel and its various blends with mineral diesel were compared with baseline data in a direct injection (DI) diesel engine. It is observed that the BSFC, BTHE and NO_x emissions for biodiesel and its blends are increased as compared with the diesel, whereas CO and HC emissions for biodiesel-fueled engine were lower. The detailed combustion characterization revealed that combustion starts earlier for higher biodiesel blends. However, start of combustion was slightly delayed for lower blends of biodiesel in comparison with mineral diesel. Rate of heat release for all biodiesel blends were almost identical to mineral diesel. Combustion duration for biodiesel blends was found to be shorter than mineral diesel.

2.3.2 Diesel engines fueled with Preheated Biodiesel

Some of researcher had successfully studied on direct use of biodiesel oil and its blends in a diesel engine for short term, the long-term endurance tests reported durability issue of the engine such as severe engine deposits, piston ring sticking, injector coking, gum formation and lubricating oil thickening (Agarwal and Agarwal, 2007). The biodiesel oils cannot be used directly without

bringing its properties closer to petroleum fuel as diesel. Mainly viscosity reduction is sufficient to improve its flow and atomization properties. The researchers were adopted different methods in a diesel engine to improve the performance biodiesel such as preheating, blending with diesel, blending with oxygenated additives and alcohols (Jain et al., 2011; Jeryraj Kumar et al., 2016; Rajagopal et al., 2015; Senthil et al., 2015a). Among them, preheating of neat biodiesel oil is one of the simple and cost effective methods to reduce the cold flow problems and improve the performance and emissions (CO and HC) with an increase of NO_x emission of an engine (Ibrahim et al., 2014; Mustaffa et al., 2014a; Venkata Ramanan and Yuvarajan, 2015). There have been some investigations on using preheated neat biodiesel in diesel engines. Nasim et al. (2013), performed experimental investigation on compression ignition engine powered by preheated neat Jatropha oil. In his study, the high viscosity of the Jatropha curcas oil was decreased by preheating. The effect of fuel inlet temperature on performance of diesel engine is evaluated and the results shows drop in BSFC with increase in fuel inlet temperature of Jatropha oil. Pradhan et al. (2014), worked on combustion and performance of a diesel engine with preheated Jatropha curcas oil using waste heat from exhaust gas. Helical coil heat exchanger is used to utilize the heat of exhaust gases for preheating CJO which reduces its density and viscosity. The BSFC and ignition delay period were decreased while BTE increased with increase in engine load. The fuel properties were improved by preheating and it can be used in the diesel engines without any modification as a substitute for diesel. Rahim et al. (2012), worked on influence of fuel temperature on a diesel engine performance operating with biodiesel blended. The effect of fuel temperatures on variation engine speed and their impact on the engine performance of a four-cylinder diesel engine has been investigated. In his result it can be found that the highest fuel temperature causes the highest injection pressure thus resulting in shorter ignition delay. The shorter ignition delay attributed to the early start of combustion thus leads to the higher in-cylinder pressure. The increase of fuel temperature represent the highest energy content thus resulted in lower BSFC as obviously desired. Kadu and Sarda (2010), investigate the use of preheated neat karanja oil in CI engine. The engine used for experiment was 4 stroke, single cylinder CI engine by preheated blend from 30-100 °C and speed between 1500-4000 rpm. Various parameters like BTHE, BSFC, emissions were compared. The result showed that SFC was higher compared to diesel for all loads. Hossain and Davies (2012), worked on the indirect injection CI engine which uses neat jatropha and karanj oil as fuel. Modifications were done on the cooling water circuit and fuel supply system such that

jacket water was preheated. BSFC increases 3%, CO₂& NO_x increases by 8%, as compared to diesel. From this it is concluded that the IDI compression ignition engine can be used with pure preheated Jatropha oil by jacket water. Chauhan et al. (2010), worked on performance enhancement of diesel engine by preheating technique while using Jatropha oil as a fuel. The aim is to decrease the viscosity of fuel using EGR system. Shell and tube type heat exchanger was used to preheat the fuel before entering to the engine which reduce the viscosity of blend. From results it was concluded that preheated blend having high BTHE and low BSEC at 80°C for optimum use of Jatropha oil. Karabektas et al. (2008), reported that the specific gravity and kinematic viscosity of the COME gradually decrease with the increase in the preheating temperature. Tests were carried out at full load conditions in a one-cylinder, four-stroke, direct injection diesel engine. Before supplied to the engine, COME was preheated to four different temperatures, namely 30, 60, 90 and 120 °C. It is seen that the kinematic viscosity is 6.54 cSt at 30 °C and decreases gradually to 1.26 cSt at 120 °C. Additionally, the specific gravity decreases from 0.882 at 30 °C to 0.851 at 120 °C. The results revealed that preheating COME up to 90 °C leads to favorable effects on the BTE and CO emissions but causes higher NO_x emissions. Moreover, the brake power increases slightly with the preheating temperature up to 90 °C. When the COME is preheated to 120 °C, a considerable decrease in the brake power was observed due to the excessive fuel leakage caused by drop in fuel viscosity. The results suggest that COME preheated up to 90 °C can be used as a substitute for diesel fuel without any significant modification in expense of increased NO_x emissions. Chauhan et al. (2010), worked on performance enhancement of diesel engine by preheating technique while using Jatropha oil as a fuel. The aim is to decrease the viscosity of fuel using EGR system. They found that the BTE (brake thermal efficiency) of engine was lower and BSEC (brake specific energy consumption) was higher when the engine was fueled with Jatropha oil as compared to diesel fuel. Increase in fuel inlet temperature resulted in increase of BTE and reduction in BSEC. Emissions of NO_x from Jatropha oil during the experimental range were lower than diesel fuel and it increases with increase in fuel inlet temperature. The CO (carbon monoxide), HC (hydrocarbon), CO₂ (carbon dioxide) emissions from Jatropha oil were found higher than diesel fuel. Kadu and Sarda (2010), investigated the use of preheated neat karanja oil in CI engine. The engine used for experiment was 4 stroke, single cylinder CI engine by preheated blend from 30-100°C and speed ranges 1500-4000 rpm. Various parameters like brake power, thermal efficiency, BSFC, emissions were compared. Result shows that SFC was higher compared to diesel

for all loads. Hossain and Davies (2012), worked on the indirect injection CI engine which uses neat Jatropa and Karanj oil as fuel. Modifications were done on the cooling water circuit and fuel supply system such that jacket water was preheated. BSFC increases 3%, CO₂& NO_x increases by 8%, as compared to diesel. From this it is concluded that the IDI compression ignition engine can be used with pure preheated Jatropa oil by jacket water. Shivaji and Gowreesh (2014), worked on the Pongamia biodiesel preheated to 80°C to study the effect of preheating of biodiesel on engine characteristics. The performance characteristics like total fuel consumption, brake specific fuel consumption, brake specific energy consumption, brake thermal efficiency and the combustion characteristics like variation in cylinder pressure and net rate of heat release are investigated. The engine characteristics are also investigated with preheated biodiesel at injection opening pressure of 200, 220 and 240 bars and at CRs of 17, 17.5 and 18. Improvement in engine characteristics is observed by preheating the biodiesel. Among different injection opening pressure and compression ratios, highest brake thermal efficiency was obtained at 200 bars and 18 compression ratios respectively. Martin and Prithviraj (2011), investigated, the viscosity of cottonseed oil (CSO) by blending it in different proportions with diesel, and its viscosity at various temperatures was analyzed. Then, they are used as a fuel in a CI engine. Performance, combustion and emission parameters at various loads were calculated using a single cylinder CI engine and compared with neat diesel and cottonseed oil. A remarkable improvement in the performance of the engine is noticed as the viscosity of the oil is reduced. Brake thermal and volumetric efficiencies of the engine increased with a significant reduction in the exhaust gas temperature. Reductions in smoke, CO and HC emissions are also noticed. Results show that a blend containing 60% of cottonseed oil with diesel, which is heated to a temperature of 70°C, can be used as an alternate fuel without any engine modification. Augustine et al. (2012), investigated the effects of preheated cottonseed oil methyl ester on performance parameters on a single cylinder diesel engine (660 cc). They concluded that BSFC is higher than that of diesel engine for all loads tested. This was due to more blended fuel which is used to produce same power as compared to diesel fuel. Moreover, brake thermal efficiency was lower than diesel fuel but increased by the preheated temperature ranging from 40°C up to 80°C, beyond 100°C there is vapor locking in the fuel line and hence more fuel consumption is noticed for the same power compared to other mode of operation. Results also shown that, preheated cottonseed oil methyl ester caused to lower the CO, unburned HC and smoke compared to diesel fuel. This were attributed to the higher O₂ content of

biodiesel which could improve the combustion process and heating process decreases the viscosity of biodiesel, thus improves the oxidation of biodiesel in the cylinder. However, preheated cottonseed oil methyl ester yields higher NO_x emission at all loads than that of diesel fuel. Yilmaz and Morton (2011), studied the performance of three vegetable oils at two different engines Yanmar and Kubota engines. Brake thermal efficiency, exhaust gas temperature, and CO, O₂, Unburned HC and NO emissions are determined as a function of load at ambient and preheat conditions for peanut, sunflower and canola oils. They found that preheating increases thermal efficiency and vegetable oil shows higher thermal efficiencies than diesel fuel for all of the preheated fuels and both engines. Vedharaj et al. (2015), investigated the performance emission and economic analysis of preheated CNSL biodiesel as an alternate fuel for a diesel engine. Preheating the biodiesel reduced the viscosity, and the performance and emission of preheated CNSLME has been improved. At an inlet fuel temperature of 80°C, the CNSLME discerns a 20% increase in BTE (brake thermal efficiency), 66% and 52% decrease in CO and HC emission, respectively than unheated CNSLME. To purport the economic viability of CNSL, a detailed economic analysis has been conducted.

2.3.3 Diesel engines fueled with Preheated Biodiesel–Diesel Blends

Blending biodiesel with diesel fuel plays an enormous role for a reduction of kinematic viscosity and density of biodiesel oil, and the important effects on fuel droplet formation, atomization, vaporization and fuel-air mixing process, which is influenced the exhaust emissions and performance parameters of the diesel engine. But if greater blend ratio of the oil (above 20%), it has considerably higher viscosity compared with diesel fuel, and needs modification of the properties of the biodiesel oil. However, the fuel properties of biodiesel are reduced by preheating along with blending with diesel fuel (combined heating and blending) can be a viable option to reduce the viscosity and density of biodiesel oil and its blends with diesel close to that of diesel (Agarwal and Agarwal, 2007; Krishnamoorthi et al., 2016). Preheated biodiesel and its blends results in a better fuel consumption and thermal efficiency due to decrease of the arithmetic diameter of the fuel droplets. There have been some investigations on using preheated biodiesel and blending with diesel in diesel engines. Hazar and Aydin (2010), investigated the effect of preheated raw rapeseed oil diesel blend on the engine performance and emission characteristics of a CI engine. The effects of fuel preheating to 100 °C fueled with rapeseed oil diesel blends were clarified. Results showed that preheating of RRO was lowered RRO's viscosity and provided

smooth fuel flow Heating is necessary for smooth flow to avoid fuel filter clogging. It can be achieved by heating RRO to 100 °C, 20% oil –80% diesel (O20) gives optimum results. It can also be concluded that preheating of the fuel have some positive effects on engine performance and emissions when operating with vegetable oil. Jain et al. (2017), studied the performance and emission characteristics of preheated and blended thumba vegetable oil in a single cylinder compression ignition engine. It has been observed that preheated thumba oil B-20 blend (20% thumba oil + 80% diesel) gives better performance and less emission as compared to all other blending combinations among all preheated and unheated thumba oil and diesel. Preheating of optimized thumba blend results an improvement in break thermal efficiency by 1.27% (25.93%) higher thermal efficiency and 0.02 kg/kW.hr (0.33 kg/kW.hr) lesser break specific fuel consumption than the unheated same blend (24.66%) (0.35 kg/kW.hr). It also gives reduced smoke opacity, CO, HC emissions by 2.6%, 0.2%, and 5 PPM, lesser HC, 0.08% higher CO₂, and 11 PPM higher NO_x, respectively than unheated same blend. Martin et al. (2017), Investigated the performance of the engine using preheated blends and cottonseed oil were used in a single cylinder diesel engine. The viscosity of this oil is decreased by blending with diesel and preheating the CSO–diesel blends to reduce the viscosity further. Test results show that there is a marginal increase in the brake thermal efficiency of the engine, as the fuel inlet temperature of the blend and the amount of diesel in the blend increases. It is increases from 28% to the maximum of 30.5% with preheated CSO and diesel mixture of 40%. The smoke, carbon monoxide (CO) and unburnt hydrocarbons (HC) emissions of the engine is also less with the preheated blends. Smoke emission reduces from 3.9 Bosch smoke unit (BSU) to 3.5 Bosch smoke unit (BSU) which is very close to diesel smoke value of 3.4 BSU. NO emission for the CSO operation is 703 ppm and 756 ppm with diesel at full load. Heat release rates indicated an increase in combustion rate with preheated mixtures. The cylinder peak pressure increases from 70.4 bar to 72.5 bar for optimum preheated mixture (60% CSO and 40% diesel at 343 K). From the engine test results it has been established that 60% of cotton seed oil at 343 K can be substituted for diesel. Nagaraja et al. (2015), investigated the CR effect on preheated blend. Palm oil diesel blend is used as a fuel with 5%, 10%, 15%, 20% blends with diesel. Preheating up to 90°C giving efficiency near to diesel fuel. Various CR of 16:1, 17:1, 18:1, 19:1 and 20:1 is used for experimentation. From result it is found that, the blend O20 is found to give maximum mechanical efficiency at higher compression ratio and it is 14.6% higher than diesel. Also the brake power of blend O20 is found to be 6% higher

than standard diesel at higher compression ratio and indicated mean effective pressure of blend O20 is found to be lower than diesel at higher compression ratio. Exhaust gas temperature is low for all the blends compared to diesel. The emission of CO, HC dropped with an increase in blending ratio and compression ratio of maximum load. Also CO₂ emission found to be higher than diesel. The engine performance was found to be optimum when using O20 as fuel at compression ratio 20:1 during full load condition. Khalid et al. (2014), studied significant effects of preheated biodiesel blends on the engine performance and emission of a diesel engine. The blending of biodiesel was varied from 5vol% (B5)~45vol% (B45) and preheated fuel temperature from 40°C~60°C, and the load test conditions of 100% are considered. Under high load condition, preheated biodiesel blends were found enhancing the combustion process, resulting in better performances. Increased preheated fuel temperature, higher in torque value and brake power increases significantly as the engine speed increases. The production of NO_x did not show any improvement when using preheated CPO biodiesel fuel. The increment between the different inlet temperature just give slightly change for the NO_x emission. However, preheated CPO biodiesel fuel reduces CO and CO₂ emissions as compared to the diesel fuel. The effect of temperature only gives the slightly changes to CO emission for all the fuel tested.

2.3.4 Diesel Engine run with Intake Air Preheating Mode

Air preheating is a technique to increase the temperature of inlet air by utilizing the thermal energy of exhaust gas with a prime objective of increasing the thermal efficiency of the engine. Air preheating not only improves overall efficiency of the engine but also act as an emission control technique. There are several studies which reports improvement in the engine performance and emission by using intake air preheating in diesel engines. Yilmaz (2012a), conducted research to observe the effect of preheating the intake air on engine performance on a D.I. diesel engine fuelled by ethanol diesel blend. The main objective of this project is to study the performance, and to control the emissions of the diesel engine using blended fuel by preheating the inlet air. They proved that, intake air preheat is one of the effective solutions to reduce CO and HC emissions significantly, and engine performance was improved when air is preheated. Reduction of diesel concentration in waste coconut oil blends also showed similar effects to preheating intake air temperature. Higher the inlet air temperature increases brake thermal efficiency, decreases the specific fuel consumption, reduces emissions and also reduces the lower ignition delay. Due to

preheating of inlet air uniform combustion takes place that to reduces engine noise. Saddu et al. (2016), have experimentally proved that the effect of preheated air on standard diesel fuel engine indicated a good result on emission control. The NO_x and CO emissions at intake air temperature of 55°C were less when compared at intake are temperature of 32°C. The results proved that higher inlet air temperature will be affect some factors, i.e. lower ignition delay, lower NO_x formation, uniform or better combustion, lower engine noise, easy vaporization, better mixing an air and fuel occur due to warm up of inlet air, lower CO emission. The heat input required for the engine reduces with increase in intake air temperature. Fuel consumption reduces and brake thermal efficiency increases. The CO content in the exhaust gas slightly reduces with increase in intake air temperature. The CO₂ and O₂ content in the exhaust gas remains unaltered in the exhaust with increase in intake air temperature. The NO_x content in the exhaust gas slightly increases with increase in intake air temperature. Kumar et al. (2013), under take experimental investigation on DI diesel engine fuelled by ethanol diesel blend with varying inlet air temperature. Performance and emission characteristics of various concentration of ethanol diesel blended fuel at different inlet air temperature is tested and compared with neat diesel fuel and following results were obtained. The brake thermal efficiency of ethanol diesel blend is lower without pre heating condition, but at 40°C and 50°C inlet air condition, for 10% ethanol diesel blends gives the much better BTHE compare to the neat diesel fuel. The CO and HC emission is higher for the pre heated condition. Without pre heating condition, it produces less smoke compare for ethanol diesel blends. Paykani et al. (2011), analyze the effect of exhaust gas recirculation and intake preheating on performance and emission characteristics of dual fuel engines at part loads. The use of EGR at high levels seems to be unable to improve the engine performance at part loads. However, it is shown that EGR combined with preheating of inlet air can slightly increase thermal efficiency, resulting in reduced levels of both unburned hydrocarbon and NO_x emissions. The CO and UHC emissions are also reduced. Pan et al. (2015), investigated the impact of intake air temperature on performance and exhaust emissions of a diesel methanol dual fuel engine. The experimental results show that there was a strong coupling between the intake air temperature and the methanol fraction in the performance and emissions of the engine. At dual fuel operation mode, decreasing intake air temperature reduced the indicated thermal efficiency and exhaust gas. Decreasing of intake air temperature also prolonged the ignition delay, which caused a later combustion phasing and smaller peak cylinder pressure. Papagiannakis (2013), investigated the study of air inlet preheating

and EGR impacts for improving the operation of compression ignition engine running under dual fuel mode. By comparing the investigation results, it is revealed that the simultaneous increase of both parameters examined could be a promising solution to improve engine efficiency (increase up to 5%) and reducing CO emissions (decrease up to 10%) from a pilot ignited dual fuel diesel engine, without imparting any serious problem to engine operational lifetime. Yilmaz (2012a), studied the effect of intake air preheat and fuel blend ratio on performance and emissions of a compression ignition engine operating on fuel concentrations of biodiesel (85%)–methanol (15%), biodiesel (90%)–methanol (10%), biodiesel (95%)–methanol (5%), neat biodiesel (B100). Fuels were evaluated in the engine at two elevated intake air temperatures at 30 °C and 85 °C. The resulting emissions indicate that the high heat of vaporization of methanol affects CO and HC emissions significantly. Preheating the intake air or lowering the methanol concentration in biodiesel–methanol blends tends to reduce the production of CO and HC while increasing NO emission. Due to preheating of inlet air uniform combustion takes place causes to reduce engine noise. Krishnamoorthi and Malayalamurthi (2018), studied the influence of charge air temperature and exhaust gas recirculation on the availability analysis, performance and emission behavior of diesel - bael oil - diethyl ether blend operated diesel engine. Performance, emission and combustion characteristics along with exergy analysis of ternary test fuel of 60% diesel + 30% bael oil + 10% diethyl ether (DEE) were performed. When operating the diesel engine with 47 °C CAT, brake thermal efficiency (BTHE) is improved to 29.33 %, carbon monoxide (CO), hydrocarbon (HC), and emissions have been reduced by 8.57%, 4.28% and 6.01% at peak engine load. The oxides of nitrogen (NO_x) have been reduced by 20.12% at 100% engine load for 30% EGR mode. The maximum exergy efficiency of 54.61% has been observed at full engine load for the 47 °C CAT. Tamilvendhan and Ilangovan (2011), have performed that the performance and emission and combustion Investigation on hot air by using the preheater in single cylinder, air cooled, vertical and direct injection diesel engine. The inlet side of the engine consists of anti-pulsating drum, air heater and air temperature measuring device. The amount of preheat required for the intake air depends upon the load condition of the engine. Basically the engine requires more preheat during starting idling condition and at lower loads but it requires less preheat at higher loads. The brake thermal efficiency increases with increase in intake temperature, reaches a maximum condition and thereafter decreases considerably at all loads. It reduces ignition delay

and advances the occurrence of peak pressure. The volumetric efficiency decrease and the density of air decreases.

2.4 Modified Engine Operating Parameters (CRs–IPs–ITs)

The engine operating parameters also play a key role in tuning the engine conforming to the better performance and emission standards. Many experiments were carried out with modified parameters i.e. varying CRs, IPs, ITs and it is having different combinations using biodiesel and its blends as a fuel in CI engine. Focusing on previous work, there is quite significant changes in engine performance, combustion and emission characteristics by varying injection pressure, injection timing and compression ratio as compared to standard value.

2.4.1 Effect of Modified Fuel Injection Pressure (IP) on Engine Characteristics

In this section, the impact of changing IP on the characteristics of diesel engine at different operating conditions has been studied and notable works are collectively presented here. **Table 2.1** presents, the percentage change in of performance, combustion and emission characteristics of various blends at varying fuel IP as compared to diesel as a base fuel (Channapattana et al., 2015; Gumus et al., 2012; Jaichandar and Annamalai, 2013; Jindal et al., 2010; Lalvani et al., 2016; Sayin et al., 2012; Wamankar et al., 2015). Investigated the effect of fuel injection pressure (180, 210 and 240 bar) on the performance and emission characteristics of single cylinder variable compression ratio, CI DI engine fuelled with the blends of Honne biodiesel and diesel (Channapattana et al., 2015). The results of biodiesel and diesel blends are compared with that of diesel. It indicated that the BSFC for Honne biodiesel is higher by 0.042 kg/kW.hr as compared to that of diesel oil at IP of 240 bar. A marginal increase in BMEP with increase in IP is observed for all the fuels tested. The variations of thermal performance parameters at different Injection pressures from 180bar to 240 bar in steps of 30bar. The NO_x emissions increase with increase in IP. Gumus et al. (2012) has discussed the impact of varying IP (180, 200, 220 and 240 bar) on exhaust emissions of Lombardini 6 LD (single cylinder DI) diesel engine using canola oil methyl ester and its B0, B5, B20, B50 and B100 blends with diesel. They had found out that at higher IP (240 bar), BTHE was higher and BSFC was lower for higher blends (B100) because at high pressure, air fuel mixing and atomization in spray forms well. Results also show that HC, CO and smoke opacity decreases although NO_x, O₂ and CO₂ emissions increased at higher IP due to increase in the biodiesel percentage in blends and the temperature of the combustion chamber also

increases. Puhan et al. (2009) had also investigated the effect of three varying IP (200, 220 and 240 bar) on diesel engine characteristics using high linolenic linseed oil methyl ester (LOME) as a fuel in Kirloskar make a single cylinder, 4- stroke air cooled DI diesel engine. The result shows that BTE is slightly improved while BSFC is decreased at higher IP (240 bar). It also shows that except NO_x all emissions i.e. UHC, smoke and CO were found to be lower as compared to diesel at 240 bar. Behera and Murugan (2013) had carried out experimentation to find out the effect of IP (from 200 to 250 bar with the gap of 10 bar) on 1-cylinder, 4-stroke, air cooled diesel engine at a constant speed of 2200 rpm running with used transformer oil (UTO). The experimental result shows that when IP increased to 230 bar, BTE decrease in percentage by 8.92 and 7.89 than that of diesel and UTO at 200 bar respectively. They had also found out that UHC, CO and smoke opacity were decreased 43.3%, 60% and 8.92% respectively while NO_x emission increases by 29.7% at 230 bar as compared to diesel at 200 bar IP. Sayin et al. (2012) had carried out experiments on single cylinder four stroke DI diesel engine running with fuel canola oil BD (COME) and its blends at different IPs (180, 200, 220 and 240 bar) to find out its combustion, emissions and performance characteristics. The BTHE, BSFC and brake specific energy consumption (BSEC) were better at higher IP, because, at this IP, cylinder pressure increases which improve spray pattern and it promotes better air-fuel mixture. The peak of HRR was found to be lower (i.e. 9.78%) for COME as compared to diesel fuel because of the lower heating value of COME. Jaichandar and Annamalai (2013) had tested biodiesel fueled diesel engine for different injection pressure and combustion chamber (CC) geometry to examine the performance and emission characteristics. They had concluded that BSFC and BTE improved for higher IP having toroidal re-entrant combustion chamber geometry. They also found out that increase in the maximum rate of pressure rise (MRPR) and peak pressure in cylinder, at higher IP. Consequential improvements in reduction of emissions percentage were found at higher IP while NO_x increases. Liu et al. (2015) had tested the emission and performance characteristics using blend of diesel methanol in heavy duty common-rail diesel engine with different values of injection pressure (70, 85, 100, 115, 130, 140 bar). They had reported that the cylinder pressure and HRR increase with the increase in IP (115 bar). BSFC was founded to be lower at this injection pressure. In the emissions, oxides of carbon (CO) and UHC increases but CO₂, smoke. and NO_x emissions decreases as compared to diesel. Lalvani et al. (2016) had investigated emission, performance and combustion characteristics of 'Kirloskar' make water cooled single cylinder four stroke DI diesel

engine. The experimental result shows that BTHE was lowered at all injection pressure except 220 bar for B20 blend while CO, hydrocarbon and smoke emissions were decreased by 48.47%, 32.82% and 13.62% respectively for the same blend as compared to diesel although NO_x emission increases by 9.48% as compared to diesel at 200 bar IP. Celikten (2003) had investigated the performance and emission characteristics at different injection pressure (100, 150, 200 and 250 bar) on 4-cylinder four-stroke indirect injection diesel engine. They found out that the best results in terms of performance were at 150 bar. At higher speed, BSFC and CO emission decreases. They had also reported that NO_x emission increases at 100% throttle position due to increase in cylinder temperature. Nanthagopal et al. (2016) had used Calophyllum inophyllum BD (B100) as a fuel in DI diesel engine for different injection pressures i.e. 200, 220 and 240 bar. The result shows that BSFC, CO, unburnt hydrocarbons and SO (smoke opacity) were reduced at 220 bar as compared to other IP i.e. 200 bar and 240 bar whereas NO_x increases with increase in IP. Hwang et al. (2014) had carried out experiments on a single cylinder direct injection diesel engine using two blends of diesel and waste cooking oil methyl ester (WCO) with injection quantity 20, 40 mg/stroke of diesel with 22.1, 44.3 mg/stroke BD for both blends respectively. It ensures the equivalent injection energy per cycle i.e. (20:22.1 and 40:40.3 in mg/cycle) at varying IP (80 and 160 MPa) and IT (from -25 to 0° CA) after top dead center (aTDC). The result shows that BSFC was higher whereas peak cylinder pressure and HRR were slightly lower for biodiesel as compared to diesel fuel at advance injection timing. In terms of emissions, HC, CO and smoke were decreased especially at higher IP but NO_x emissions were found to be higher than those of the diesel. Labecki and Ganippa (2012) had conducted a combustion and emission tests on Multi-cylinder turbocharged DI diesel engine using rapeseed oil methyl ester and its blends (10%, 20%, 30% and 50%) as a fuel with varying IP (800, 1000 and 1200 bar) and IT (9° and 4° bTDC). The result shows that increase in peak cylinder pressure and maximum HRR were obtained with increase in IP and IT. This is due to a shorter ignition delay. They also stated that UHC, carbon monoxide and SO emissions were reduced while NO_x emission increases at higher IP. Rao and Saketh (2014), investigated the effect of injection pressure of diesel engine performance and emission fueled with pongamia pinnta methyl ester (PPME) blends with diesel in different compositions from B0 to B100 in steps of 20% at varying injection pressures of 150, 200 and 250 bar. Tests were conducted on a single cylinder diesel engine at maximum load of 12 kg, constant speed of 1500 rpm, CR 18. Brake thermal efficiency and specific fuel consumption increases as the injection pressure increase. Among all

the tests, B20 blend fuel has higher brake thermal efficiency of 31.3% in comparison 32.7% for diesel, but marginally at higher specific fuel consumption than diesel. The biodiesel blend has maximum nitric oxide emission of 195 ppm, while it was 182 ppm for diesel. Substantial reduction was observed in carbon monoxide, carbon dioxide, HC emissions and smoke density in the full range of load and at higher injection pressure. The results reveal that the biodiesels can be used safely to replace diesel, as alternative and renewable fuel without any modifications to the engine. Channapattana et al. (2015), worked on the effect of injection pressure (180, 210 and 240 bar) on the performance and emission characteristics of VCR engine using blends of Honne biodiesel and diesel as a fuel. The results with biodiesel and diesel blends are compared with that of diesel. The result indicated that the BSFC for Honne biodiesel is higher by 0.042 kg/kW.hr as compared to that of Diesel oil at IP of 240 bar. A marginal increase in BMEP with increase in IP is observed for all the fuels tested. At a higher compression ratio of 18 and injection pressure of 240 bar, fairly reduced emissions are observed irrespective of the fuel used. The NO_x emissions increase with increase in IP. The NO_x emissions increase as the blend proportion is increased. Biodiesel blend B20 gives better thermal performance compared to other blend of Honne biodiesel, but it causes higher levels of exhaust emissions.

Table 2.1: Reported engine characteristics of various feedstock of biodiesel as compare to diesel at varying IPs.

Compound biodiesel [Ref.]	Common fuel/ operating pressure (bar)[A]	Variation injection pressure (bar)	Performance analysis (%) [C]			Combustion analysis (%) [D]			Emission analysis (%) [D]			Optimal condition [B]	Effect in optimal IP (B) to operating IP (A)		
			BSFC	BTHE	EGT	HRR	MHRR	PCP	HC	CO	NOx		C	D	E
COME (B100) (Sayin et al., 2012)	Diesel/200	180	-8.38	+20.74	-	-10.2	-6.57	-	-	-	-	Best result shows at 180 bar where BTHE increase and max. rate of pressure rise and heat release rate decrease.	↑	↑	
		200	-0.053	+11.43	-	-20.21	-6.25	-	-	-	-				
		220	+16.41	-4.78	-	-21.27	-2.36	-	-	-	-				
		240	+21.47	-8.77	-	-21.55	-1.94	-	-	-	-				
CBDWD (B10) (Wamankar et al., 2015)	Diesel/200	200	-	-3.24	+2.82	-	-	-	+60.0	+90.0	-16.0	BTE, EGT increase at 220 bar. also min. emission at 220 bar.	↑	-	↑
		220	-	-1.29	+0.94	-	-	-	+34.0	+76.10	-20.0				
		240	-	-8.76	+1.88	-	-	-	+59.5	+80.20	-15.0				
JME (B100) (Jindal et al., 2010)	Diesel/210	150	+41.9	-22.08	-	-	-	-	-36.84	+18.51	-10.41	The best performance was shown by engine at 250 bar IP, NOx, HC were at reduced level while CO increased.	↑	-	↑
		200	+38.7	-14.29	-	-	-	-	-57.89	+33.33	-24.16				
		250	+38.7	-5.33	-	-	-	-	-55.26	+77.77	-31.25				
POME (B20) (Jaichandar and Annamalai, 2013)	Diesel/200	185	+27.83	-19.70	-2.44	-27.65	-17.68	-15.52	+6.36	-9.88	+0.72	Slightly higher IP (220bar) gave better characteristics as compare to original IP.at modified IP BSFC decrease and BTE increase while HC and CO were decreased and NOx increased.	↑	↑	↑
		200	+5.46	-2.35	+9.18	-12.76	-9.75	-5.59	-9.09	-34.88	+2.14				
		210	+5.08	-1.17	+17.34	-8.51	-6.70	-4.34	-14.54	-36.04	+4.28				
		220	+2.96	-0.59	+18.77	-4.26	-3.65	-1.86	-11.81	-40.69	+7.14				
		230	+11.44	-10.0	+12.24	-14.89	-18.29	-10.18	-7.27	-18.02	+2.85				
AOME (B20) (Lalvani et al., 2016)	Diesel/200	185	+23.30	-20.82	-0.41	-6.57	-	-17.48	-12.97	-8.92	+1.45	At 200 bar BSFC and RHRR were minimal and BTE was higher. Emission characteristics decreased at 220 bar except NOx	↑	↑	↑
		200	+6.79	-0.293	+14.16	-24.67	-	-19.33	-27.48	-34.52	+6.56				
		210	+5.82	-0.010	+18.33	-4.38	-	-5.78	-30.53	-35.71	+8.75				
		220	+1.84	+0.586	+21.45	-17.32	-	-3.07	-32.82	-40.47	+9.48				
		230	+19.41	-7.62	+14.79	-12.71	-	-8.49	-25.19	-16.67	+3.65				
COME (B50) (Gumus et al., 2012)	Diesel/200	180	+12.10	-5.82	-5.79	-	-	-	+8.77	+51.78	+18.84	Increasing BTE, EGT and decreasing BSFC while in emission HC and CO decrease while NOx increase.	↑	↑	↑
		200	+3.89	+1.64	+1.56	-	-	-	-14.03	-18.94	+22.46				
		220	-0.16	+5.56	+2.83	-	-	-	-24.56	-40.84	+32.60				
		240	-1.94	+7.01	+2.97	-	-	-	-31.57	-70.10	+24.63				
HnOME (B40) (Channapattana et al., 2015)	Diesel/180	180	+3.43	-5.48	-1.04	-	-	-	-29.93	-24.68	+5.65	All blends B40 gave acceptable results at 240 bar. BSFC, HC, CO, and smoke decrease in significant amount while EGT, NOx were increased at higher IP.	↑	-	↑
		210	-3.43	-3.47	+6.21	-	-	-	-63.40	-44.83	+24.23				
		240	-10.0	+0.71	+12.12	-	-	-	-83.88	-61.71	+50.68				

2.4.2 Effect of Modified Compression Ratios (CRs) on Engine Characteristics

In this section, impact of changing CR on the characteristics of diesel engine at different operating conditions has been studied. The Notable works are collectively presented here. **Table 2.2** tabulated, the various studies carried out the percentage change in of performance, combustion and emission characteristics of various blends at varying CR as compared to diesel as a base fuel (EL_Kassaby and Nemit_allah, 2013; Kumar and Chauhan, 2013a; Muralidharan and Vasudevan, 2011; Raheman and Ghadge, 2008; Wamankar et al., 2015). Other studies examined the combined effect of using blends of diesel–palm stearin biodiesel as fuels and the compression ratio on different performance, combustion and emission characteristics of the said engine (Ambarish and Mandal, 2017). The experiments have been carried out on a single-cylinder, direct injection diesel engine at varying compression ratio of 16 to 18 in four steps. It is observed that the BTHE reduces by 7.9% when neat biodiesel is used instead of diesel. But, it increases with the increase in compression ratio for all the blends. The BSFC and EGT increase with the addition of biodiesel to diesel and also with the increase in compression ratio. Heat release rate decreases with biodiesel, and it is minimum at the rated compression ratio of 17.5 for all the fuels considered here. On the other hand, ignition delay is found to be more with neat diesel, and it increases with the decrease in compression ratio. Significant reductions in emissions of carbon monoxide (CO), hydrocarbon (HC) and smoke are observed with biodiesel, while the emissions of oxides of nitrogen (NO_x) and carbon dioxide (CO₂) increase. The decrease in compression ratio increases the emissions of CO, HC and smoke, but the emissions of NO_x and CO₂ decrease with the decrease in compression ratio. Hirkude and Padalkar (2014), investigated the performance and emission characteristics with CR variations (14.5, 16.5 and 17.5) in a single cylinder four stroke DI diesel engine using waste fried oil methyl ester and diesel blends (B0, B50 and B70). The effect of variation in the compression ratio on brake thermal efficiency indicated that higher compression ratios improve the engine efficiency. The best results for BSFC were obtained at increased CR. The results showed that CO and particulate matter emissions decreased while nitrogen oxide emission increased with an increase in CR. For all tested fuels an increase in compression ratio leads to an increase in exhaust gas temperature. The reductions in exhaust emissions and brake specific fuel consumption together with increased brake power, and brake thermal efficiency made the WFOME from waste fried oil a suitable alternative fuel for diesel at higher compression ratios and thus could help in controlling air pollution. Muralidharan and Vasudevan (2011), had demonstrated engine characteristics using

waste cooking oil BD and diesel blends as a fuel in 4- stroke water cooled CI engine with rated power 3.7 kW at constant speed of 1500 rpm at variable CR from 18 to 22 (5 values). The experimental results revealed that blend B40 gives the better performance than other blends. As compared to diesel, BTE and BSFC for B40 was about 20.7% and 17.5% higher respectively. In addition, NO_x emission was slightly higher (nearly 3.05%) as well as hydrocarbon emissions showed piling up effect as compared to diesel. The CO₂ emission decreases however CO emissions was found to close to diesel. Mohanraj and Mohan Kumar (2013), had conducted an experiment on Tamanu oil methyl ester with 5 different CR (14 to 18) on single cylinder Kirloskar engine at fixed speed of 1500 rpm. The result shows that BSFC and BP at compression ratio 18 were 0.24 kg/kW.hr and 3.51 kW respectively which were slightly higher as compared to diesel. The hydrocarbon emission is higher at low compression ratio and vice versa. The increase in compression ratio decreases the HC emission for esterified Tamanu oil. The emission of oxides of nitrogen (NO_x) from the esterified Tamanu oil is higher for high compression ratio than that of low compression ratio. The CO emission of the biodiesel is very less at compression ratio 18. The CO₂ emission is maximum for the same compression ratio 18. From the experimental results, it has been found that the performance parameters like SFC, BTHE, and volumetric efficiency are comparably better for high compression ratio in VCR engine run with esterified Tamanu oil. The exhaust gas analysis also gives acceptable emission values for esterified Tamanu oil used in the VCR engine. The experimental result proves that the esterified Tamanu oil can be used in diesel engine. EL_Kassaby and Nemit_allah (2013), had investigated the effect of variable CR from 14 to 18 in four stroke direct ignition diesel engine running on waste oil BD and diesel blends (B10, B20, B30 and B50). The result shows that the engine torque for all blends increases as the compression ratio increases. The BSFC for all blends decreases as the compression ratio increases and at all compression ratios BSFC remains higher for the higher blends as the biodiesel percent increase. The change of compression ratio from 14 to 18 resulted in, 18.39%, 27.48%, 18.5%, and 19.82% increase in brake thermal efficiency in case of B10, B20, B30, and B50 respectively. On an average, the CO₂ emission increased by 14.28%, the HC emission reduced by 52%, CO emission reduced by 37.5% and NO_x emission increased by 36.84% when compression ratio was increased from 14 to 18. In spite of the slightly higher viscosity and lower volatility of biodiesel, the ignition delay seems to be lower for biodiesel than for diesel. On average, the delay period decreased by 13.95% when compression ratio was increased from 14 to 18. From this study,

increasing the compression ratio had more benefits with biodiesel than that with pure diesel. Kim et al. (2015), had evaluated the emission and performance characteristics of diesel engine running with the wood pyrolysis oil (WPO) and butanol blends (0%, 10%, 20% and 30%) and results were compared with the base fuel (diesel). The authors had reported that B30 gave the combustion stability with decrease in NO_x emissions because of B30 has higher water contents (about 5.2%) than other blends which vaporised at high value of CR and decreases the cylinder temperature. At lower load, carbon monoxide emissions were found increase. HC emission was decreased at all the load conditions as compared with diesel. EL_Kassaby and Nemit_allah (2013), had used waste cooking oil and its blends with diesel as a fuel in CI engine with different values of CR (14, 15, 16, 17 and 18). The BSFC for all blends decreases as the compression ratio increases and at all compression ratios BSFC remains higher for the higher blends as the biodiesel percent increase. The change of compression ratio from 14 to 18 resulted in, BTHE increase to 18.39%, 27.28%, 18.5% and 19.82% for the blends B10, B20, B30 and B40 respectively. On an average, the CO₂ emission increased by 14.28%, the HC emission reduced by 52%, CO emission reduced by 37.5% and NO_x emission increased by 36.84% when compression ratio was increased from 14 to 18. In spite of the slightly higher viscosity and lower volatility of biodiesel, the ignition delay seems to be lower for biodiesel than for diesel. On average, the delay period decreased by 13.95% when compression ratio was increased from 14 to 18. From this study, increasing the compression ratio had more benefits with biodiesel than that with pure diesel. Senthil et al. (2015b), had analyzed the effect of variable CR (16.5, 17.5, 18.5, 19.5 and 20.5) and IT (24, 27, 30 and 33° bTDC) on combustion, emission and performance characteristics of Annona methyl ester (AME) blend (B20) with diesel operated in CI engine at 50% load. The result shows that, injection timing of 30° bTDC, along with compression ratio of 19.5 gives better performance, combustion and lower emissions when compared with standard Injection timing of 27° bTDC and CR of 17.5. For all tested values, A20 provides best results in terms of BTHE, higher heat release rate, and lower emissions of HC, CO and NO_x. Hence, A20 can be effectively used as an alternative biodiesel with injection timing of 30° bTDC along with compression ratio of 19.5 in tested engine. Chavan et al. (2015), had studied the performance and emission characteristic of a single-cylinder VCR engine, fuelled with Jatropha oil methyl ester and its blends (B0, B10, B20, B30 and B100) with diesel at a constant engine speed of 1500 rpm and CR varied from 14 to 18. The results showed that, JB30 shows reduction in emissions of CO and HC up to 43 and 50%, respectively, with an increment of NO_x.

emission up to 20% at the lowest load and compression ratio (CR) of 15. The optimum parameter for the lowest pollutant emission for JB30 was found with a load of 6 kg at CR of 15. Jindal et al. (2010), had also used Jatropha methyl ester as a fuel with same conditions. They found out that BTE and BSFC improved by 8.9% and 10% respectively, at CR 8 and IP 250 bar. It is found that the combined increase of compression ratio and injection pressure increases the BTHE and reduces BSFC while lowering emissions. For small sized direct injection constant speed engines used for agricultural applications (3.5 kW), the optimum combination was found as CR of 18 with IP of 250 bar. Singh and Shukla (2016), had carried out an experiments on four-stroke water-cooled CI engine running with castor oil BD and its blends (0%, 10%, 20%, 30%, 40% and 50%) with diesel on variable compression ratio (15 to 18) to investigate the combustion characteristics. It is observed that, mean gas temperature increases with compression ratio for all blending ratios and gives maximum value for B00. Cylinder pressure increases with compression ratio and maximum value is obtained at compression ratio 18 for B50. Net heat release decreases with compression ratio and maximum value is obtained for B00 at compression ratio 15. Mass fraction burned is better for B20 and poor for B50 for compression ratio 15 and 18. Kumar and Dixit (2014), studied, emission and combustion characteristics of diesel engine when the engine operated using the different blends (B10, B20, B30, and B40) and normal diesel fuel (B0) as well as when varying the compression ratio from 16.5 to 18.5. The change of compression ratio from 16.5 to 18.5 resulted in 27.1%, 27.29%, 26.38%, 28.48%, and 34.68% increase in cylinder pressure for the blends B0, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. Higher peak heat release rate increased by 23.19%, 14.03%, 26.32%, 21.87%, and 25.53% for the blends B0, B10, B20, B30, and B40, respectively, at 75% of rated load conditions. When compression ratio was increased from 16.5 to 18.5, the delay period decreased by 21.26%, CO emission reduced by 14.28%, and NO_xemission increased by 22.84% for B40 blends at 75% of rated load conditions, when compression ratio was increased from 16.5 to 18.5. It is concluded that Jatropha oil ester can be used as fuel in diesel engine by blending it with diesel fuel. Raheman and Ghadge (2008), had varied IT (35°,40°and45°bTDC) and CR (18,19 and 20) in a single cylinder engine. The result shows that BTE decreases whereas BSFC, EGT increases as the proportion of biodiesel in fuel increase. However, an opposite trend was observed with advanced IT and high CR. EGT, BTE increases and BSFC reduces for biodiesel and its blends as compared with HSD fuel. Due to higher viscosity and low vaporizing properties of biodiesel, high value of CR will perform relatively

better. De and Panua (2014), investigated the performance and emission characteristics of single cylinder, four-stroke VCR, direct injection diesel engine running with pure diesel, and Jatropa oil-diesel blended fuels with various blended rates. The measurements are recorded for the compression ratio of 16, 17 and 18 varying the load from idle to rated load of 3.7 kW. The results show that various parameters such as thermal efficiency, CO and NO_x emissions are very close to mineral diesel for lower blend concentrations. But, it is observed that for higher blend concentrations, performance and emissions were much inferior compared to diesel. The thermal efficiency, exhaust gas temperature and emission parameters such as NO_x and CO at CR of 18 for the blends containing up to 30% (by volume) Jatropa oil is close to that of diesel fuel. But with the increase of Jatropa oil concentration in the blends increasing the exhaust gas temperature and emission parameters like NO_x, CO and decreases the thermal efficiency of the engine. Wamankar et al. (2015), investigated the combined effects of compression ratio, nozzle opening pressure and injection timing on the performance and emissions of a compression ignition engine operated with an emulsion fuel obtained from CB (carbon black). The emulsion contained CB 10%, 2% water, 85% diesel and 3% surfactant was denoted as CBWD10. The experimental results revealed that, the brake thermal efficiency of the CBWD10 emulsion is 1.7%–5.1% higher at the compression ratio of 18.5 with different nozzle opening pressures and injection timings compared to that of diesel. The exhaust gas temperature of the CBWD10 emulsion is found to be about 3.3%–9.8% lower at the compression ratio of 18.5 than that of diesel. The CO and HC emissions are found to be about 16% and 35% lower at 26°CA bTDC injection timing with 220 bar nozzle opening pressure with a higher CR of 18.5 at a full load respectively.

Table 2.2: Reported engine characteristics of various feedstock of biodiesels as compare to diesel at varying CRs.

Compound biodiesel [Ref.]	Common fuel/ operating compression ratio (A)	Variation compression ratio	Performance analysis (%) [C]			Combustion analysis (%) [D]			Emission analysis (%) [D]			Optimal condition [B]	Effect in optimal IP (B) to operating IP (A)		
			BSFC	BTHE	EGT	HRR	Mcp	PCP	HC	CO	NOx		C	D	E
WCO (B40) (Muralidharan and Vasudevan, 2011)	Diesel/18	18	-0.66	+0.75	+4.24	+2.74	-	-3.97	-12.92	+13.86	-4.79	B40 blend was maximum BTE at CR 21 than other blends. Emissions increases as blends % increase.	↑	↑	↑
		19	-2.65	+2.51	-6.78	-9.89	-	+0.52	-36.89	-10.94	-8.71				
		20	-8.58	+9.38	-6.62	-22.52	-	-4.15	-39.12	-11.67	-8.27				
		21	+18.5	+17.02	-7.80	-43.95	-	+5.88	-26.02	+38.68	+3.48				
		22	-0.66	+0.70	-8.48	-30.76	-	+9.34	-28.24	-8.75	+19.17				
HLLOME (B20) (EL_Kassaby and Nemit_allah, 2013)	Diesel/18	17	-4.23	+7.24	-	-	-	-	-17.01	-11.98	+14.41	At CR 17 BTE increase, BSFC and emission decrease except NOx	↓	-	↓
		18	-3.14	+6.01	-	-	-	-	-53.23	-18.64	+22.34				
		19	-2.01	+4.89	-	-	-	-	-55.47	-19.01	+34.61				
JME (B40) (Kumar and Chauhan, 2013a)	Diesel/17	16	+38.3	-19.27	-	-	-	-	+105.9	+35.0	-	BTHE increase at CR18 and BSFC and emissions decrease except smoke	↑	-	↑
		17	+41.67	-12.04	-	-	-	-	-18.81	+113.3	-				
		18	+42.91	-0.61	-	-	-	-	-22.77	-50.0	-				
CBWD (B10) (Wamankar et al., 2015)	Diesel/17.5	16.5	-	-25.32	+7.22	-	-	-	-	-	-4.73	BTE increase, and EGT and NOx decrease at CR 18.5	↑	-	↑
		17.5	-	-3.24	+2.82	-	-	-	-	-	-2.61				
		18.5	-	+6.49	-1.57	-	-	-	-	-	-2.12				
Mahua oil (B60) (Raheman and Ghadge, 2008)	Diesel/18	18	+39.77	-26.09	+19.63	-	-	-	-	-	-	BTE more increase than blend B60 and BSFC, EGT decrease as CR increase	↑	-	-
		19	+1.10	-0.085	+3.27	-	-	-	-	-	-				
		20	-10.49	+15.35	-5.43	-	-	-	-	-	-				

2.4.3 Effect of Modified Fuel Injection Timings (ITs) on Engine Characteristics

In this section, impact of changing fuel IT on the characteristics of diesel engine at different operating conditions have been studied and notable works are collectively presented here. In this section, it is presented in **Table 2.3** the significant effects on engine characteristics were found by varying injection timing advancing/retarding/operating (Balusamy and Marappan, 2010; Bari et al., 2004; Belagur and Chitimini, 2012; Sayin et al., 2008; Wamankar et al., 2015). They had studied the performance, combustion and emission characteristics of engine with different injection timing (340° , 345° and 350° CA) run on 5%, 10% and 20% blends of algal oil methyl ester with diesel in Kirloskar DM10 diesel engine (Hariram and Kumar, 2012). It was found out that BSFC decreases and brake thermal efficiency of engine increases with the advancement in injection timing for all the blends than diesel at all loading conditions due to better combustion and atomization. They had also reported that at 5° advancements in IT from original, UHC, CO and smoke were decreased due to longer ignition delay. At the same condition heat release rate, brake mean effective pressure and NOx emission increases due to higher oxygen contents and large premixed combustion phase. Sayin et al. (2008), had studied canola oil biodiesel (COME) and diesel blends as feedstock in single cylinder air cooled naturally aspirated DI diesel engine at IT values of 15, 20 and 25° bTDC to investigate the emission characteristics. They had reported that when IT changes from 20° to 15° bTDC for B50, CO emission increases by 31.25% due to decrease in premixed combustion phase along with retarded injection timing. CO emission decreases as percentage of biodiesel in blend increases because BD has high oxygen content (about 11%) which helps in better combustion. Also IT change from 20° to 25° bTDC, affects CO emission and reduces its content by 18.75% in exhaust emission for B50 because cylinder temperature increases with the advancement in IT. HC emissions increase by 9.1% with the retarding in IT (20° to 15°). It is due to less fuel burned at retarded IT produce less heat which results in maximum gas temperature in the cylinder will be lower. Also combustion space continuously increases during the expansion stroke. As IT changes from 20° to 25° bTDC, NOx emission shows reduction of 8.59% at 20 N-m load conditions. Ganapathy et al. (2011), had investigated the effect of variation in IT (340 , 345 and 350° CA), speed (1800, 2500 and 3200 rpm) and torque (5, 10 and 15 N-m) on engine characteristics using Jatropha biodiesel as fuel in CI engine. Result shows that at 15 N-m, 1800 rpm and 340° CA, BSFC decreased by 5.1% while BTHE increased by 5.3% at 20° bTDC. The HC, CO and smoke emissions were reduced by 1.2%, 2.5% and 1.5% respectively as

compared to diesel (at 350° CA). Belagur and Chitimini (2012), investigated the performance, combustion and emission characteristics using Honge oil methyl ester at different load (25%, 50%, 75% and 100%) and injection timing (23°, 25°, 27° and 28° bTDC) in a single cylinder, 4-stroke DI diesel engine. They had observed that BTE improves and BSFC decreases with the advancement in injection timing. Maximum rate of pressure rise was obtained at 28° bTDC because of longer ignition delay due to advance IT, which improves premixed combustion and increases cylinder pressure and hence rate of heat release. It is also reported that carbon monoxide, UHC and smoke increases with advance in IT. Gnanasekaran et al. (2016), had evaluated characteristics of single cylinder four stroke direct injection CI engine using with Ethyl Ester of Fish Oil (EEFO) and its blends at different loads and at different injection timings, 21°, 24° and 27° bTDC. Oxides of Nitrogen (NO_x), Unburnt Hydrocarbon (UBHC) and Carbon Monoxide (CO) emissions in biodiesel blends were lower than diesel, whereas smoke was found to be higher. The brake thermal efficiency for B20 was higher compared to diesel in the entire load spectra. The ignition delay and combustion duration were shorter for biodiesel blends than diesel which results in lower heat release rate, peak pressure and rate of pressure rise. Retardation of injection timing caused decrease in emission and combustion parameters like NO_x, HC, CO, peak pressure, ignition delay, combustion duration and heat release rate which increased with advancement in injection timing. Gumus et al. (2010), had used canola oil methyl ester (COME) blends (B0, B5, B20, B50 and B100) with diesel as a fuel in single cylinder 4-stroke direct injection diesel engine at three different ITs (15°, 20°, 25° CA) to investigate its performance and combustion characteristics. They had found out that the peak of MRPR, maximum cylinder pressure and maximum HRR were lower for all the blends as compared to diesel at advanced IT (15° CA) due to degradation in mixing of A-F at the time of ignition delay period. Hence, it shortens the premixed combustion phase because of lower volatility and higher viscosity of COME as compared to diesel fuel. Also It has been reported that BTE was found to be decreased by 12.85% at retarded and 4.45% at advanced IT for 100% blend and 20 N-m load while BSFC found to be lower at original injection timing which was 346.48 g/kW.hr and BSEC reduces by 7.82% as load change from 10 to 20 N-m. Jaichandar et al. (2012), had studied the engine characteristics with varying combustion chamber geometry (toroidal re-entrant combustion chamber) and injection timing using 80% diesel – 20% Pongamia oil methyl ester blend as fuel in single cylinder 4 stroke water cooled diesel engine. The result shows that BTHE increased by 5.64% while BSFC decreased by 4.6% for

toroidal re-entrant combustion chamber at retarded IT(21°bTDC) because of better air-fuel mixture which leads to improved combustion of the biodiesel. The result also shows that CO, HC and smoke emission were slightly increased at 21° bTDC because of inadequate premixed combustion phase and NO_x emission increases due to decrease in premixed combustion phase as a result of decrease in the cylinder wall temperature. Kannan and Anand (2012), had investigated the engine characteristics using waste cooking oil methyl ester as a fuel in a single cylinder water cooled direct injection diesel engine having varying injection pressure and ignition timing. IP was varied from 220 to 300 bar and IT from 23° to 28° bTDC. The result shows that at advance injection timing (25.5° bTDC) and pressure (280 bar), HRR, BTE and cylinder pressure shows considerable improvement because at advance IT, combustion also start earlier which overcomes the effect of high viscosity and low calorific value of BD. The smoke and NO_x emission decreases due to the low unsaturated fatty acid content of biodiesel. Jia et al. (2011), had carried out experiments with different ITs from -35° to -5° CA by closing the inlet valve from -140° to -80° CA after top dead centre (aTDC) with 1.8 bar intake pressure and 50% exhaust gas recirculation (EGR). The significant results were obtained at -20° CA aTDC due to inlet valve closing earlier and hence knocking reduces. Raheman and Ghadge (2008), had varied IT (35°,40° and 45°bTDC) and CR (18,19 and 20) in a single cylinder engine. The result shows that BTE decreases whereas BSFC, EGT increases as the proportion of biodiesel in fuel increase. However, an opposite trend was observed with advanced IT and high CR. EGT, BTE increases and BSFC reduces for biodiesel and its blends as compared with HSD fuel. Due to higher viscosity and low vaporising properties of biodiesel, high value of CR will perform relatively better. Kumar and Chauhan (2013a), had evaluated the performance and emission characteristics of single cylinder four-stroke water cooled DI compression ignition engine running with B40 Jatropha biodiesel as a fuel. They have performed total 27 set of experiments with varying CR (16, 17 and 18), IT (20°, 23° and 26° CA) and IP (180, 210 and 240 bar). It has been reported that best results were achieved at higher CR, higher IP and advance IT i.e. CR 18, IP 240 bar and IT 26° bTDC. The BTHE and BSFC were improved (+5.2%, - 4.1%), respectively as compared to diesel. The result also shows that emission characteristics like CO, HC and smoke were decreased by 35.41%, 64.7% and 73.7% respectively at the combination of 'IP240IT26CR18'. Balusamy and Marappan (2010), had conducted experiments using vetia peruviana biodiesel as a fuel in Kirloskar engine with variable IT (23, 25, 27 and 29° bTDC) and IP (210, 215, 220, 225, 230 and 235 bar). Result shows that increase in IT

by 4° CA and IP by 15 bar affects the engine characteristics at 100% load. While use of BD at 27° bTDC, BTE was increased from 31.22% to 33.41% while there was a reduction in HC, CO and smoke by 18.94%, 25% and 13% respectively as compared to standard IT (23° bTDC). Increasing IP by 15 bar at 27°bTDC increases BTE from 33.41% to 35.63% and reduced CO, HC and smoke level by 20%, 41.55% and 12% respectively as compared standard IP (210 bar) conditions. Pandian et al. (2011), had studied the performance and emission characteristics having different combination of IP (150–250 bar) and IT (18° bTDC to 30° bTDC) on twin cylinder naturally aspirated water cooled DI diesel engine. Experiments were designed using response surface methodology. Result shows that with the increase in IP (150 to 225 bar) and IT (18°bTDC to 30° bTDC), the BTHE was increased and BSEC decreases. At the same condition, the emissions in terms of HC, CO and smoke decreases while NOx increases.



Table 2.3: Reported engine characteristics from various feedstock as compare to diesel at varying ITs.

Compound biodiesel [Ref.]	Common fuel/ operating IT (° bTDC) (A)	Variation injection timing (° bTDC)	Performance analysis (%) [C]			Combustion analysis (%) [D]		Emission analysis (%) [E]			Optimal condition [B]	Effect in optimal IT (B) to operating IT (A)		
			BSFC	BTHE	EGT	HRR	PCP	HC	CO	NOx		C	D	E
HOME (B100) (Belagur and Chitimini, 2012)	Diesel/23	23	+27.02	-12.98	-	-15.01	-4.49	+12.58	-68.10	-8.81	Performance, combustion and emission analyses improved at 28° bTDC except NOx	↑	↑	↑
		25	+21.98	-9.24	-	-2.10	-1.45	+6.21	-76.8	-6.14				
		27	+17.81	-4.78	-	-1.58	+0.60	+2.32	-80.0	-3.52				
		28	+17.01	-4.62	-	+4.45	+1.45	-9.51	-76.6	-3.31				
COME (B20) (Sayin et al., 2008)	Diesel/20	15	+24.78	-18.27	+10.75	-18.18	-14.45	-28.96	+5.76	-26.58	Finest result for BTE and BSFC gave at original IT (25°) best for combustion and emission analysis.	↔	↑	↑
		20	+5.50	-3.32	+12.91	-1.17	-0.75	-46.36	-14.07	-4.75				
		25	+11.92	-8.86	+14.11	+48.19	+16.33	-53.50	-22.69	+3.23				
CBWD (B100) (Wamankar et al., 2015)	Diesel/23	23	-	-3.24	+2.82	-	-	+60.21	+90.0	-15.45	At 26° CA bTDC, .EGT and emissions except NOx were lower and BTHE is higher	↑	↑	↓
		24.5	-	-0.65	-1.20	-	-	+64.87	+80.5	-19.0				
		26	-	+4.20	-1.53	-	-	+21.54	+6.24	+16.84				
WCO (B100) (Bari et al., 2004)	Diesel/23	15	+19.23	-5.11	-0.60	-8.56	-2.82	-	+13.09	-33.09	Optimum condition for WCO biodiesel shows at 19° in place of 15° bTDC.	↑	↑	↑
		13	-	-	-	-23.42	-2.48	-	-	-				
		19	+11.48	+7.10	-6.90	-24.59	+6.94	-	+2.81	-29.31				
METPSO (B100) (Balusamy and Marappan, 2010)	Diesel/23	23	-	-15.60	+7.96	-	-	-12.43	-33.33	+19.91	BTHE, EGT and emissions are optimizing level at 4° advance IT.	↑	↑	↑
		25	-	-12.63	+8.66	-	-	-19.54	-36.66	+16.91				
		27	-	-9.13	+7.25	-	-	-34.05	-53.33	+21.62				
		29	-	-20.16	+10.53	-	-	-26.68	-4.33	+27.40				

2.5 Exhaust NO_x Emission Treatment

NO_x emissions from diesel engines are an environmental problem recognized worldwide. Several countries have long had regulations pending that will limit NO_x emissions from the engine. Manufacturers and researchers have put considerable effort toward meeting those regulations. Several solutions have been proposed for controlling NO_x emissions from diesel engines. One set of approaches focuses, implementing exhaust gas recirculation techniques in a diesel engine in order to reduce NO_x emissions significantly.

It is known that biodiesels have higher oxygen content, which improves combustion quality. However, this comes at the expense of higher NO_x emissions. For this reason, some diesel engines are designed with EGR systems which recirculate a certain proportion of the exhaust gas into the intake manifold. Since exhaust gas is used instead of fresh air, this reduces the amount oxygen available for combustion, which reduces the cylinder temperature and heat release during the combustion process and leads to lower NO_x emissions (Tsolakis et al., 2007). However, recirculating the exhaust gas into the intake manifold transports unburned inert gases back into the diesel engine, which reduces engine efficiency. For this reason, researchers have investigated the effects of EGR rate on the engine performance and emission parameters of diesel engines fueled with various types of biodiesels at different engine operating conditions. Qi et al. (2011), studied the effect EGR rate in a diesel engine, and it can be observed that the BSFC increases whereas the NO_x emissions decrease when the EGR rate is increased for the soybean biodiesel. Hussain et al. (2012), have been carried out an experiment to investigate the effect of EGR on performance and emissions in a three cylinders, air cooled and constant speed direct injection diesel engine. They mainly focus on different EGR rate. They were measured the emission of hydrocarbons (HC), NO_x, carbon monoxide (CO), exhaust gas temperature, and smoke opacity and also calculated the performance parameter such as thermal efficiency and brake specific fuel consumption (BSFC). They concluded that thermal efficiency is slightly decreased and BSFC is increased with EGR. Exhaust gas temperature is decreased with EGR, and NO_x emission decreases significantly. They observed that 15% EGR rate is found to be effective to reduce NO_x emission substantially without deteriorating engine performance in terms of thermal efficiency, BSFC, and emissions. EGR can be applied to diesel engine without sacrificing its efficiency and fuel economy and NO_x reduction can thus be achieved. The increase in CO, HC, and PM emissions can be reduced by using exhaust after-treatment techniques, such as diesel oxidation catalysts (DOCs) and soot traps. Hawi et al.

(2015), worked on the effect of exhaust gas recirculation on performance and emission characteristics of a diesel-piloted biogas engine, with the focus on reduction of harmful exhaust emissions while maintaining high thermal efficiency. The EGR system was developed and tested with different EGR percentages, *i.e.* 0%, 10%, 20% and 30%. The results showed that EGR led to a decrease in specific fuel consumption and an increase in brake thermal efficiency. From experimental results it was found that an EGR percentage of around 20% resulted in maximum BTE and minimum BSFC. Brake thermal efficiency increased at low EGR ratios due to the recirculation of active radicals from EGR that made the combustion process to be enhanced, so resulting in an improvement in brake thermal efficiency. It was found that EGR reduced the exhaust gas temperature by up to 7.6% at minimum engine loading and up to 2.3% at maximum loading. EGR caused a slight increase in Carbon monoxide (CO) and Hydrocarbon (HC) emissions. Increasing EGR reduced the amount of oxygen and led to incomplete combustion and therefore increased CO and HC emission due to lower combustion temperature. Bedar et al. (2016), investigated the effect of exhaust gas recirculation (EGR) on diesel engine using Simarouba glauca biodiesel blends B10, B20 with EGR rates of 10%, 15%, and 20% for different load conditions. Parameters like brake thermal efficiency (BTE), nitrogen oxides (NO_x), carbon monoxide (CO), hydrocarbons (HC) and smoke opacity were evaluated from the experimental study. The results show that, the BTE improves in lower concentration of the blend (B10), but for higher concentration (B20) it is almost equal to the neat diesel operation. With the application of EGR, BTHE improves for 10% and 15% EGR rate and starts declining with the application of 20% EGR rate. The CO, HC and opacity emissions decrease with the use of biodiesel blend but NO_x values increase. On the whole it is concluded that NO_x and other emissions is attained with simultaneous application of EGR (15%) and biodiesel blend (B10) without compromising engine performance. Agarwal et al. (2011), investigated the effects EGR rates on twin cylinder direct injection air cooled diesel engine running at 1500 rpm. It was found that BTE increased slightly at lower loads with EGR due to re-burning of HCs and is unaffected at higher loads. Reduction in NO_x and exhaust gas temperature were observed but emissions of particulate matter (PM), HC, and CO were found to have increased with usage of EGR. The engine was operated for 96 hour (hr) in normal running conditions and the deposits on vital engine parts were assessed. The engine was again operated for 96 hr with EGR and similar observations were recorded. Higher carbon deposits were observed on the engine parts operating with EGR. Higher wear of piston rings was also

observed for engine operated with EGR. Saleh (2009), in his studies used the jojoba methyl ester (JME) as a renewable fuel in two cylinder diesel engine. The study mainly focused to quantify the efficiency of exhaust gas recirculation. The results showed that EGR is an effective technique for reducing NO_x emissions with JME fuel especially in light-duty diesel engines with a very little penalty of fuel economy. With the application of the EGR method, the CO and HC concentration in the engine-out emissions increased. For all operating conditions, a better trade-off between HC, CO and NO_x emissions can be attained within a limited EGR rate of 5–15% with very little economy penalty. Yasin et al. (2015), investigated the effects of exhaust gas recirculation (EGR) to evaluate the engine performance and exhaust emission particularly NO_x content of diesel engine fueled with palm-biodiesel. The tests were performed under a steady state condition where conventional diesel fuel was used as a baseline fuel. According to the experimental results, diesel engine operating with palm-biodiesel and EGR reduced the brake power output, decreased the engine torque, increased fuel consumption, decreased NO_x and absolute slight increment in other emissions include CO₂, CO, and particulate matters.

2.6 Thermodynamic and Thermoeconomic Analysis of Diesel Engine

In recent years, systematic efforts have been made by several research workers to use biodiesel as fuel engines. Studies on in internal combustion engines (ICEs) using alternative fuels are extensively based on the first law of thermodynamics (energy analysis). Although the first law analysis is sufficient for general performance computations, the insight provided by a second law analysis (exergy analysis) is invaluable in understanding the details of the overall thermodynamics of engine operation (Caton, 2000). Furthermore, without second law analysis, it is not possible to determine the energy degradation that causes a reduction in work output of the engine (Alkidas, 1988). The second law of thermodynamics provides a distinction between the quantity and quality of energy by taking into account the irreversibilities (Moran et al., 2010). Therefore, the application of exergy analysis to engineering system is very useful to provide quantitative information on irreversibilities and various exergy losses, and in recent decades, it has been widely applied to a diesel engine (Sezer and Bilgin, 2013). Zheng and Caton (2012), evaluated the energy and exergy distribution of various processes associated with injection timings and EGR levels in a low-temperature combustion diesel engine. Yasar (2008), performed a comparative first and second law analysis of a standard and a low heat rejection turbocharged diesel engine. Parlak et al. (2005),

experimentally investigated the effect of the thermal barrier coating on a turbocharged diesel engine performance and exergy potential of the exhaust gas. Özkan et al. (2013), investigated the effect of pre-injection timing on thermal and exergy efficiencies of a diesel engine.

There are relatively few studies have been carried out on energy and exergy analysis of diesel engines using neat biodiesel and biodiesel–diesel blends. Canakci and Hosoz (2006), conducted a comparative energy and exergy analysis of a turbocharged diesel engine using different biodiesel fuels and diesel fuel which tested 100% load and 1400 rpm. Misra et al. (2013), performed energy and exergy analyses in a single-cylinder, four-stroke diesel engine operated with diesel and palm biodiesel. They evaluated both the performance and the characteristics of the engine at 85% engine loading and finally concluded that exergetic efficiency is almost equal to energetic efficiency. Azoumah et al. (2009), evaluated the use of exergy analysis using exhaust experimental data to optimize the performance of a four-stroke two-cylinder diesel engine using two different biodiesel fuels (cottonseed and palm oils), and their blends with diesel operated in different engine loads. The results exhibited that exergy analysis combined with exhaust gas emissions analysis is a very useful method for determining the optimal engine load. Gokalp et al. (2008), conducted an experimental and computational first and second law analysis for a four cylinder diesel engine at 1600 rpm by using several fuels: diesel fuel, marine fuel, pure soy oil methyl ester and their mixtures in different percentages. Caliskan et al. (2009), have carried on a study of energy and exergy analyses in a turbocharged Diesel engine fueled with pure Diesel as a base fuel and two different biodiesel fuels (soybean oil methyl ester and high-oleic soybean oil methyl ester). Energy and exergy efficiencies were 40.5% and 37.8%, respectively, for both used biodiesel fuels. Sekmen and Yılbaşı (2011), employed the first and second law of thermodynamics to analyse the quantity and quality of energy in a diesel engine operated with diesel fuel and soybean oil biodiesel at constant engine speed. Tat (2011), has experimentally analyzed the cetane number effect on the energetic and exergetic efficiency of a diesel engine fuelled with biodiesel. Rath et al. (2014), applied the energy and exergy analysis on a single cylinder four-stroke diesel engine using karanja methyl ester under varying compression ratios. Recently, there are only few studies about the economic analysis of diesel engines. Meisami and Ajam (2015), performed energy, exergy and economic analysis of a diesel engine using blends of diesel and castor oil methyl ester at full load condition and a constant speed of 1700 rpm. They concluded that among various biodiesel blends, using a 15% blend of castor oil biodiesel with diesel fuel results in an

optimum performance condition. In this case, the engine power slightly decreases, and both the thermal efficiency and the exergetic efficiency increase. The results of emissions economic analysis showed that all biodiesel blends were more affordable than Diesel fuel. The results of full economic analysis (emissions, fuel cost, fuel consumption and engine power loss) showed that only the 5% biodiesel–diesel blend was more affordable compared to diesel. Meisami et al. (2018), studied the thermo-economic analysis of a four cylinder direct-injection diesel engine while using biodiesel (waste cooking oil) and its blends with diesel fuel. The combustion reactions were determined using experimental data and, finally, the thermo-economic analysis was performed for steady-state control volume of the engine by using and solving energy, exergy and economic balances. The results showed that the best thermodynamic condition of the engine was achieved when using pure diesel. The most economical and costly operating conditions occurred when using 50% and 10% biodiesel–diesel by volume. From the literature review, it can be seen that there is a scarcity of the studies based on energetic and exergetic analysis together with economic evaluation for CI engines running on biodiesel–diesel fuel blends.

2.7 Limitations of the previous research

An effort has been made to comprise all the important contributions and highlighting the most pertinent literature available for investigating the feedstocks of biodiesel. From the exhaustive literature survey, the following important limitations are summarized as follows: -

- In the existing literatures, it is described that numerous studies have been conducted in different types of engines with a variety of low blend ratios of biodiesel. Only limited literatures are available on the high biodiesel blends ratios with diesel in a diesel engine; however, it has worsened the performance parameters and NO_x emissions. The performance, combustion and emission characteristics of different preheated biodiesel, discussed in most of the studies and discussion on engine characteristics with preheated biodiesel/diesel blends are inadequate.
- The emission constituents such as carbon monoxide, unburnt hydrocarbon and NO_x levels of diesel engine using high blend ratios preheated biodiesel/diesel are not discussed completely.
- The effect of intake air preheating addition with preheated biodiesel/diesel blends on performance, emission and combustion parameters seems lacking.
- The effect of parameters on the performance of the engine with emission and combustion characteristics of the low biodiesel blends has been emphasized in many studies. It has been

distinguished that the study on variable compression ratio, fuel injection pressure and fuel injection timing on variable compression ratio compression ignition engine using preheated biodiesel blends is limited.

- Very few experimental works have been carried out using preheated biodiesel blend with diesel and analyzed performance and emission characteristics of a diesel engine. The studies on combine effect of compression ratio, fuel injection pressure and timing engine using preheated biodiesel blends with intake air preheating mode are limited.
- The use of exhaust emission treatment operations of internal combustion engines is a more recent progress. The effect of EGR rates on the NO_x emission characteristics of the biodiesel has been studied and analyzed with the aid of biodiesel/diesel blends. However, it has to be noted that the study on variable compression ratio engine using EGR system is limited.
- The combined effects of fuel and intake air preheating addition with EGR rates on variable compression ratio direct injection compression ignition engine performance, emission and combustion parameters are not discussed completely.
- Most of the research work has been reported on energy and exergy analysis of diesel engine using biodiesel blends operated in natural aspirated mode. However, limited findings on thermodynamics and thermo-economic analysis for engine operated using a biodiesel blends or preheated biodiesel blends with diesel in a variable compression ratio direct injection compression ignition engine mode have been reported.

2.8 Research gap

The facts cited in relevant published articles have been analysed critically and the following salient features are found not being addressed properly.

- The literature reviewed shows that the study on variable compression ratio engine using preheated biodiesel blends is very much limited. In most of studies, the authors concentrate separately either the effect of compression ratio, effect injection pressure, and effect injection timing of load on engine performance. The effect of compression ratio, fuel injection pressure and injection timing with different EGR rates on engine performance parameters, emission and combustion characteristics of VCR DI compression ignition engine for different loading conditions is not studied exhaustively.

- The literature reviewed shows that the study on VCR DI CI engine using preheated biodiesel blends with intake air preheating mode is very much limited. In most of studies, the authors concentrate separately, the combined effect of fuel and intake air preheating with different operating conditions (CR, fuel IP and IT) on engine performance parameters, emission and combustion characteristics of VCR DI compression ignition engine at fixed engine speed and loading conditions is not studied exhaustively.
- The combustion parameters (heat release rate, cylinder pressure and ignition delay) of VCR DI CI engine using fuel and intake air preheating modes are also not considered extensively with combinations of CR-IP-IT at rated EGR rate and loading conditions.
- Few studies have been performed on the effects of various biodiesels blends with/without air preheating mode on engine performance, emissions and combustion characteristics. However, in those studies, less attention has been paid to the thermodynamics and thermo-economics of VCR DI CI engine processes with EGR system.
- A lot of research work has been reported on energy and exergy analysis of the diesel engine using variety mode of biodiesel such as blending with diesel, preheating, blending with ethanol and diesel etc. However, a limited work has been reported on economic analysis of biodiesel blends as fuel in a variable compression ignition engine.
- The use of numerical simulations in engine parameters prediction is also limited. Many of the researches have used simulation analysis for predication of engine parameters for diesel engines only. Prediction of performance characteristics of biodiesel fueled in a VCR CI engine with the aid of using software such as Computational Fluid Dynamics (CFD) simulation is an upcoming research. Experimentally investigating VCR engine characteristics is complex, time consuming and expensive, particularly for studies which use different blends.

From the literature review, it is clear that the study on a VCR DI CI engine fueled with higher percentage preheated palm oil methyl ester and diesel blends for performance, combustion and emission characteristics with intake air preheating mode of the CI engine is yet to be done. The study contributed to existing study research by including mainly the following points, which was often not done in prior studies. This study contributed to the current understanding of how fuel and engine modifications are vital for enhancing the engine characteristics of diesel engine operating with a high percentage blend ratios of biodiesel. It also showed how intake air preheating is so important for improving the combustion characteristics of a diesel engines. Moreover, the present

study contributed to control toxic NO_x emissions that were not much emphasized in existing literature in a VCR DI CI engines using biodiesel blends. Improvising additional instruments in the test bed engine experimental setup concerning the use of biodiesel fuel and intake air preheating. Thus, this study has been taken up with the preheated biodiesel blended ratios (up to preheated BD100). Acquiring knowledge of preparation of different blends of biodiesel with mineral diesel fuel, quality testing of all blends along with diesel fuel (such as density, viscosity, flash point, fire point, cetane number etc.). The engine testing (performance, combustion characteristics and exhaust emission analysis) with intake air preheating mode to obtain best preheated blends for diesel fuel without modification in diesel engine are assessed.

2.9 Scopes of Work

The compressive literature survey has indicated the presence of a significant amount work on the use of methyl ester oil biodiesel in compression ignition engines. Much of the methods used for biodiesel can also be used for diesel fuel as a results of similarities in properties of diesel. Even though the use of biodiesel as an alternative diesel fuel results in reduction of carbon monoxide (CO), unburnt hydrocarbon (HC), smoke and PM emissions, many researchers have reported, it affects the engine performance parameters (BSFC and BTHE) with a slightly higher NO_x emission. This is because of its diesel alien properties. Hence, there were very little attempts to use neat biodiesel or high blending ratios of biodiesel in compression ignition engines. The author's research work concerns the use of neat biodiesel in compression ignition engines. Including this, much of the literature reviewed on biodiesel has been used in CI engine for low blending ratios application. Thus, first it is essential to modify and tackle the behavior of the fuel properties of biodiesel effectively and instantly. The alteration of operating parameters (such as CR, IP, IT) of the engine can considerably improve the performance and NO_x exhaust emissions and in turn will lead to better engine efficiencies and low environmental impact. The NO_x emissions increase may limit the use of biodiesel fuels in a CI engine. It can be reduced with a direct application of EGR system in CI engine running with biodiesel and they have already been discussed in literature chapter 2. Hence, there is a scope of detailed research in this field.

1. Application of fuel modification techniques namely, blending biodiesel, preheating biodiesel and preheating biodiesel/diesel blends needs attention. It causes a significantly decreases fuel

properties of biodiesel (viscosity and density), and helps in better fuel flow and improved atomization. They are a good option as it makes the biodiesel less viscous.

2. The literature is almost quiet about the preheated biodiesel-diesel blends for fuel modification technique to improve the behavior of fuel properties of biodiesel problems.
3. Considering energy crises and pollution problems much pervious research works were used biodiesel with/without blending with in a CI engine as a substitute for diesel. But, the studies with intake air preheating mode to improve the operating range of the engine in blending biodiesel or preheated biodiesel/diesel blends fueled CI engines are limited.
4. A number of literature is found, in which methyl ester oil biodiesel is used in CI engine with/without blending with diesel. However, the conducting experiments with biodiesel in CI engine, for a variation of engine operating parameters (namely, CR, fuel IP, fuel IT, etc.) are lacking. Therefore, a study elaborating the consequence of load, CR, fuel IP, fuel IT, EGR rates and various possible compositions of these parameters can have uniqueness.
5. The literature is almost silent about the thermodynamic and thermo-economic analysis of preheated biodiesel/diesel blends fuel. Hence, first and second law analysis coupled with the variation of engine operating characteristics is essential to know about the maximum energy utilization and economic balances of preheated biodiesel blends fuel run engine.
6. Many researchers have used EGR in diesel engines to reduced NO_x emissions. But the studies on application of EGR in with/without blending with diesel or preheated biodiesel/diesel blends fueled CI engines are very limited. This studies show that the NO_x emissions can be reduced at middle volumetric percentages of EGR rates similar to those of diesel while the smoke can be kept low in biodiesel fueled engines.
7. The use of retarding injection timing, it effectively reduces NO_x emission but fuel consumption, CO, HC and smoke emissions increase considerably.
8. Most of the CI engine biodiesel with/without blending with diesel shows a reduction of engine thermal efficiency. The reasons are in addition to the higher kinematic viscosity biodiesel, engine running with biodiesel at the default operating parameters may not be optimum. Therefore, it is required to modify the engine design and operating parameters, specially CRs, fuel IPs and ITs (biodiesel), to enhance the engine performance parameters as well as lower ignition delay and exhaust emissions of CO and HC. The biodiesels having high Cetane number can be generate lower ignition delay (ID) than diesel. In this regard the use of preheated biodiesel/diesel blends is rare and can alternative fuel for CI engine.

9. The use of POME, and its preheating and blending with diesel as a main fuel in a variable compression ratio direct injection compression ignition engine with intake air preheating mode study is almost nil. Hence, there are enough scope to investigate POME and its effect of preheating and blending at various blend ratios as a fuel with variations engine operating parameters (CRs-IPs-ITs) in VCR DI CI engine. Especially, the exceptional emission quality of POME and preheated and blended fuel (lower carbon monoxide and unburnt hydrocarbon emission than diesel) can be coupled exhaust gas recirculation (EGR) with a reduction of NO_x formation (which is higher in case of POME than to diesel). Also added, in some countries, palm tree plantations in India are some of the major sources of employment and job providers, enhancement of energy security and maximization of waste utilization. Economic prospects for this fuel are not yet promising in India because of factors such as production cost and fuel economy. However, if research to reduce production cost is intensified, then energy produced by engines using palm oil are likely to become economically competitive with virtually with diesel as well as all other forms of non-edible biodiesels. More importantly, new markets opportunities can be developed whether local or overseas with more employment chances created, which could provide better living quality and standards for the public.

2.10 Summary

This chapter contains some significant research reported by the investigators by using different types of biodiesels (methyl esters). Later, the inferences are drawn on the summary of the details experimental works done in VCR diesel engine. The discussions are concentrated on the effects of biodiesel blending, preheated biodiesel, preheated biodiesel/diesel blends, intake air preheating using biodiesel blends, exhaust gas recirculation rates, engine load, engine operating parameters (CR, fuel IP and IT), and its variation on engine performance, combustion and emission parameters. Finally, the thermodynamic and thermo-economic analyses performed in diesel engines with various fuels (biodiesel and/or biodiesel-diesel blends). With reference to above extensive literature survey, there seem to be little work in the area of used preheated biodiesel oil blends as an alternative fuel in compression ignition (CI) engines with intake air preheating mode. This review would suggest that this area of investigation, whilst promising, requires further work in order to demonstrate the viability of different preheated blend ratios as a diesel oil substitute in CI engines. The motivation of this research investigation has been laid down as objective the thesis as highlighted in Section 1.6.

CHAPTER-3

Experimental Facility and Instrumentation

Overview

Before investigation on the use of biodiesel in a CI engine, it is essential to plan for the selection of engine and instrumentation. The knowledge and the methodology to be adopted in each technique and calculations for determining different parameters are also required. Similarly, modification of the engine has to be carried out for the investigation. In this regards, appropriate experimental test rig is developed to undertake a different experimental analysis of a VCR DI compression ignition engine running with different test fuels. The uses of non-edible crop of methyl ester biodiesels (preheating + blending) as fuel for CI engines have got popularity as alternative of diesel based fuel. However, owing to the reductions of higher viscosity and density of biodiesel, combined effects of biodiesel preheating and blending with diesel have not got proper attention. When combusted in CI engine, the proper adjustment of engine operating limits, namely, engine load, compression ratio (CR), fuel injection pressure (IP) and fuel injection timing (IT), with the presence of 'intake air preheating', may amend the performance of biodiesel (preheating + blending). Alongside, the variation of these operational parameters may lead to the optimum settings for which the performance of the compression ignition engine run by a fuel and intake air preheating at elevated temperature will be superior. In order to realize this fact, the preheating mechanisms of biodiesel and intake air are mounted in experimental test rig of the VCR diesel engine. This chapter describes the preparatory work carried out before the start of the investigation; experimental set up; test procedure and method of calculating the parameters. Prior to this study, the fuel properties of different neat biodiesels (COME and POME) were measured and compared with known international standards (ASTM D6751, EN14214 and IS) and mineral diesel fuel. Subsequently their performance were investigated under a standard engine operating conditions (CR 17.5, fuel IP 200 bar and IT 23° bTDC) to have initial baseline assessments (comparison).

3.1 Preface

The use of any alternative fuel in a variable compression ratio (VCR) direct injection (DI) compression ignition (CI) engine, requires proper selection of engine and instrumentations. An experimental test rig is developed to undertake the performance, combustion characteristics and emission evaluation of a variable compression ratio compression ignition engine run with diesel fuel and preheated biodiesel-diesel blend fuels. The knowledge and the methodology to be adopted in each technique and calculations for determining different parameters are also required. Similarly, modification of the engine has to be carried out for the investigation. Test rigs is suitably developed to conduct various test runs under different compositions of load, EGR rate, CR, IP and IT working conditions to evaluate and know the current thermal performance, combustion characteristics, and emission constituents of VCR direct injection diesel engine running with different test fuels. This chapter describes the preparatory work carried out before the start of the investigation, and is structured in following sections; experimental set up; instrumentations for measurements, experimental procedure and method of calculating the parameters.

3.2 The Experimental Setup

The experimental test rig consists of a single cylinder, four stroke, VCR DI CI engine. It is equipped with dynamometer as loading system, fuel supply system for both diesel fuel and biodiesel, water cooling and lubrication system, various sensors and instruments, integrated with computerized data acquisition system for online measurement of load, air and fuel flow rate, instantaneous cylinder pressure, injection pressure, position of crank angle and exhaust emissions. **Figure 3.1** represents the schematic representation of the experimental test setup. **Table 3.1** gives the technical specifications of different components used in the test rig. The setup enables the evaluation of performance parameters, combustion characteristics and emission constituents of the VCR engine. The thermal performance parameters include bp, BSFC, BTHE, exhaust gas temperature (EGT), volumetric efficiency, cylinder pressure, heat release rate (HRR), ignition delay (ID), heat equivalent of bp and exhaust gas. Commercially available LabVIEW based Engine Performance Analysis software package —Enginesoft LVI is used for online performance evaluation. The exhaust emissions of the engine are analyzed using an AVL DIGASS 444 exhaust gas analyzer.

Table 3.1: The specifications of the VCR diesel engine and its accessories.

System specifications	
Parameter	Specification
Product	Research engine test setup, Code 240
Engine Type	Single cylinder, four stroke, VCR DI diesel engine
Engine size, Bore × Stroke	87.5 mm × 110 mm
Engine Power and capacity	3.5 kW (@ 1500±50 rpm) 661 cc
Type of cooling	Water cooled
CR range	Fixed CR 17.5, Modified to VCR of CR ranges 12:1 to 18:1
Injection pressure and timing variation	200 bar, 0 – 25° bTDC
Combustion chamber	Hemispherical bowl in piston type
Dynamometer	Eddy current type, water cooled with loading unit
Air box	MS fabricated with orifice meter and manometer (100 - 0 - 100)
Fuel tank	Capacity 15 liter with measuring tube (0-450 ml)
Calorimeter	Pipe in pipe type
Rotameters	Engine cooling 40-400 lph, calorimeter 25-250 lph
Data acquisition Software	'Enginesoft' engine performance analysis software
Transmitters, sensors and indicators	
Fuel flow transmitter	DP transmitter, range 0-500 mm WC
Air flow transmitter	Pressure transmitter (-) 250 mm WC
Pressure sensors	Piezo type, range 5000 PSI, with low noise cable
Temperature sensors and transmitters	PT100 (RTD) type, range 0-100° C, output 4-20 mA (4 nos) K (ungrounded) type, range 0-1200° C, output 4-20 mA (2 nos)
Load sensor and indicator	Strain gauge type load cell with digital indicator, range 0-50 kg
Speed sensor and indicator	Strain gauge type load cell with digital indicator, range 0-50 kg
Speed sensor and indicator	Resolution 1°, range (5500 rpm) with TDC pulse
Data acquisition device	NI USB-6210, 16-bit, 250 kS/s
Setup constants	
Pulse per revolution	360°
No. of cycles	10
Fuel measuring interval	60 s
Speed scanning intervals	2000 ms
Bore x Stroke	87.5 mm x 110 mm
Capacity	661 cc
Orifice diameter	20 mm
Dynamometer arm length	185 mm
Connecting rod length	234 mm
Theoretical constants	
Orifice coefficient of discharge	0.6
Specific heat of exhaust gas	1.00 – 1.25 kJ/kg-K
Specific heat of water	4.186 kJ/kg-K
Density of Air	1.174 kg/m ³

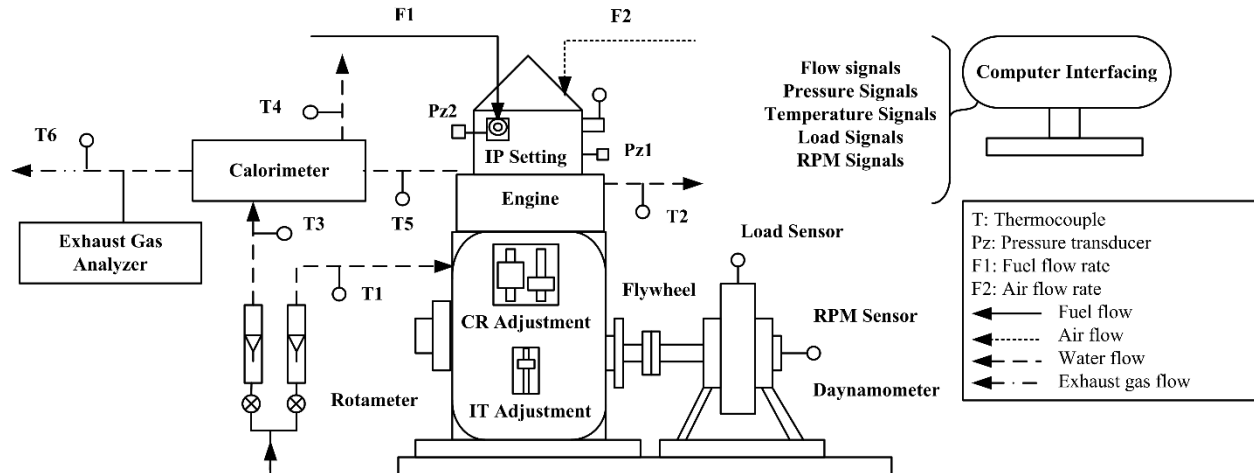


Figure 3.1: Schematic diagram of the original VCR engine experimental setup.

3.2.1 The VCR Compression Ignition Engine

In this investigation, the test engine consists of VCR based CI engine technology to adjust the compression ratio while the engine is in running conditions. This is primarily done to increase fuel efficiency under varying loads. Higher load requires lower ratios to be more efficient and vice versa. Variable compression engines allow for the volume above the piston at ‘Top dead center’ to be changed. Sir Harry Ricardo (1920) first built the variable compression ratio (VCR) engine for solving a knocking problem. Afterwards, various researchers tried to increase the range of compression ratios (Kemper, 1978; Ozcan and Yamin, 2008). It has been used mostly, for automobiles to have better output and lower emission at varying load condition (Grundy et al., 1976; Sobotowski et al., 1991). Nowadays, with the increasing trend of the use of alternative fuels, especially biodiesels, the details about their performance, combustion and emission behavior are unknown from the recent works with VCR engine (Jindal et al., 2010; Raheman and Ghadge, 2008; Selim, 2004). The variable compression ratio (VCR) diesel engine used to conduct the experiments is a single cylinder, four stroke, water cooled, direct injection engine. The technical specifications of the engine are given in **Table 3.1**. The engine is mounted on a stationary frame with a suitable cooling system. The lubricating system is inbuilt in the engine. The photographic images of the VCR DI engine are given in **Figure 3.2**. The fuel injector in the engine has three circular holes having 0.3 mm diameter, which injects fuel with a spray angle of 120°. The piston top of the engine is bowl type. Hence, the combustion chamber is of hemispherical type when piston reaches TDC. The liquid fuel reaches engine fuel pump from fuel tank by gravity.

The engine with fixed compression ratio can be modified by providing additional variable combustion space. There are different arrangements by which this modification can be achieved. Tilting cylinder block method is one of the arrangements which can be used to vary the combustion space volume. A photographic image of the tilting cylinder block installed on the engine cylinder is given in **Figure 3.3**. The engine is made to operate as a variable compression ratio (VCR) engine by providing a tilting block arrangement to suitably change the compression ratio (CR) to the desired value in the given range without stopping the engine and without altering the combustion chamber geometry. The VCR diesel has provisions for eight (8) step CR variation from 12 to 18. This is done by tilting cylinder head with the help of locknut and adjuster arrangement. However, the engine starting should be done at the standard CR (17.5) and later on CR change is done online. For setting a chosen compression ratio, the allen bolts are to be slightly loosened (refer **Figure 3.5**). Then, the lock nut on the adjuster is to be loosened and the adjuster is to be rotated to set a chosen compression ratio by referring to the compression ratio indicator and to be locked using lock nut. There are six (6) socket headed vertical allen bolts fitted on two supporting blocks on the two sides of the cylinder, which needed to be loosen for CR variation. Finally, all the allen bolts are to be tightened gently. The compression ratios considered for conducting the experiments are 15, 16, 17.5 and 18. The appropriate values of CR should be entered manually in the software for data acquisition. The basic principle of the tilting cylinder block assembly are as shown in **Figure 3.4** and **Figure 3.5**. When the CR is to be reduced the block is tilted so that the clearance volume increases and swept volume remains a constant.

3.2.2 Instrumentations for Measurements

The experimental setup is consisting of a several sensors, transmitters and indicators. These are interfaced with the data acquisition computer (DAC), established for automatic measurement of almost all of the direct and indirect performance parameters. Various measurement systems used to capture the experimental data used in the test rig are, load measurement, fuel injection pressure, cylinder pressure, engine emission and data acquisition system. The elementary measurement parameters are described in the following sections.



Figure 3.2: Variable compression ratio diesel engine.

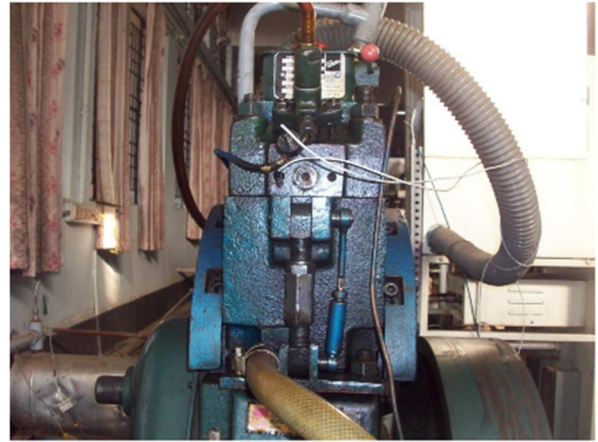


Figure 3.3: Tilting cylinder block arrangement.



Figure 3.4: Principle of tilting cylinder block assembly.

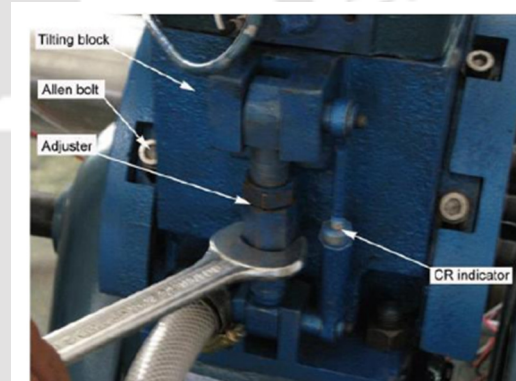


Figure 3.5: Compression ratio setting.

3.2.2.1 Load Measurement System

The experimental study is conducted at various loads and hence an accurate and reliable load measuring system is an essential requirement. The load measuring system of this experimental test rig consists of a dynamometer of eddy current type, a load cell of strain gauge type and a loading unit. The load is applied by supplying current to the dynamometer using a loading unit. The load applied to the engine is measured by a load cell. The test engine is connected to an eddy current water-cooled dynamometer to provide the load to the engine crankshaft in constant speed mode with the help of electromagnetic force (field current). The dynamometer was calibrated statically by applying a known current. The load sensor, fitted with the dynamometer, sends the load signal

to the digital display in kg. A dynamometer is a device, which is used for measuring force, torque or power produced by an engine. It can also be used to apply load or torque on the engine. The dynamometer used in this study is an eddy current type with a water cooling system. It provides an advantage of quicker rate of load change for rapid load setting. The VCR diesel engine is directly coupled to the eddy current dynamometer with a loading unit in which desired loads up to 12 kg can be applied.

3.2.2.2 Fuel Injection Pressure Measurement System

Fuel injection system admits the fuel into an internal combustion engine. Fuel injection pressure (also called as fuel inline pressure) is the pressure at which fuel is injected into the engine cylinder. In the present experimental study, a Piezo Sensor, (Make PCB Piezotronics), Model HSM111A22, Range 5000 psi (345 bar), is used to record the fuel injection pressure (refer **Figure 3.6-a**). Its location is indicated as No. 1 in the **Figure 3.6(b)**. Initially the default value of pressure is 200 bar. The fuel injection pressure can be adjusted to a desired point while the engine runs with or without a range given by the company to examine the performance, combustion characteristics and emission analysis of diesel engine. Fuel injection pressure setting can be done online when engine running. When we rotate the slotted nut to anti clockwise from the default setting the injection pressure decrease whereas clockwise rotation increases injection pressure. During experimentation, it is adjusted approximately for about 188 bar, 210 bar and 224 bar. The injection pressure is changed by adjusting the fuel injector spring tension, which is carried out by tightening or loosening the nut for higher, or lower injection pressures respectively (refer **Figure 3.7**). Important setting procedures are: Online diesel injection plot being displayed on the monitor using software, note the injection point displayed on the monitor, and turn the injection point-adjusting nut gradually and note its effect on diesel injection plot. The diesel injection plot shifts horizontally to retard/advance injection point depending upon the direction of rotation. Adjust the nut till desired injection point is obtained.

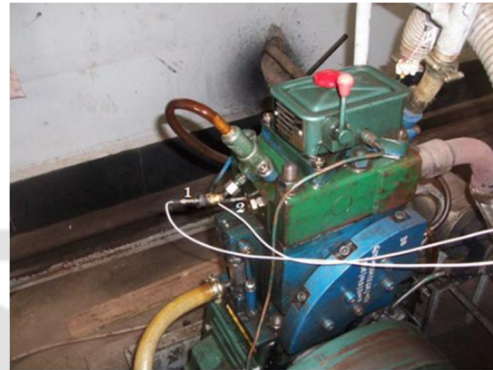
3.2.2.3 Air and Fuel Flow Measurement

Both air and fuel flow measurement can be performed manually and automatically. Manual airflow measurement is carried out by recording the difference in height of water column in the manometer. It is interconnected across the orifice meter, through which air comes into the engine panel box, before leaving towards the engine manifold. Manual measurement of fuel is executed

by transferring fuel from the tank through the measuring tube for known duration. Airflow transmitter (WIKA Instruments Ltd.) and differential pressure transmitter (Yokogawa Electrical Corporation) that are lined with DAC assess the automatic air and fuel flow amount.



(a) Piezo Sensor



(b) Location of fuel pressure sensors

Figure 3.6: Fuel pressure sensor: (a) Photographic view of piezosensor; (b) Location of fuel pressure sensor.

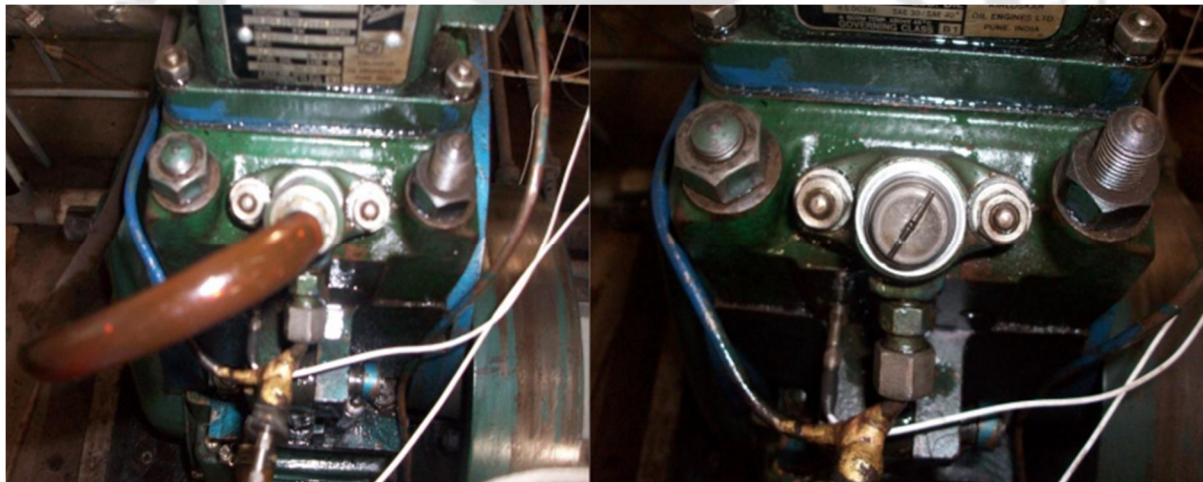


Figure 3.7: The nut adjustment for setting fuel injection pressure (IP).

3.2.2.4 *P- θ Measurement*

The PCB Piezotronics made two dynamic pressure sensors are fitted on the cylinder head and fuel injector. Both of them has identical specification and capable of distinguish pressure of compression, combustion, explosion, pulsation, cavitations, blast, pneumatic, hydraulic, fluidic

etc. An optical crank angle sensor (Kubler make) is used to measure each degree rotation of crank with TDC pulse.

3.2.2.5 Temperature Measurement

Four PT100 (RTD) temperature sensors measure the inlet and outlet temperatures of engine and calorimeter cooling water flow. The inlet and outlet temperatures of exhaust gas to calorimeter are measured by two K type thermocouples. All of these are interfaced with computer for automatic data recording. The thermocouples used in this work, have a response time more than 0.08 seconds (for the 1500 rpm constant speed engine). Hence, they cannot show the pulsation nature of the exhaust gas in the form of temperature readings and are found almost steady after a certain time (nearly 5 minutes) at a particular load.

3.2.2.6 Cylinder Pressure Measurement System

The cylinder pressure is measured using a Piezo sensor of Make PCB Piezotronics, Model HSM111A22, Range 5000 psi (345 bar), and diaphragm stainless steel and hermetically sealed type, by mounting it on the cylinder head. The piezo sensor is mounted on engine head. Its photographic view is shown **Figure 3.6(a)**, and its location is indicated as No. 2 in the **Figure 3.6(b)**. The piezoelectric transducer produces a charge output, which is proportional to the in-cylinder pressure. This charge output is supplied to a piezo powering unit (Make Cuadra model, AX-104). The piezo sensor consists of a quartz crystal. One end of the sensor is exposed to the cylinder pressure through the diaphragm. As the pressure inside the cylinder increases the crystal is compressed. Since the piezoelectric crystals have a tendency to generate electric charge when deformed, the sensor generates electric charge proportional to the pressure. The charge generated is smaller in magnitude and difficult to measure. Hence a charge amplifier is incorporated in the sensor to produce an output voltage proportional to the charge.

3.2.2.7 Injection Timing Variation Control

The IT of the liquid fuel can be tuned online and updated timing is learnt from the fuel pressure data at certain crank angle in the software. It is presumed that engine is running in diesel mode and on-line diesel injection plot is being displayed on the monitor using software. Note the injection point displayed on the monitor. Turn the injection point adjusting nut gradually and note its effect on diesel injection plot (refer **Figure 3.8**). The diesel injection plot shifts horizontally to

retard/advance injection point depending upon the direction of rotation. Adjust the nut till desired injection point is obtained. The injection point can be varied in the range of 0-25° bTDC. During experimentation it is adjusted approximately for about 19° bTDC (retard), 23° bTDC (standard or default) and 27° bTDC.

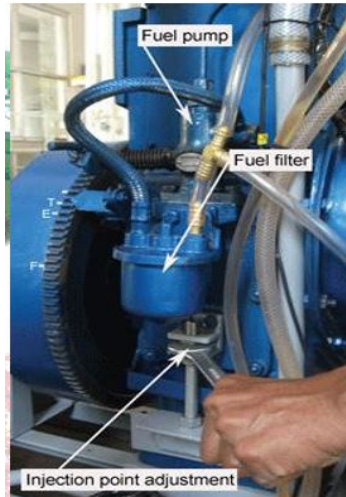


Figure 3.8: The nut adjustment for setting fuel injection timing (IT).



Figure 3.9: Photographic view of the exhaust gas analyser.

3.2.3 Emission Measurement

AVL DIGAS 444 flue gas analyzer was used to measure the emissions, HC, CO, CO₂ and NO emissions, and also oxygen from the engine exhaust. The HC, CO, CO₂ were measured with the help of sensors working on non-dispersive infrared (NDIR) principle and NO was measured with a photochemical sensor. The calibration of the instrument is performed automatically by measuring oxygen quantity in the air, each time it is restarted. The measurement is performed by letting the flue gas samples to surge through a probe in steady operation of the engine. A condensation trap is used next, to dry out the gas sucked. The samples are investigated inside the flue gas analyzer and return the values of CO, CO₂, O₂, NO_x and HC emissions on the display of control unit. A photographic view of the exhaust gas analyzer is shown in **Figure 3.9**. The detailed specifications of the AVL five gas analyzer are presented in **Table 3.2**.

3.2.4 Fuel Supply System

In the present investigation diesel fuel and vegetable origin biodiesel were used. First, standard mineral diesel fuel is used for reference as baseline (also cited in the texts as “BD0” for diesel mode) for comparison. Further, palm oil methyl ester “POME” and castor oil methyl ester “COME” (also cited in the texts as “PBD0” and “CBD0” for 100% biodiesel mode) has been considered. Two separate fuel tanks have been mounted for the fuel supply in the engine. The liquid fuels were supplied to the engine injection pump from the fuel tank under gravity feed. The fuel from the tank is connected to a calibrated glass burette by using a manually operated control valve. The fuel supply to engine from the fuel tank connects the burette outlet line from the engine. During this time, fuel is supplied directly to the engine as well as to fuel measuring burette. One-minute fuel consumption measurements were performed on a volumetric basis using graduated glass burette. The burette was filled with liquid fuel just prior to metering from the fuel tank. These measurements were converted into gravimetric basis by measuring fuel density. Subsequently, the data are recorded in the computerized control system. Baseline measurements were taken with 100% diesel for the experimental diesel engine. Then, for experiments, diesel, preheated biodiesel, preheated biodiesel-diesel blends were supplied to the engine for ignition purpose.

3.3 Engine Conversion Methodology

The design and development of various components was one of the major challenges to achieve desired experimental test rigs, that includes, fuel preheating system development, intake air preheating system development, and EGR system development with EGR cooler.

3.3.1 Fuel Preheater

The purpose of constructing a fuel preheater is to get an efficient method of heat transfer from engine exhaust gas to biodiesel by indirect contact. In this study, a thorough review has been done on this kind of techniques. The exhaust heat recovered from a diesel engine is used to preheat biodiesel in order to use it as an alternative for conventional diesel fuel. For this purpose, a helical coil tube heat exchanger designed and developed in-house, is chosen for fuel preheating (“biodiesel preheater” refer **Figure 3.10**). For the selections of helical coil heat exchanger, several heat exchangers can be used to recover heat depending on temperature of exhaust gas and its utilization. Helical coil heat exchanger was chosen taking three major factors into consideration:

Firstly, space in exhaust pipe is limited. Secondly, under the condition of laminar flow or low flow rates a shell and tube heat exchanger would become uneconomical because of the resulting low heat transfer coefficients. Thirdly, the heat transfer coefficient (h) for gases is generally several times lower than that for biodiesel oil and other liquids. Hence, the heat transfer surface for the exhaust gas needs to have a much larger area and be more compact to increase the exposure area and residence time for oil. On the coiled tube heat exchanger both ends of the coil will have control valves attached, namely, the inlet valve and the outlet valve. Since the viscosity of biodiesel is high and the tube diameter is small, so it would be economical and efficient to allow it to flow naturally on the basis of gravity or pressure difference. We will have to attach two-way control valves (v1 and v2) to maintain the required flow rate and to monitor the optimal amount of preheated biodiesel required to regulate the flow to graduated glass burette. The preheating device (refer **Figure 3.10**) consists of a helical copper tube aligned perpendicular for better heat exchange process and is placed in a shell maintained at a different heating temperature. A digital temperature-measuring device and thermocouples have been used to regulate the temperature. While designing a helical coil heat exchanger, the geometric effects of curvature ratio and pitch ratio have been considered, and the mathematical relationship used for the design of helical coil heat exchanger, the design parameters, important design steps and procedures mentioned in literatures (Patil et al., 1982; Salimpour, 2009), and discussed in Appendix-E.

Table 3.2: Exhaust gas analyzer specifications.

S.N	Measured Quality	Measuring range	Resolution	Accuracy
Voltage \approx 11-22 V DC; Power consumption \approx 25W; Warm up time \approx 7 min; Operating Temperature \approx 5-45°C; Dimension (W*D*H) = 270 x 320 x 85.				
1	CO	0 –10% vol	0.01%vol	< 0.6% vol: \pm 0.03% vol > 0.6 % vol \pm 5% of ind. value
2	HC	0 –20000 ppm vol	< 2000: 1ppm vol > 2000: 0ppm vol	< 200 ppm vol: \pm 10 ppm vol > 200 ppm vol: \pm 5% of ind. value
3	CO ₂	0 –20%vol	0.1% vol	< 10% vol \pm 0.5% vol > 10% vol \pm 5% vol
4	O ₂	0 –22% vol	0.01% vol	< 2% vol \pm 0.1% vol > 2% vol \pm 5% vol
5	NOx	0 –5000ppm vol	1 ppm vol	< 500 ppm vol: \pm 50ppm vol > 500ppm vol : \pm 10% of ind. value

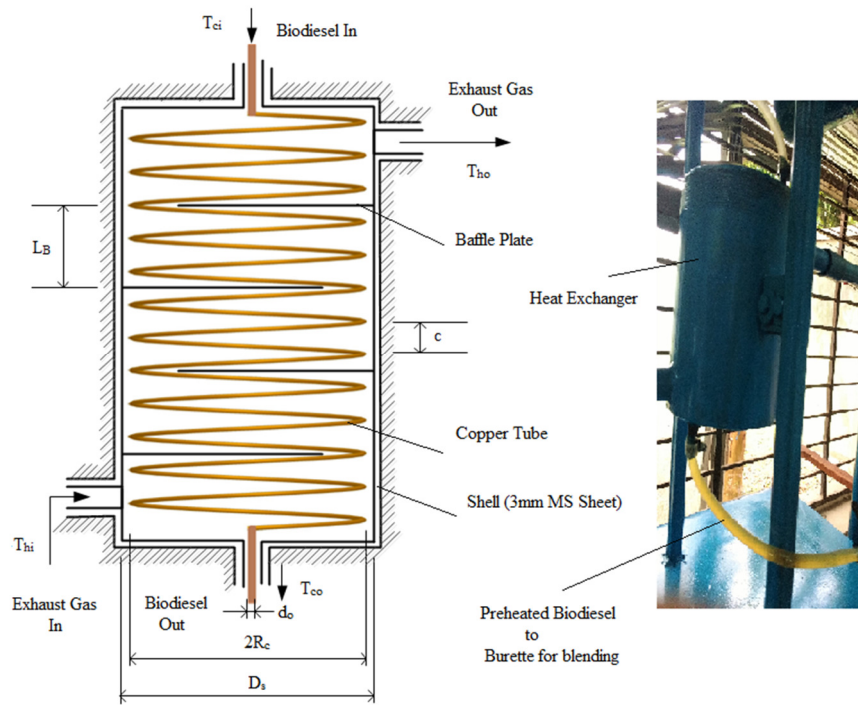


Figure 3.10: Helical coil heat exchanger-fuel preheater: (a) schematic layout diagram, (b) photographic image.

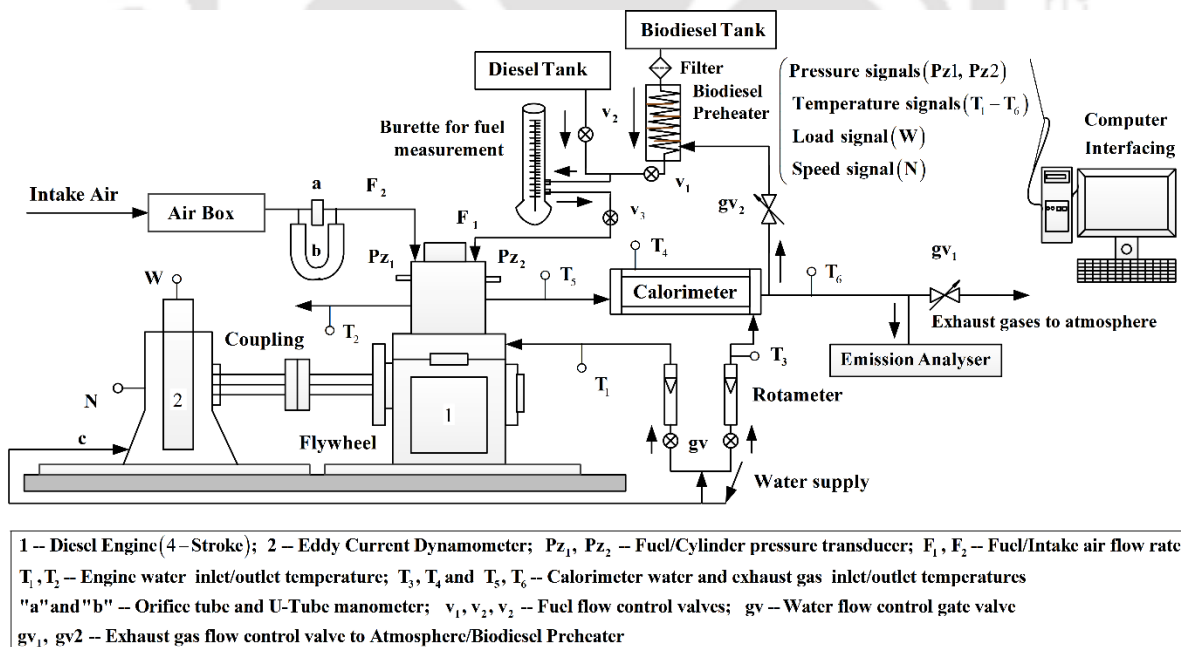


Figure 3.11: Schematic layout diagram of the modified experimental setup with fuel preheater.

The schematic layout diagram of the modified experimental setup comprising fabricated helically coiled heat exchanger is shown in **Figure 3.11**. The setup is a well instrumented heat exchanger system in which a biodiesel (hot flowing inside the coiled tube) is heated by a hot exhaust gas stream flowing in the shell side. The detailed dimensional parameters of the helical coiled heat exchanger are presented in Appendix-E, **Table E.1**.

3.3.2 Intake Air Preheater

An air preheater is a general term to describe any device designed to heat air before it supplies into combustion chamber. Its primary objective is to increase the thermal efficiency of the process. The air is heated using an exhaust gas and is supplied from the intake manifold to engine cylinder. By heating air in an intake passage and a circulation passage to a sufficiently high temperature, high-temperature air flows into a combustion chamber during cranking and, therefore, vaporization of fuel is promoted. Air intake preheaters are installed in the intake manifold to preheat the combustion air to the required temperature for ignition of fuel. In this study, engine exhaust gas temperature was used to preheat the inlet air passing through a newly designed air preheating system. In inlet air preheating attachment, a counter flow heat exchanger was used to transfer heat from exhaust gases to inlet air. This type of engine preheating is cost-effective and good for the environment. In addition to producing faster start, heating air intake reduces white smoke ("cold emissions"), engine wear, and fuel consumption during start up. Lower temperature intake air leads to inadequate final compression temperature, increase in emission delay, and longer time between the injection of the fuel to ignition, local over-enrichment, incomplete combustion and high pressure gradients due to abrupt mixture conversion in the cylinder. These factors lead to knocking of the diesel engine, increase in emission of hydrocarbons in the exhaust gas leading to severe loading of the environment.

There are many types of heat exchanger based on their application. The type of heat exchanger here is a simple shell and tube, counter flow type heat exchanger, which is installed in the intake manifold. In intake air preheating attachment, a heat exchanger was used to transfer heat from exhaust gases to inlet air. The intake air preheater consists of a six baffle plates, single shell and five tubes. The schematic layout diagram and photographic view of intake air preheater "shell and tube heat exchanger" is illustrated in **Figure 3.12**. The cold fluid flows in the tubes, while the hot fluid flows through the shell. The thermal design involves the calculation of heat transfer

coefficients in the shell side and tube side, surface area and pressure drops on the shell side and tube side, and effectiveness. While designing shell and tube heat exchanger, the design parameters, important design steps and procedures mentioned in literatures (Cao, 2010; Hewitt et al., 1994), and discussed in Appendix-E. In preheating attachment, the exhaust gas inlet/outlet pipe and intake air inlet/outlet pipe were installed. These pipes were made of mild steel having thickness of 3 mm. The gate valve “gv3” (refer **Figure 3.13**) was used in the experimental setup, in order to control the direct passage of exhaust gases into the preheater. The modified experimental setup comprising intake air preheater “Hx2” and fuel preheater “Hx1” is shown in **Figure 3.13**. The detailed dimensional parameters and design procedures of the helical coiled heat exchanger are presented in Appendix-E, **Table E.1**.

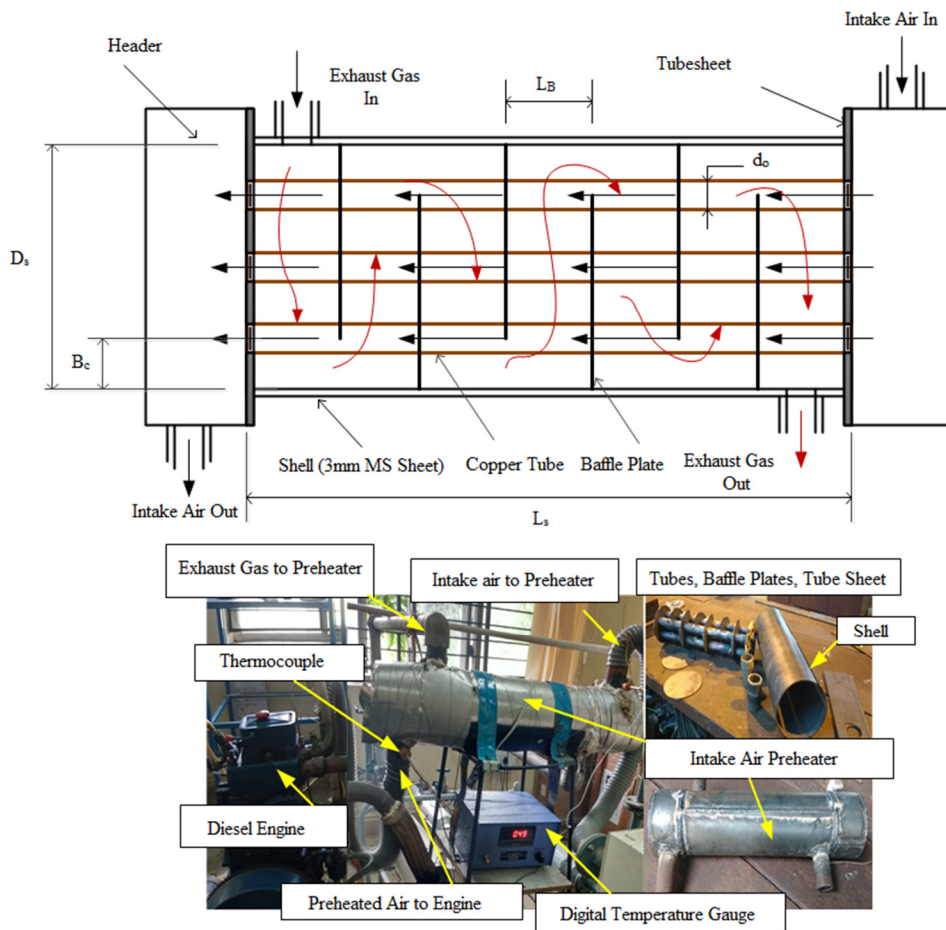


Figure 3.12: Intake air preheater: (a) the schematic diagram, (b) photographic view of preheater systems.

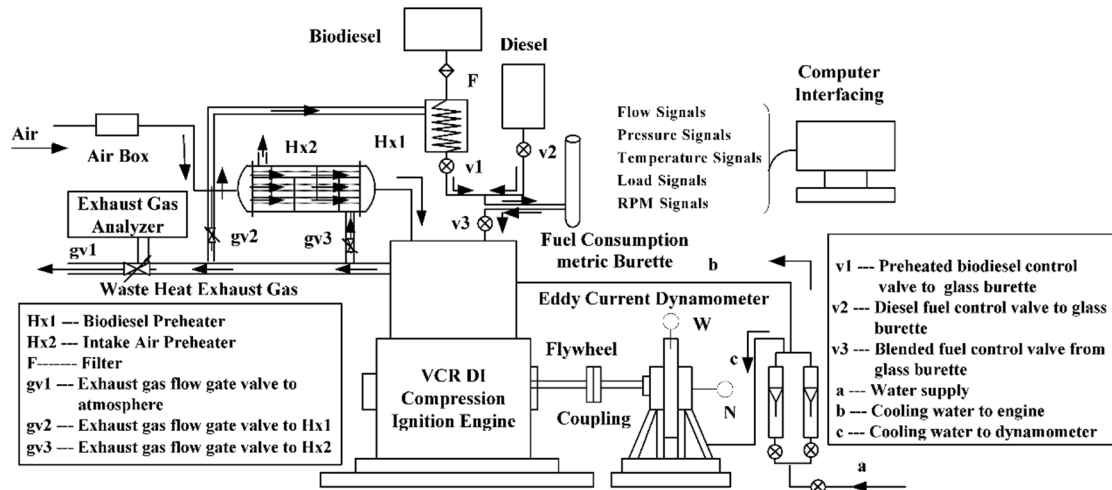


Figure 3.13: Schematic of diagram of experimental setup with fuel and intake air preheater.

3.3.3 Exhaust Gas Recirculation System

In diesel engines, EGR is nitrogen oxide (NO_x) emission reduction technique and works by recirculating a portion of an engine's exhaust gas back to the engine cylinders. The primary purpose of using EGR rate is to reduce oxides of nitrogen (NO_x) emission. It may be emphasized that EGR has no significant effect on engine performance parameters if the components work properly. The EGR system is built in the engine intake system for recirculation of exhaust gas to restrict the oxygen content in combustion chamber and control NO_x emission of the engine. Exhaust gases tapped from exhaust pipe, were fed to inlet airflow passage. The schematic diagram of exhaust gas recirculation (EGR) system is shown in **Figure 3.14**. The EGR system has valve and cooler. The valve was used to vary the recirculated flow rate. A simple heat exchanger was used to cool the drawn exhaust gas, which is to be circulated into the engine. Cooled EGR is used in order to reduce the thermal throttling. The EGR system is, a water-cooled system, being incorporated in which the exhaust gas passes in the tubes and water is circulated over these tubes. Thereby, the high exhaust gas temperature reduces but it is warmer than the intake air charge. The modified experimental setup comprising intake air preheater "Hx2", fuel preheater "Hx1" and EGR system is shown in **Figure 3.15**. The gate valve "gv4" is used to control the direct passage of exhaust gases to the EGR system.

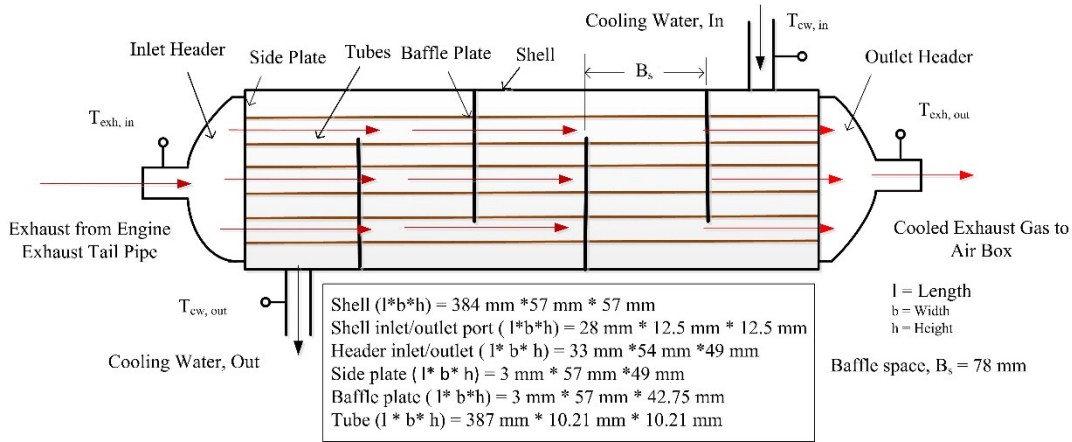


Figure 3.14: The layout diagram EGR system.

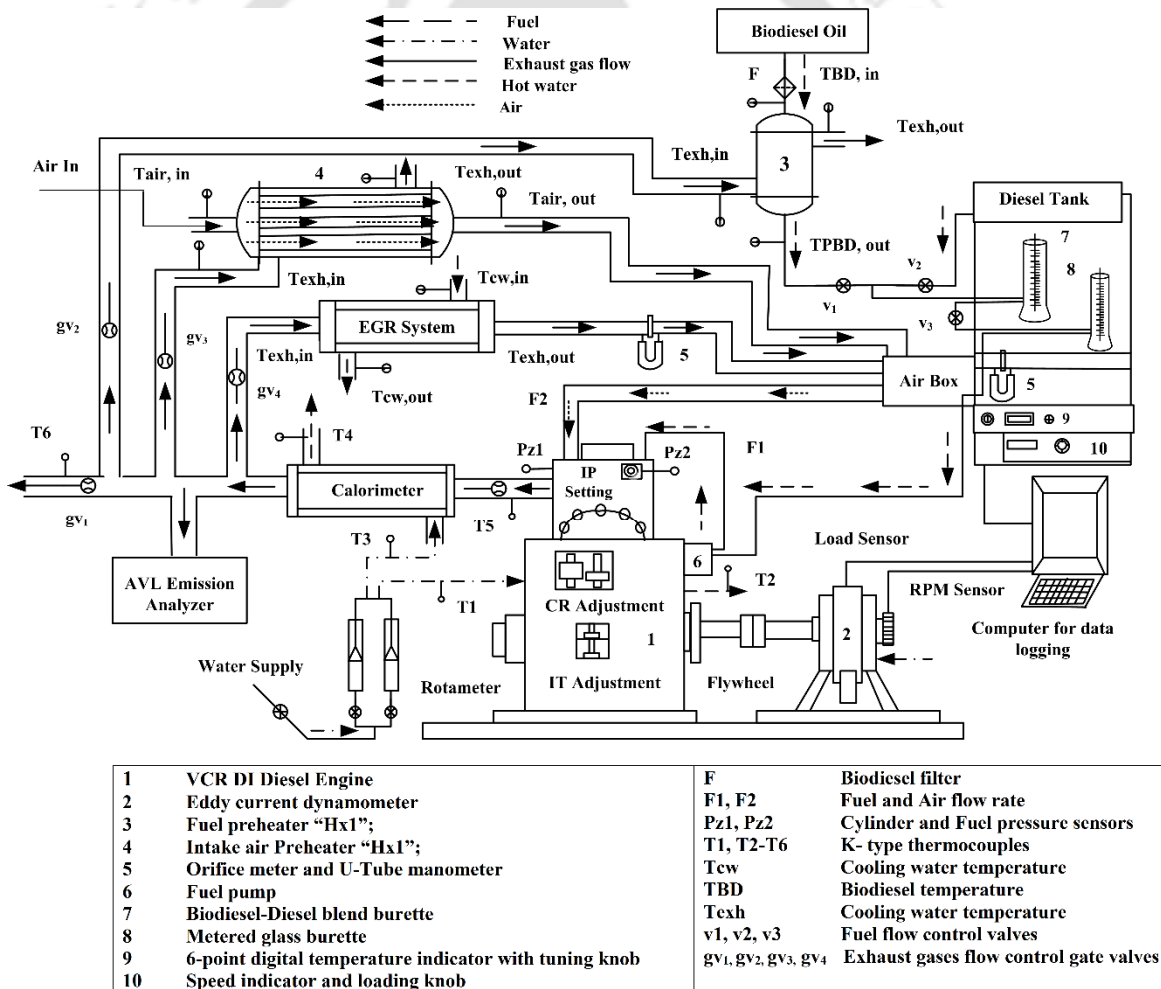


Figure 3.15: Schematic diagram of modified experimental setups (including fuel preheater, intake air preheater and EGR system).

3.4 Experimental Methodology and Procedure

From the physico–chemical properties of the biodiesel, it is understood that the viscosity and density of the COME and POME is higher than that of diesel fuel. In order to investigate the use of the biodiesel partially or fully as an alternative fuel in a VCR DI CI engine, its fuel and engine operating parameter modifications were adopted. The techniques adopted are discussed in the following subsections.

3.4.1 Methodology

The methodology adopted for carrying out the experimental work are as follows:

- Development of an experimental set-up with the necessary instruments to study the performance, emission and combustion characteristics.
- Conducting experiments in bassline mode using neat diesel operation for comparison.
- Design and fabrication of helical coiled heat exchanger for biodiesel preheating.
- Modifying the setup for fuel preheating device into the fuel injection system. Installation of graduated glass burette, fuel control valve and fuel line in the existing fuel injection system. In addition, the setup has also inbuilt with waste heat exhaust gas supply pipe line to a preheating device.
- Conducting initial experiments for various biodiesel in a diesel engine fuel preheating mode and to find out ranges of preheating temperatures of biodiesel. Subsequently examined its effect on fuel properties, and performance and emission characteristics of the engine and compare with that of neat diesel operation. Make a recommendation of the optimum fuel preheating temperature, which is maximizing performance parameters with less emission.
- Characterizations of fuel properties of preheated biodiesel at a different temperature.
- Installation of EGR system in the VCR engine. Then, conducting experiments in preheated biodiesel blends at various EGR flow rates and recommend the optimized EGR rate.
- Development and fabrication of intake air preheating (shell and tube heat exchanger).
- Then, modifying the setup and fix the intake air preheating device into the intake manifold system. Installation exhaust gas flow control valve and exhaust supply line in the device.

- Conducting initial experiments for preheated biodiesel blends in a diesel engine fuel and intake air preheating mode, and then find out ranges of preheating temperatures of intake air, subsequently examine its combined effect on engine characteristics and compare with that of neat diesel operation.
- Conducting major experiments using preheated biodiesel/diesel blends with intake air preheating mode for varying CR, fuel IP and fuel IT at 90% engine load and optimal 30% of EGR rate for investigations of engine characteristics. Recommend the suitable combinations of CR, fuel IP, and fuel IT.

3.4.2 Experimental Procedure

The present work is focused to test whether the biodiesel can be preheated and used as a blend with diesel in the engines. It is also intended to investigate the performance, emission and combustion characteristics of the engine using preheated biodiesel blends with diesel and intake air preheating using modified operating parameters of conditions of CR-IP-IT of VCR DI compression ignition. In this regards, the important experiments carried out in the entire thesis are mentioned in test matrix of the experiments in **Table 3.3**. Firstly, to establish a basis for comparison of results and to ensure the consistency of the experimental observations, baseline performance tests were carried out with the operating on diesel fuel only. Secondly, characterization the different blends of biodiesels (POME and COME with diesel), which are tested in a diesel engine characteristics investigation. Thirdly, characterization preheated neat biodiesels oils (POME and COME) at different heating temperatures, which are tested in a diesel engine characteristics investigation. Fourthly, characterization preheated biodiesel (POME) and blended with diesel at different blends ratios, which are tested in a diesel engine characteristics investigation. Fifth, the effect of preheating intake air at different preheating temperatures were investigated in a diesel engine characteristics investigation. Finally, the combined effects of fuel and intake air preheating is investigated in the VCR engine for various combinations of CR (16, 17.5 and 18), fuel IP (188, 200, 212 and 224 bar) and fuel IT (19, 23, 27° bTDC) at 90% of engine load with 30% EGR rate.

3.4.2.1 Investigations with the Baseline Tests

The experiment is performed to launch the foundation for comparison of results and to ensure the consistency of the experimental observations. The baseline performance tests were carried out

with the engine operating on mineral diesel fuel only. The variations of loads ranges from a minimum 'no load (0.1 kg) to maximum full load (12 kg) with a step of 2.4 kg. The engine tests were conducted for the entire load range i.e., 0% to 100% with an increments of 20%. The standard setting (CR=17.5, fuel IP = 200 bar and IT=23° bTDC) is maintained while operating the engine with diesel at constant speed of 1500±50 revolution per minute (rpm) thorough out the load range. Initially, the engine is allowed to run at no-load condition for a few minutes to warm up. The water flows are adjusted to 300 and 100 liters per hour for the engine cooling and calorimeter respectively according to the engine supplier instructions. Then, as per experimental design, a load level was set for engine operation. In each test, the operating conditions were stabilized, the engine was ready to present the baseline results. The variables that were continuously measured were recorded. For this, the following data were recorded manually (referring to **Figure 3.1**); dynamometer load, speed, temperatures (T1-T6), the difference in liquid level in the manometer for air flow (ΔH) and time required to volume of diesel fuel consumption 5 mL of fuel, cylinder and fuel pressure variation are automatically detected by the DAC. Then important performance and combustion characteristics parameters were calculated. All these data were saved in the appropriate format for further analysis. The samples of flue gas are then allowed to pass through the AVL 444 DIGAS flue gas analyzer probe, which quantifies the amount of CO, CO₂, HC, NO_x and O₂ emissions. Thereafter, the load level is raised as mentioned in the experimental matrix (**Table 3.3**) and the same processes as described above are repeated for subsequent load.

3.4.2.2 Investigations with the Fuel Blending

In this study the effect of biodiesel and diesel blending at different ratios (0–100% by volume) on some physical and chemical properties has been studied and presented. These properties include kinematic viscosity, density, cloud point, pour point, flash point, fire point, calorific value and cetane number. The fuel properties tests were conducted in a chemical laboratory under controlled temperatures and humidity to ensure accurate results. The first objective of this study was to characterize the properties of the castor oil methyl esters (COME) and palm oil methyl esters (POME), including the kinematic viscosity, density, cloud point, pour point, flash point, fire point, calorific value and cetane number. Furthermore, the properties of the COME and POME were compared with international biodiesel standards (ASTM D6751, EN14214 and IS15607). The COME and POME showed medium fuel properties to satisfy most biodiesel specifications. Only

the kinematic viscosity and density of the COME and POME, somewhat exceeds the upper limit of biodiesel specifications. Because they showed satisfactory biodiesel properties, the COME and POME was further blended with diesel for the applications of current mandates, including neat diesel fuel (BD0) and neat biodiesel (BD100), nine COME/POME–diesel blends were prepared by blending the COME/POME with diesel at proportions of (10–90% by volume) with 10% increments, corresponding to the (BD10–BD90) fuels, respectively. The second objective was to investigate the properties of diesel and COME, diesel and POME blends at different blending ratios (BD0-BD100) to determine the optimal blending proportion in comparison with the biodiesel–diesel blend specifications. The second objective of this study was to investigate engine performance and pollutant emissions in the use of COME/POME and diesel blends (BD0, BD10, BD20, BD30 and BD40) in a variable compression ratio (VCR), direct injection (DI) compression ignition (CI) engine in which the effect of blending was addressed.

3.4.2.3 Investigations with the Fuel Preheating

Preheating of COME/POME was carried out using fuel heating equipment, i.e helical coiled heat exchanger mounted just upstream of the fuel pump. After COME oil and POME oil characterization, a heat exchanger was designed and developed in the laboratory for heating the fuel using waste heat of exhaust gases of the engine. The developed helical coil heat exchangers have been built in to exhaust gas piping lines and as shown in **Figure 3.11**. Another fuel tank used for storing biodiesel and fuel supply system were developed and attached with prepared heat exchanger to preheat the biodiesel. A two-way switch was employed to change the fuel supply to the engine from diesel fuel to biodiesel while the engine was running. The developed preheating arrangement was attached to the engine for the completion of experimental set up. The main objective of this experimental study is to determine the effects of the kinematic viscosity and density of castor oil methyl ester and palm oil methyl ester (which is decreased by means of preheating process) on the performance parameters and exhaust emissions of a diesel engine. For this aim, biodiesel was preheated up to eight different temperatures (54°C – 138°C) with increment of 12°C, and its properties of biodiesels (COME and POME) were determined and studied offline for certain ranges of fuel preheating temperatures (test matrix-**Table 3.3**). These ranges of temperatures were determined from the initial experiments carried out in a diesel engine using in the fuel preheater of the existing experimental setup. The property values are compared with

respect to diesel as per international standards (ASTM D6751 and EN 14214). Then, tested in the diesel engine at full load conditions (12 kg) and speed of 1500 rpm. The main intention of this exercise is to have an initial assessment of fuel preheating temperature at which the best performance with lower emissions are achieved. During this phase of experiments with engine using preheated biodiesels were varied at different temperatures. Finally, the engine performance and emission parameter results were compared carefully, to make recommendation for the optimal/maximum fuel preheating temperature which is suitable for best engine performance and emission parameters of the engine.

3.4.2.4 Investigations with the Fuel Preheating and Blending

An alternate fuel supply system was developed with inbuilt heat exchanger to preheat the biodiesel oil and its blends. The developed preheating arrangement was attached to the engine for the completion of experimental set up. In this experimental investigation, the sequence of experimental tests involves the following procedures:

- (a) Biodiesel blends in different proportions on a volume of biodiesel from 0% to 100% in regular steps of 20% of castor oil methyl ester or palm oil methyl ester are prepared. The blends of fuel were denoted as BD0, PBD20, BD40, BD60, BD80, and BD100. The important fuel properties of different samples blend of fuels (preheated + blended) were experimentally evaluated/measured by laboratory. All the tests were conducted the biodiesel preheated at elevated fuel temperature of 114°C before blending with diesel. The blend was stirred well with the help of a mechanical agitator, to get a homogenous stable mixture. Thus, properties of the fuel were obtained by groups of tests are included the kinematic viscosity, specific weight (density), flashpoint, calorific value, cetane number, etc. based on the American Petroleum Institute (API) standard procedure. The device specifications and test procedure are in accordance to the American Society for Testing Materials (ASTM D6751), European union (EN14214) and Indian standard (IS15607).
- (b) The various blends of preheated biodiesel has been tested in a diesel engine at operating at standard (CR 17.5, IP =200 bar and IT = 23° bTDC) conditions. Preheating biodiesel was done before blending with diesel fuel, and blending was done in a glass burette based on the requirements of percentage fractions biodiesel ratio. The flow of biodiesel and diesel flow was controlled by manually operated two-way control valve 'v1' and 'v2', respectively (refer

Figure 3.13). Initially engine test run is done with diesel fuel (BD0) for baseline case data. Then, major experiments on diesel engine for the investigation and measurement of performance and emission characteristics were executed with different preheated blends of fuel at elevated (114°C) optimum fuel temperatures with variations of engine load (10– 90 %) with 20% increments, and compared the results with a baseline diesel data. Then, with a variation EGR rates (0–40 %) with 10% increments at full engine load condition, and compared the results with a baseline diesel data. Finally, blends of fuel were optimized for better engine performances and emissions characteristics.

3.4.2.5 Investigations with the Intake Air Preheating along with Preheated Blend Ratios

It is having different combinations using both preheated intake air and preheated blends of fuel as a fuel in CI engine, and its effect on the performance, combustion characteristics and emissions of compression ignition (CI) engine. In this investigation, first the effect of intake air preheating on diesel engine performance and emission parameters are examined with variation of intake air preheating temperature (33, 41, 49 and 61°C) using various preheated blends of fuel (114°C elevated fuel temperature) at 90% of engine loading condition (refer **Figure 3.15**). Thus, the required intake air preheating temperature was obtained thoroughly investigations diesel engine parameters results. All the tests were conducted at the standard operating of diesel engine (CR=17.5, fuel IP = 200 bar, and fuel IT = 23° bTDC) with a rated speed of 1500 rpm.

3.4.2.6 Investigation with the Modified Parameters (CRs-IPs-ITs)

The main objective is to investigate the impact of modified parameters i.e. varying CR, IT, IP. All the tests were conducted by starting the engine with diesel only. After the engine was warmed up, it was switched to the VCR operation herewith online modification of CR, IP, and IT variation. The various combinations of CR, IP, and IT are incorporated in test matrix (**Table 3.3**). Each tests were conducted at elevated temperature of fuel (114°C) and intake air (61°C) with constant 90 % (10.8 kg) load running conditions at 1500rpm. According to the company user manual, all the tests were conducted by starting the engine with diesel only at the standard CR of 17.5, IP of 200 bar, and IT of 23° bTDC. Then, according to requirement, the CR/IP/IT/ has to be changed into the intended test position for operation separately or combination. In each test, the operating conditions were stabilized and the variables that were continuously measured and recorded. The sequence of experimental tests involves the following:

- (a) The percentage change in performance, combustion and emission characteristics of various blends at varying CR (16, 17.5 and 18) as compared to diesel, and the best CR, which provides improved engine characteristics are identified.
- (b) In this section, impact of changing fuel IP (200, 212 and 224 bar) on the characteristics of diesel engine at different operating conditions have studied. The percentage change in performance, combustion and emission characteristics of various blends at varying IP with optimal CR as compared to diesel. Thus, the optimal fuel IP is also identified.
- (c) The percentage change in performance, combustion and emission characteristics of various blends at varying IT (19, 23 and 27° bTDC) at optimal CR and fuel IP is compared to diesel.
- (d) Finally, the best combination of CR-IP-IT that results is recommended best engine characteristics.

The experiments are performed maintaining the procedure described by Bureau of Indian Standard (BIS). The standards are described through IS 10000 (Part I to Part XIII) – 1980 (IS 10000). The performance and emission parameters are measured thrice as per experimental design for diesel, POME/COME biodiesel–diesel blends, preheated POME/COME biodiesel, preheated POME biodiesel/diesel blends and intake air preheating mode and averaged for each operating point. The average values of the recorded experimental data are employed for analysis purpose. The equations (A1–A15) used for performance and combustion analysis are provided in Appendix A. During the analysis of heat release rate, it is necessary to use the ratio of specific heats γ . The value of γ is essentially dependent on combustion temperature, which is known to be very non-linear and ranges amid 300 K to 1700 K. Further, it is very difficult to measure the trend of actual combustion temperature. Hence, traditional practice is to use a mean value of γ for the calculation of heat release rate. As quoted by (Heywood, 1988; Pundir, 2010) and many other researchers, the mean value of γ lies within 1.3 to 1.35 for single cylinder, CI, diesel engine of low to mid ratings. In this study, after multiple inspection, the value of γ is considered is 1.35. The justification of this is provided in Appendix A, with the help of ideal gas law and correlations provided by (Goering, 1998; Hansen, 2013) spatially averaged temperature in the combustion chamber. The heat release rate measured by variable γ and constant γ of 1.35 are well matched. All the experiments are executed within $20 \pm 2^\circ\text{C}$ and atmospheric condition.

Table 3.3: The test matrix for studying the engine characteristics of preheated biodiesel-diesel blends of fuel in VCR CI engine.

Major Work	Activities	Conditions					
Blending	Characterization of the properties of COME and POME Preparation different blends fuel and its fuel properties characterization	Prepared blends of fuels are: (BD0, BD10, BD20, BD30, BD40, BD50, BD60, BD70, BD80, BD90, BD100)					
	Performance and emission parameters evaluation using blends of fuel	Test fuels (BD0, BD20, BD30, and BD40) at standard (CR=17.5, fuel IP = 200 bar, fuel IT = 23° bTDC)					
Preheating	Characterization the properties of preheated neat COME/ POME.	Preheating biodiesel at different fuel temperatures (54 °C–138 °C).					
	Evaluation of diesel engine Performance and emission parameters using preheated biodiesel.	Preheating biodiesel at different fuel temperatures (54 °C–138 °C) at full load operating condition.					
Preheating and blending	Preparation of preheated blends of fuel and its fuel properties characterization.	Test fuels are: (BD0, PBD20, PBD40, PBD60, PBD80, PBD100)					
	It is also studied thermodynamics and thermos-economic analysis of preheated blends of fuel run in the diesel engine	Test fuels are: (BD0, PBD20, PBD40, PBD60, PBD80, PBD100) at standard operating conditions (CR=17.5, fuel IP = 200 bar, fuel IT = 23° bTDC) with a variation of loads (10%, 50% and 90%).					
	Investigations of engine performance, combustion characteristics and emission parameters using preheated blends of fuel.	Test fuels are: (BD0, PBD20, PBD40, PBD60, PBD80, PBD100) at standard operating conditions (CR=17.5, fuel IP = 200 bar, fuel IT = 23° bTDC) variation of engine loads (0–100%) with increments of 20% with no-EGR rate. And then, with variation of EGR rates (0%EGR–40%EGR) with increments of 10% with fuel engine load.					
Preheating blends of fuel and intake air	Investigations engine performance, combustion and emission characteristics using preheated biodiesel blends of fuel (114 °C).	Preheating intake air at different temperatures (33 °C–61°C) in steps 12 °C at full load condition.					
Modified operating parameters	Investigation the impact of modified parameters (CRs-IPs-ITs) on diesel engine characteristics using preheated blends of fuel and intake air at 90% load.	Fuel injection pressure (bar)	Fuel injection timing (° bTDC)				
		200	212	224	19	23	27
		CR (BD0):17.5	-	-	-	CR (BD0):17.5	-
		CR (PPBD20):16	CR (PPBD20):16	CR (PPBD20):16	CR (PPBD20):16	CR (PPBD20):16	CR (PPBD20):16
		CR (PPBD40):17.5	CR (PPBD40):17.5	CR (PPBD40):17.5	CR (PPBD40):17.5	CR (PPBD40):17.5	CR (PPBD40):17.5
CR (PPBD60):18	CR (PPBD60):18	CR (PPBD60):18	CR (PPBD60):18	CR (PPBD60):18	CR (PPBD60):18		

POME-Palm Oil Methyl Ester, COME–Castor Oil Methyl Ester, BD0- Neat diesel fuel, PPBD-Preheated palm biodiesel.

3.5 Analysis Procedure

The formula used in various performance parameters and combustion characteristics, thermodynamic and thermos-economic analysis, percentage of EGR rate calculations, and fuel and intake heat exchanger design for this analysis were illustrated in Appendix A, B, C and E. The dependent variables calculated from these parameters were analyzed and compared. Three performance parameters the BSFC, BTHE and exhaust gas temperature (EGT) were determined for diesel operation. The BSFC is a product of fuels density and for different fuels the BSFC of the engine will be different, while the BTHE is a product the viscosity and the calorific value of fuel at a particular load. The combustion parameters, namely ignition delay period, heat release rate and rate of pressures rise are highly depending on Cetane number of fuel as well as the calorific value. The emission parameter such as CO, CO₂, HC and NO_x, were directly obtained with the help of exhaust gas analyzer.

3.6 Uncertainty Analysis

Any experiment is not free from error. Errors may occur due to the instrument's inaccuracy, inappropriate calibration, human inadequacy etc. Therefore, uncertainty analysis mandatory for any experimental work. The uncertainty associated with both the baseline and other modes of experiments engine performance calculations were estimated based on methodologies of (Kline and McClintock, 1953). The details of each measured independent parameters and also, each performance parameter overall relative measurement errors were included in Appendix-D. However, the uncertainty of exhaust emissions errors were calculated based on ratio of resolution into ranges of each concentrations of emission elements. The procedure used for the uncertainty analysis is given in Appendix D.

3.7 Summary

This chapter includes the discussion about the engine experimental setup, preheating devices, equipment (fuel preheating, intake air preheating and EGR system), and experimental procedures required to accomplish the experiments. The brief specifications and the schematic diagram of the engine are included. The necessary equipments are designed, fabricated and added the base diesel engine setups for fuel and intake preheating modes. Some of the important specifications of the exhaust gas analyzer are also presented. The experimental mythology and procedures of the ways

adopted for diesel, preheated biodiesel-diesel blends test fuels are elaborated corresponding to the experimental matrix. For the sake of convenience, the details of base diesel and different biodiesel-diesel blend mode of operational procedures are elaborated separately. The analysis procedures and the uncertainties associated with the collected experimental data, for the whole engine, operations have been discussed. The later chapter discussed about the estimation of blend ratios and calculated performance and emission results of biodiesel–diesel blends run engine.



CHAPTER-4

Estimation of Optimum Blend Ratios of Biodiesel for Diesel Engine

Overview

Biodiesel is completely miscible with diesel and the blending in any proportion is possible in order to improve the fuel qualities. However, different chemical nature of biodiesel and diesel may cause differences in the physicochemical properties that will affect the engine performance and emissions. Therefore, the researchers had investigated and studied the quality of biodiesel blends in several aspects such as properties biodiesel and blending ratio. Currently, biodiesel blends are regarded as the most widely available alternative fuel in many countries. Blended biodiesel–diesel fuel has been approved as a commercial fuel at a low blending ratio. However, problems related to fuel properties are persistent at high blending ratios. Hence, in this study, the feasibility of biodiesel produced from castor oil methyl ester and palm oil methyl ester were investigated. Characterization of blended fuel properties with increasing biodiesel ratio is conducted to evaluate engine performance and emission test results. The qualifying of blended fuel properties was used to indicate the maximum blending ratio suitable for use in unmodified diesel engines according to the blended fuel standard ASTM D7467. In the present study, diesel (BD0) and various POME biodiesel/diesel blends (PBD10–PBD100) with 10% increment, and COME biodiesel/diesel blends (CBD10–CBD100) with 10% increment were used for characterization of the fuel properties. Finally, the experimental investigation has been carried out, to evaluate the performance and exhaust emissions of a VCR CI engine running with some mixture of blends POME (BD0, PBD10, PBD20, PBD30 and PBD40), and then COME biodiesel blends (BD0, CBD10, CBD20, CBD30 and CBD40) used as fuel respectively. The experimental works have been carried out on engine at default CR 17.5, 200 bar and 23 °bTDC of fuel IP and IT under full engine load. These experiments have been conducted for comparative analysis of performances (brake power, brake specific fuel consumption, brake thermal efficiency), and exhaust emissions (carbon monoxide (CO), carbon dioxide (CO₂), hydrocarbon (HC) and nitrogen oxides (NO_x) of biodiesel blends with diesel.

4.1 Preface

Currently, fuel energy content is one of the most important technical issues that indicates the use of blended biodiesel–diesel fuel at a high blending ratio, as the engine power output is influenced directly by fuel energy content (Karmakar et al., 2010). Typically, the energy content of biodiesel fuel is less than that of mineral diesel due to different chemical compositions, which affect the blended fuel energy content with increasing biodiesel fuel ratio in the blend with mineral diesel. The previous research work conducted to measure blended fuel energy content was limited and did not detail the measuring methods and equipment used. Moreover, research concerning the energy content of palm oil methyl ester (POME) and castor oil methyl ester (COME) biodiesels remains scarce. Blending of biodiesel with mineral diesel is the most common method for enhancing the biodiesel property and overcoming the fuel cold flow problems. Biodiesel from different feedstock's can be blended with mineral diesel for a substitute fuel for diesel engine. This blend is approved as a fuel for the existing diesel engines at low blending ratio up to 20% biodiesel (B20) according to the ASTM D7467 fuel standard specification (Balat, 2011; Biswas et al., 2010; Handling and Guide, 2009). Accordingly, it is used as a commercial fuel in many countries. However, at high blending ratio problems related to fuel properties are worse. The measurement and evaluation of blended fuel property is an important indicator for the maximum blending ratio of biodiesel from different sources that can meet the fuel specification requirements. Furthermore, it will be easier to analyze the engine performance results with increasing biodiesel ratio in the blend when the key properties of the used fuel are known. Biodiesel fuels properties differ from those of mineral diesel fuel, which means different combustion characteristics. Therefore, investigations of engine characteristics with increasing biodiesel ratio in the blend are needed before using in a diesel engine. In this study biodiesels (COME and POME) have been blended with mineral diesel in different volumetric percentage and properties of blended biodiesel have been evaluated against the relevant standards.

The aim of this study is to characterize the properties of POME biodiesel–diesel blended fuel and COME biodiesel–diesel blended fuel compared to the blended fuel standard ASTM D7467. Investigation of engine characteristics have been conducted with the blended fuel that meet the fuel standard specification. The engine test results with blended fuel are evaluated compared to the results of diesel fuel engine test as a standard fuel for comparison.

4.2 Methodology and Methods

4.2.1 Characteristics and Properties of Biodiesel

Biodiesel is evolving to be one of the most employed for partial replacement of petroleum based diesel fuel, especially in recent years. The most widely used non-edible feedstocks for biodiesel production are vegetable oils. However, biodiesel from edible feedstock crop are not widely used as alternative to diesel fuel in a diesel engine. Being a consumable oil, biodiesel from edible feedstock crop, they compete with food materials, thus it is not encouraging by the Governments of any nations for diesel engine application. In this work, biodiesel production from castor oil (non-edible feedstocks) and palm oil (edible feedstocks) has been chosen as the experimental fuels. It is obvious that palm oil in India, is widely used as a consumable oils and the Government of India is not encouraging biodiesel from this feedstock because they compete with food materials. The oils were imported, high price and impossible for fuel be used in a diesel engine. There are numerous investigations on non-edible product of biodiesels on diesel engine analysis, whereas, on a scientific level, there are not much exhaustive studies based palm oil methyl ester (biodiesel) on edible feedstock crops. However, in recent days, many private companies in India, are engaged as mass production of biodiesels from edible feedstock crops for alternative to diesel fuel in a diesel engine application. During these days, biodiesel from palm oil produced in India by different private biodiesel producer provide same price like other non-edible biodiesel resources. Hence, authors purposefully used edible product of biodiesel in a diesel engine to indicate opportunities palm oil methyl ester (POME) for substituting of diesel fuel in future in diesel engine application. Additional, in India there are no enough studies on POME in a diesel engine. Moreover, the characteristics of the components of palm oil methyl ester oil (POME) composition may plays a significant role for reductions of the exhaust emissions. It has a higher percentage compositions carbon and oxygen content with absence of soot and sulphur oxide as compared to fossil diesel fuel. These appearances may result to a fully combustion of injected fuel with reduce the level of emissions. On the other hand, among many non-edible feedstocks castor oil is one of the most widely used vegetable oil. It has certain interesting behaviours i.e. they are not suitable for human consumption, their usage as energy source does not compete with food production and its cultivation does not need high inputs. However, the major emphasis has been given on POME as a potential competitive fuel with respect to diesel in this present study. The performance parameter,

combustion and emission characteristics analysis considering a different operating and design conditions in a diesel engine.

The sample of castor oil methyl ester biodiesel (COME) and palm oil methyl ester biodiesel (POME) used in this study was purchased from a local industrial company SVM Agro Processor, Nagpur, India. **Figure 4.1** shows the sample of the pure biodiesel oil used in the experimental fuels. These Biodiesel should not be exposed to air in order to prevent it from oxidation, it was always stored in clean airtight a high-density plastics container to prevent contamination with a fast growing microbe. Moreover, storage conditions are important - heat, sunlight, and oxygen will also cause biodiesel to degrade more rapidly, so storage should minimize exposure to these conditions. The dissolved water in biodiesel can affect biodiesel storage time and can also cause problems, if the biodiesel is stored for more than a few months. This water can cause acids to form in the biodiesel/fuel, which can eventually eat a hole in the storage tank. Thus, it was stored maximum two weeks. A commercial fuel supplier provided diesel fuel that is available at public fuel stations in Guwahati, Assam, India. In order for biodiesel to be used commercially as a fuel, the finished biodiesel must be analyzed using sophisticated analytical equipment to ensure its international standards (**Table 4.1**). The density was measured using a test Hydrometer apparatus. Kinematic viscosity measurements were made with a Red wood viscometer. The calorific value was determined by a bomb calorimeter according to the standard ASTM D240. A Pensky martin's apparatus was used for the flashpoint and fire point measurements whereas, the test Petrotres apparatus was used for the cloud and pour point measurements. Initially, the biodiesel (POME- Palm oil methyl ester and COME- Castor oil methyl ester) was characterized by determining its viscosity, density, cetane number, cloud and pour points, flash and heating value according to standard methods. Fuel properties have a noticeable influence on engine characteristics. For this reason, the most important properties of tested fuels have been determined experimentally and the analyzed results are shown in **Table 4.2**. Those tests were conducted in the laboratory (Indian Institute of Technology Guwahati Department of chemical Engineering), under controlled temperatures and humidity to ensure accurate results. The test apparatus and methods are conformed to the strict ASTM procedures as recommended by manufacturers (refer **Table 4.1**). The testing was repeated five times and were carefully recorded from the digital apparatus. It can be seen from the table that the fuel properties are greatly influenced by the feedstock's of biodiesel. Finally, the various fuel properties of POME and COME biodiesel were compared well with biodiesel standards; ASTM D6751– American Society for Testing and Materials, EN 14214–

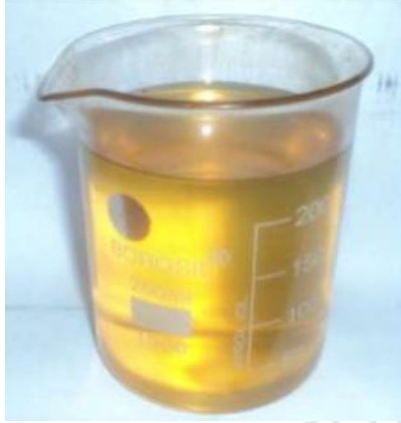
European union, and IS15607 – Indian biodiesel standards. American standard ASTM D751 identifies that the parameters of the pure biodiesel(B100) should fulfill before being used as a pure fuel or blended with diesel fuel. On other hand, European Union EN14214 and Indian IS15607 describes the minimum requirements for FAME (fatty acid methyl ester). Biodiesel (B100) specifications ASTM D6751, EN14214 and IS1507 standards are shown in **Table 4.2**.

Table 4.1: Measuring apparatus and standard test methods for measuring fuel properties.

Property	Measurement apparatus	Standard test method		
		ASTM	EN	IS1448
Density	Hydrometer	D941	ISO3675/12185	Part16
Viscosity	Red wood viscometer	D445	ISO3104	Part25
Cloud and pour point	Petrotres	D2500 and D97	-	Part10
Calorific value	Bomb calorimeter	D240	-	-
Flash and fire point	Penkys martins apparatus	D93	-	Part21
Cetane number	Ignition quality tester	D613	ISO51665	Part10

Table 4.2: The fuel properties of test fuels, and different standards “ASTM D6751, EN 14214 and IS 15607 properties (Saravanan, Nagarajan et al. 2010, Atabani, Silitonga et al. 2013, Silitonga, Masjuki et al. 2013).

Properties	Unit	Test fuels			Biodiesel standard Test limits		
		Diesel	COME	POME	ASTM D6751	EN 14214	IS15607
Kinematic viscosity at 40 °C	mm ² /s	3.23	21.91	8.71	1.9-6.0	3.5-5.0	2.5-6.0
Density	kg/m ³	838	943	946	880	860-900	860-900
Calorific value	MJ/kg	44.69	38.412	39.794	-	35	-
Flash point	°C	62	155	265	100–170	>120	>120
Fire point	°C	64	162	273	-	-	-
Cloud point	°C	1	7	9	-3 to -12	-	-
Pour point	°C	-8	4	5	-15 to -16	-	-
Cetane number, minimum	–	51	52.83	60.84	min. 47	min. 51	min. 51



(a)



(b)

Figure 4.1: Sample of methyl ester of castor oil and palm oil (a) COME, (b) POME.

4.2.1.1 Determination of Kinematic Viscosity

Viscosity refers to the thickness of the oil, and is determined by measuring the amount of time taken for a given measure of oil to pass through an orifice of a specified size. This is a critical property because it affects the behavior of injector lubrication and fuel injection. Fuels with low viscosity may not provide sufficient lubrication for the precision fit of fuel injection pumps, resulting in leakage or increased wear. Fuel atomization is also affected by fuel viscosity. Fuels with high viscosity leads to poor fuel atomization, can cause larger droplet sizes, poor vaporization, a narrower injection spray angle, and greater in-cylinder penetration of the fuel spray which can cause poor combustion, increased exhaust smoke and emissions (Tirado et al., 2010). A low viscosity can result in an excessive wear in injection pumps and power loss due to pump leakage whereas high viscosity may result in excessive pump resistance, filter blockage, high pressure, coarse atomization and low fuel delivery rates. The kinematic viscosity oil (ν) is defined as the ratio of dynamic viscosity to the density of oil. The kinematic viscosity of selected samples (a conventional diesel, neat biodiesel of Palm and Castor oil, and their blends with different blending ratios up to 100% with diesel fuel) was determined with the help of the redwood viscometer at elevated temperature as give below. Redwood viscometer is based on the principle of laminar flow under a falling head. Kinematic viscosity of a fuel was determined with the help of the redwood viscometer as give below (Madiwale and Bhojwani, 2017).

$$v = [0.0026(t) - 1.175/(t)] \times 100 \quad (4.1)$$

Where, v is the kinematic viscosity in cSt, t is the time in seconds required to collect a volume of 50 ml of liquid in a measuring flask.

The tested physical properties of the different biodiesels, and diesel fuel are shown in **Table 4.1**. Castor oil and Palm oil has a greater kinematic viscosity compared to mineral diesel. The kinematic viscosity of the POME and COME is approximately 2.69 and 6.78 times that of the diesel. At 40°C temperature, the kinematic viscosity of values of palm oil methyl ester (POME) and castor oil methyl ester (COME) biodiesel were 8.71 mm²/s and 21.91 mm²/s, respectively whereas diesel is 3.23 mm²/s. However, the ranges of kinematic viscosity values of biodiesels standards were between (1.9–6.0 mm²/s for ASTM D6751), (3.5–5.0 mm²/s for EN14214), and (2.5–6.0 mm²/s for IS15607). The kinematic viscosity of diesel fuel is 3.23 mm²/s within the ranges both standards. Thus, POME and COME have a high viscosity oil and a direct usage in a diesel engine may leads to poorer atomization of the fuel spray and less accurate operation of the fuel injectors. Thus the biodiesels have to be modified to lower the viscosity of the biodiesels as to make it easier to pump and atomize and achieve finer droplets.

4.2.1.2 *Determination of Density/ Specific Gravity*

Density is another important property of biodiesel. Fuel density affects the mass of fuel injected into the combustion chamber and thus, the air-fuel ratio. This is because fuel injection pumps meter fuel by volume not by mass and a denser fuel contains a greater mass in the same volume. Density is the mass per unit volume of any liquid at a given temperature. Specific gravity is the ratio of the density of a liquid to the density of water. Density has importance in diesel engine performance, since fuel injection operates on a volume metering system (Song, 2000). Determination of Specific Gravity (SG): Density bottle (volumetric cylinder) was used to determine the density of the oil. A clean and dry bottle of 50 ml capacity was weighed (W_0) and then filled with the oil, stopper inserted and reweighed to give (W_1). The oil was substituted with water after washing and drying the bottle. It is weighed to give the weight W_2 . The expression for specific gravity (Akpan et al., 2006), is:

$$SG = (W_1 - W_0) / (W_2 - W_0) \quad (4.2)$$

It is seen in **Table 4.1** that, the densities of COME and POME biodiesel fuels are generally higher than those of mineral diesel. The density of the POME and COME was approximately by 12.89% and 12.53% higher than that of the diesel, respectively. At 27°C, the density of POME and COME was 946 kg/m³ and 943 kg/m³, respectively whereas its value referred diesel is 838 kg/m³. However, the ranges of density limits of biodiesels standards are the range of (860-890 kg/m³ for EN14214) and (860-890 kg/m³ for IS15607). All the samples of pure biodiesel are out of in the limits of EN 14214 and IS15607 standard whereas, diesel fuel is in the range of all standards.

4.2.1.3 Determination of Flash Point and Fire Point

Flash point of a fuel is defined as the minimum temperature at which the fuel will ignite (flash) on application of an ignition source. The fire point of a fuel is defined as the temperature at which the fuel, when heated under specified conditions, will burn for at least five seconds (Kgathi et al., 2012). The flash point is also of importance in connection with legal requirements and for the safety precautions involved in fuel handling and storage, and is normally specified to meet insurance and fire regulations. A sample of the heated biodiesel kept in a close vessel are ignited. When the sample burns, the temperature is recorded. The pensky-martens cup tester measures the lowest temperature at which application of the test flame causes the vapor above the sample to ignite. The biodiesel is placed in a cup in such quantity as to just touch the prescribed mark on the interior of the cup. The cover is then fitted onto the position on the cup and Bunsen burner is used to supply heat to the apparatus at a rate of about 5°C per minute. During heating, the oil is constantly stirred. As the oil approaches its flashing, the injector burner is lighted and injected into the oil container after every 12 s intervals until a distinct flash is observed within the container. The temperature at which the flash occurred is then recorded, it is repeated three times and the average value is taken. It is seen in **Table 4.1** that, the measured flash point of POME and COME biodiesel are considerably higher than the prescribed limits (> 93 °C for ASTM D6751), and (> 120 °C for both EN14214 and IS15607). Minimum flash point temperatures are required for proper safety and handling of fuel. The flash point and fire point of palm oil methyl ester (POME) and castor oil methyl ester (COME) biodiesel was observed to be highest (265 and 273 °C), and (155 and 162 °C), respectively, whereas in a diesel fuel. it was observed to be lowest (62 °C and 64 °C).

4.2.1.4 Determination of Pour Point and Cloud Point

The lowest temperature of utility for petroleum products are indicated by cloud point and pour point. Cloud point is a test used to characterize the low temperature operability of diesel fuel. It defines the temperature at which a cloud or haze appears in the fuel under prescribed test conditions. The cloud point for biodiesel is generally higher than it is for diesel fuel. The pour point (PP) is defined as the temperature at which the fuel ceases to flow. The cloud point (CP) is the temperature at which a sample of the fuel starts to appear cloudy, indicating that wax crystals have begun to form which can clog the fuel lines and filters in a vehicle's fuel system. The sample is cooled in the cloud and pour point apparatus. The highest temperature at which haziness is observed (cloud point) or the lowest temperature at which movement of oil is observed (pour point), is reported as the test result. During execution of experiment, a sample of the biodiesel the test jar is placed inside a cooling bath. The temperature at the bottom of the test jar is the temperature at which the biodiesel starts to form cloud is taken as the cloud point. Beginning at 9°C above the expected pour point, and then below that, the samples were tested for pour point by slowly removing the sample from the bath, tilting to the side, and observing any movement of sample. The test was repeated until no movement of sample was observed upon holding the sample horizontally for 5 seconds. Then, a sample of the biodiesel is kept in the freezer to about 500 °C then placed in a heating mantle to melt. The temperature at the bottom of the test jar (i.e., the temperature at which the biodiesel starts to pour) is taken as the pour point. The test results shown in **Table 4.1** indicated that methyl ester biodiesel has higher cloud and point temperatures compared with diesel. The cloud point and pour point of palm oil methyl ester (POME) and castor oil methyl ester (COME) biodiesel was observed to be highest (9 and 5 °C), and (7 and 4 °C), respectively. In a diesel fuel, it was observed to be lowest with (1 and 0°C), respectively. The limits of cloud point and pour point ranges between (-3 to -12 °C) and (-15 to -16 °C), respectively.

4.2.1.5 Determination of Calorific Value

Calorific value of a fuel is defined as the amount of heat produced by the combustion of a unit value of fuel within the engine that enables the engine to do the useful work. It is an indicator of energy amount that can be harnessed from fuels. It is measured by means of combustion of the fuels in a bomb calorimeter (Aydogan, 2011; Cengel and Boles, 2008). Generally, methyl biodiesels have lower calorific values compared with mineral diesel fuel. **Table 4.1** also presents

the measured calorific values different test fuels. It is seen that; the diesel has the highest calorific value at 44.69 MJ/kg while the POME and COME biodiesel has relatively lower values with 39.794 MJ/kg and 38.412 MJ/kg, respectively.

4.2.1.6 Determination of Cetane Number

Cetane number is an important parameter in the quality of biodiesel fuel (Boerlage and Broeze, 1933). The Cetane Number (CN) indicates the natural tendency of the fuel to ignite. It is widely used as a fuel quality parameter related to the ignition delay time and combustion quality. The higher the CN, the better the ignition properties of the fuel. It ensures good cold-start properties and minimize the formation of white smoke (Kgathi et al., 2012; Kumar et al., 2012). This means that higher Cetane number causes the better performance of the engine and shorter ignition delay (the interval between injection and ignition) and finally the combustion is quiet and more uniform. A low CN causes a deterioration in this behavior and higher exhaust gas emissions (hydrocarbons and particulates). Generally, Cetane number (CN) is an inverse function of a fuel's ignition delay, and the time period between the start of injection and the first identifiable pressure increase during combustion of the fuel. In a particular diesel engine, higher cetane fuels will have shorter ignition delay periods than lower Cetane fuels. The Cetane number of the palm oil methyl ester (POME) and castor oil methyl ester (COME) biodiesel is significantly high when compared to mineral diesel. **Table 4.1** illustrates different Cetane number of the tested fuels. It is seen that; the diesel has the lowest cetane number at 51 while the POME and COME biodiesel has the higher values with 60.84 and 52.83, respectively.

4.2.2 The Ultimate and Proximate Analysis of Fuels

A characterization of biodiesel analysis is mainly defined by its “proximate and ultimate” analysis. Thermogravimetric analysis (TGA) is employed for proximate analysis and it discusses about the moisture content of the fuel (% M), the fixed carbon (% FC) and the ash content (% A). The fixed carbon (FC) of fuels are calculated using eqn. (4.3). The ultimate analysis gives the elemental composition of the COME, POME and diesel, usually in moisture and ash free basis. It is given in **Table 4.3**.

$$\% \text{ FC} = 100 - (\% \text{ M} + \% \text{ A}) \quad (4.3)$$

Then, percentage of oxygen can determine using eqn. (4.4) after the ash content obtained from the proximate analysis.

$$\% O = 100 - (\% C + \% H + \% N + \% S + \% A) \quad (4.4)$$

The chemical composition of the POME and COME indicates that the biodiesel has lower percentage of carbon, which was found by 12.93% and 18.18% lower that of diesel fuel, respectively. The hydrogen present in the POME and COME is 3.71 and 2.39 times less than that of diesel, respectively. It is also clear from the table that the POME and COME has considerable oxygen present. The results of the proximate and ultimate analysis are presented in **Table 4.3** below.

Table 4.3: The proximate and ultimate analysis of POME and COME biodiesels, and diesel fuel.

Fuel type	Proximate Analysis (%)			Ultimate Analysis (Elemental composition C/H/N/S/O) ~%				
	Moisture (M)	Ash (A)	*Fixed carbon (FC)	Carbon (as C)	Hydrogen (as H)	Nitrogen (as N)	Oxygen (as O)	Sulphur (as S)
Diesel	< 0.025	< 0.015	99.96	86.46	13.14	0.13	0.03	0.24
POME	< 0.1	< 0.05	> 99.85	75.28	3.54	1.51	16.5	3.12
COME	< 0.1	< 0.1	< 0.1	70.2	5.5	1.55	19.2	3.5

*Calculated value

4.2.3 Biodiesel Diesel Blending

A different chemical nature of biodiesel and diesel may cause difference in the physicochemical properties that will affect the engine performance and pollutant emissions. Therefore, in this study experimental works are carryout to examine the quality of biodiesel blend ratios that can be used in all diesel engines without modification. Characterizing the key properties of biodiesel diesel blends can assist the researchers who work on alternative fuels for diesel engines.

4.2.3.1 Blend Preparation

Blending means simple mixing & homogenizing the mixture of fuel and biodiesel. Being almost similar properties, it will get mixed very easily. Diesel fuel and biodiesel, with certain mixing percentage on volume basis, should be miscible in each other at the ambient conditions and this already exists as mixed fuels. Homogenizers and Ultra sonicators for effective blending of the biodiesel with petroleum diesel were used for effective mixing and to form homogeneous mixture.

The biodiesel was blended with diesel at heated state. In all cases, blends need to be confirmed with a density and kinematic viscosity check in order to ensure they meet specifications, the property values keep on rising with increasing the percentage ratio of biodiesel in the blend. However, preheating biodiesel blend with diesel caused the property values fall between international biodiesel blend standards. Samples of palm oil methyl ester/castor oil methyl ester with diesel were prepared through mixing and blending using electrical magnetic stirrer shown in **Figure 4.2**. Briefly, biodiesel was added to diesel at a low stirring rate. The mixture was stirred continuously for 15–20 min and left for 20–25 min at room temperature to reach equilibrium before subjected to any tests. All the fuel the subdivision the blending ratio is (10–100% by volume) with 10% increments (which corresponded to (CBD10, CBD20, CBD30, CBD40, CBD50, CBD60, CBD70, CBD80, CBD90 and CBD100 fuels, and PBD10, PBD20, PBD30, PBD40, PBD50, PBD60, PBD70, PBD80, PBD90, and PBD100 fuels) for COME and POME, respectively. The diesel fuel with Indian Baharat stage-III diesel rated is tested along with blends fuel. They are nominated as a reference standard fuel. In this study, the fuel blends were analyzed by measuring the viscosity, density, cloud point, pour point, flash point, cetane number and calorific value. Subsequently, they are compared with the blended biodiesel standards (ASTM D7467 and BIS) and diesel fuel. The physical test standard and specification of biodiesel diesel blends were included in **Table 4.4**.



Figure 4.2: Fuel preparation using magnetic stirrer.

4.2.4 Engine Test

In this study, the entire experiments were carried out at default company setting of CR 17.5, 200 bar fuel IP and 23° bTDC of fuel IT (Test matrix-**Table 3.3**). The emissions analyses have been measured using AVL 444 DIGAS analyzer (Indian made). Details of gas analyzer are shown in **Table 3.2**. To get the average values, all tests were repeated three times. Tests were conducted at a constant engine speed of 1500rpm and full engine loading condition. The engine was equipped with an exhaust gas recirculation (EGR) system, however, in this experiment the EGR mode was set to off. Investigation of engine performance and emission have been conducted with the blended fuel that meets the fuel standard specification. This investigation provides specific information on the optimum combination between engine performance and emission parameters with respect to the tested blend of fuels. The engine test results with blended fuel are evaluated compared to the results of diesel fuel engine test as a standard fuel for comparison.

4.3 Results and Discussion

4.3.1 Analysis of Biodiesel-Diesel Blend Properties

In this study, POME/COME biodiesel diesel blends (vol. 10%–vol. 90%) along neat mineral diesel fuel (BD0) and neat biodiesel (BD100) were prepared and presented in **Figure 4.3** to **Figure 4.8**. The results are presented and discussed in the following sections.

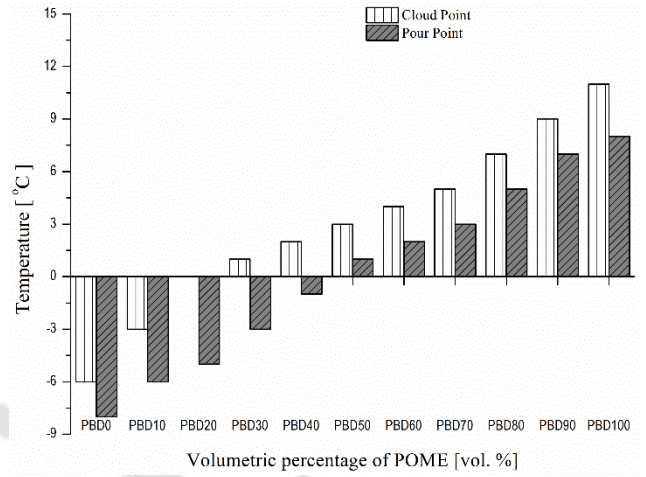
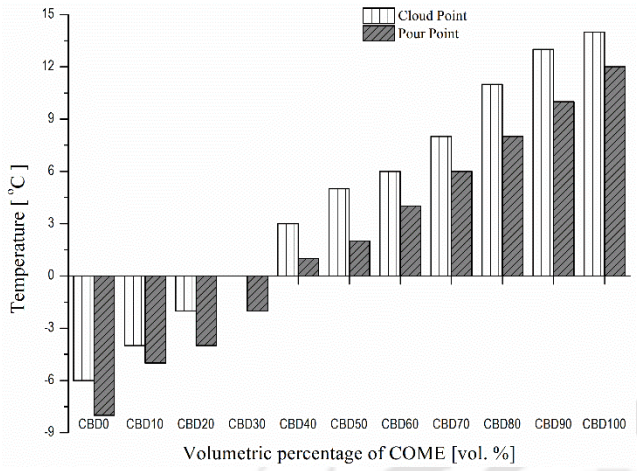
4.3.1.1 *Effect of Blending on Cloud Point and Pour Point*

The pour point is the lowest temperature used to characterize cold flow properties and the cloud point is the highest one. **Figure 4.3** demonstrates the variations of the cloud point and pour point with the volumetric percentage of the COME and POME diesel blended fuel, respectively. It indicates that, biodiesel blends have higher cloud point and pour points than diesel fuel and this is one of the most critical obstacles against the wide- spread of biodiesel usage (Demirbas, 2008; Morad et al., 2006). The fatty acid composition of biodiesel greatly influences the cloud point and pour point. Furthermore, the temperature gap between the cloud point and pour point is very narrow for COME due to the lack of chemical diversity in biodiesel fuels (Chiu et al., 2004). It is evident from these data that cloud point and pour point of the blends decrease with the increase quantity of diesel in the blend due to the synergistic interaction between fatty acid methyl ester and diesel molecules that affect the orientation of the molecular arrangement during crystallization

(Lim et al., 2009). The higher cloud point and pour point was found for CBD100 (14 and 12°C) and PBD100 (11 and 8°C). However, the blended fuels ratio 30% or less of cloud point and pour point was less than of the blended fuel standard requirements BIS (3 max for winter) at biodiesel blending ratios (\leq CBD30 and \leq PBD30). Thus, cloud point and pour point for biodiesel diesel blends (CBD10, CBD20, CBD30, and PB10, PBD20, PBD30) are suitable to be used and closer within the recommended specified limits for biodiesel diesel blending standards.

4.3.1.2 Effect of Blending on Flash Point and Fire Point

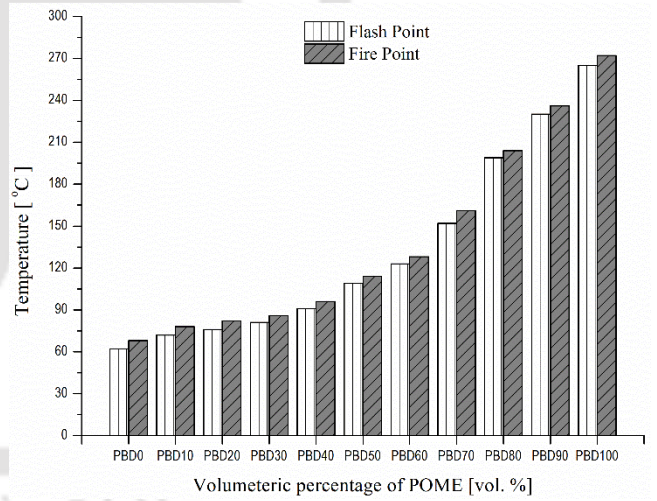
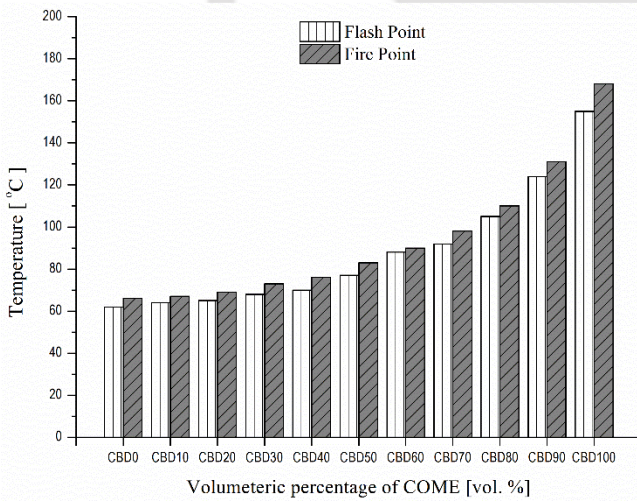
Flashpoint is the minimum temperature at which the vapor is given off by a fuel when heated will flash with a test flame held above the surface without the fuel catching fire. It plays a vital role when determining the fire hazard of the fuel. **Figure 4.4** shows that the flash point and fire point of all biodiesel diesel blends are higher than diesel. The flash point and fire point increases on increasing the percentage of biodiesel in diesel, as shown in **Table 4.4**. It is observed that the increasing quantity of COME and POME biodiesel oil in the diesel-biodiesel blends, due to the higher value of flash point of biodiesel oil, flash point of the blend, is found to rise. It is seen in **Figure 4.4** that, CBD10 and PBD10 have lowest flashpoint (67 and 64 °C) and fire point (78 and 72 °C). In contrast, CBD100 and PBD100 have highest flashpoint (168 and 155 °C) and fire point of (272 and 265 °C) which is relatively higher than the ASTM D7467 and BIS specified minimum of 52°C and 35°C, respectively. This shows that the main advantage of biodiesel diesel blends makes it possible to store and safer to handle for transportation sector. Therefore, the results of all biodiesel blends are considered to be safe to store for use in the engine (Agarwal and Das, 2001).



(a)

(b)

Figure 4.3: Blending effect on cloud and pour point: (a) COME-diesel blends (b) POME-diesel blends.



(a)

(b)

Figure 4.4: Blending effect on flash and fire point: (a) COME-diesel blends (b) POME-diesel blends.

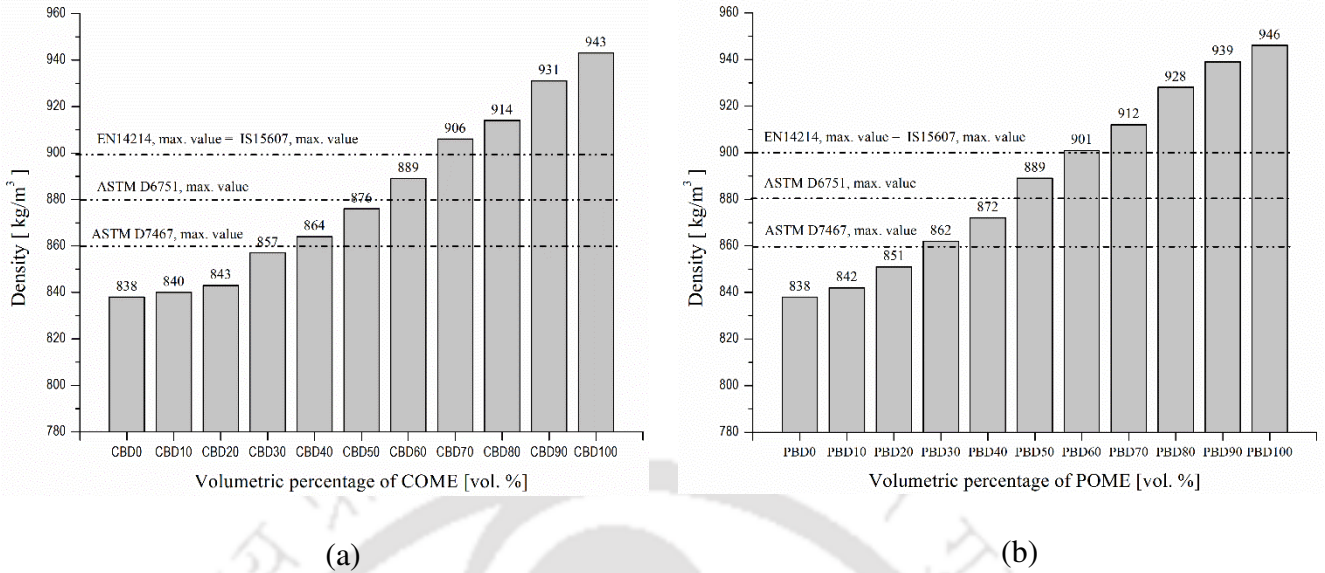


Figure 4.5: Blending effect on density: (a) COME-diesel blends (b) POME-diesel blends.

4.3.1.3 Effect of Blending on Density

The fuel density is an important parameter for various aspects of diesel engine performance, as the high biodiesel fuel density can influence the engine power output due to the difference in fuel injected mass, where the fuel is measured by volume in a fuel injection system (Alptekin and Canakci, 2009). The lower value of the density is desirable to obtain the maximum engine power through the fuel flow control in the injection pump. The density of a fuel influences its fuel consumption. Thus, it is preferred to have a less denser fuel whose consumption would be less. From the test results it is found that the density of the CBD100 and PBD100 are the highest 943 kg/m³ and 946 kg/m³ as shown in **Figure 4.5**. It is clearly shown that the rising of biodiesel content in the fuel blend will increase the density of the fuel. The density recorded for diesel is 838 kg/m³. It is obvious that the rise of the biodiesel content in the blended mixture increases the density of the blended fuel (**Figure 4.5** and **Table 4.4**). It is seen that, the ratio blending up to 30% of COME and POME biodiesel diesel blending (CBD10, CBD20, CBD30) and (PBD10, PBD20, PBD30) were very close to diesel. There is no significant difference among the density of the blends with up to 30% of methyl esters. The maximum difference is about (0.24–2.27 %) and (0.48–2.86 %) for the density of biodiesel diesel blends, respectively. This result matches with (Alptekin and Canakci, 2009), which reported that the ratio blending up to 20% of biodiesel diesel blending.

However, the blended fuel density is out of the blended fuel standard requirements ASTM D7467 (820–860 kg/m³) and BIS (820–860 kg/m³) at biodiesel blending ratio more than 40% (BD40). Thus, density for all biodiesel diesel blends must be closer within the recommended specified limits for biodiesel diesel blending standards.

4.3.1.4 Effect of Blending on Kinematic Viscosity

A viscous fuel has poor atomization characteristics and narrow spray angle, one with poor viscosity, leads to excessive wear and poor lubrication (Sarin, 2012). Thus, it is desired for a fuel to possess optimum value of kinematic viscosity. **Figure 4.6** shows viscosity of the fuel samples measured using Red wood viscometer in Chemical Laboratory. The diesel fuel is less viscous than biodiesel, it clearly shows that the viscosity of CBD100 and PBD100 are 6.78 and 2.69 times higher than the mineral diesel, respectively. This because the free fatty acid (FFA) concentration in biodiesel. As a result, biodiesel diesel blends of all fuels have higher kinematic viscosities as the percentage of biodiesel was increase (**Table 4.4** and **Figure 4.6**). This result was in agreement with Benjemea et al. (Benjumea et al., 2008; Tat and Van Gerpen, 1999). The viscosity of mineral diesel is recorded as the lowest number at 3.23 mm²/s. The behavior at each blend level among the CBD10, CBD20, CBD30, CBD40, CBD50, CBD60, CBD70, CBD80, CBD90, and CBD100 blends did not vary significantly with regard to viscosity. The viscosity of CBD10 is recorded as the lowest number at 4.01 mm²/s and 5.18 mm²/s which greater than the mineral diesel viscosity but still meet the ASTM D7467 requirement, while, CBD20 is recorded as 5.18 mm²/s and close to BIS standard requirement. However, the behavior of each blend level of palm biodiesel-diesel blends (PBD10– PBD100) vary significantly with regard to viscosity. The viscosity of PBD10, PBD20 are recorded as the lowest number at 3.51 mm²/s and 3.98 mm²/s which greater than the mineral diesel viscosity but still meet the ASTM D7467 requirement, and PBD30 is recorded as 4.58 mm²/s that meets the BIS standard requirement. Thus PBD10, PBD20, PBD30 blends exhibited kinematic viscosities that were satisfactory according to diesel biodiesel blend (ASTMD7467) and BIS standards. Therefore, it can be noticed that CBD10, PBD10, PBD20 and PBD30 can be used in the diesel engine without any modifications.

4.3.1.5 *Effect of Blending on Calorific Value*

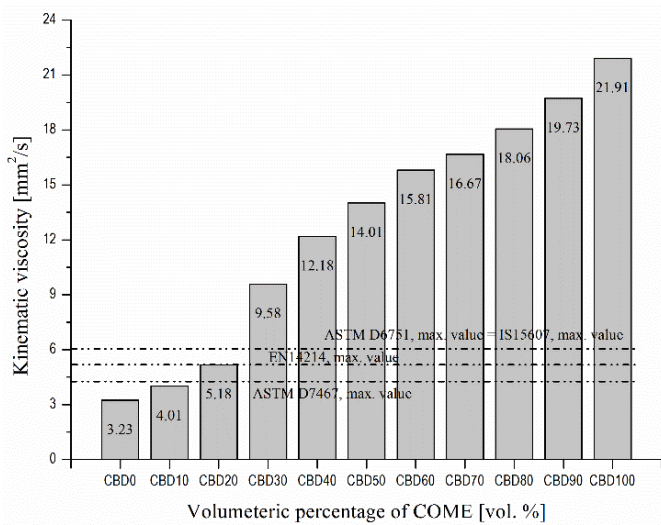
Calorific value is the amount of heating energy released by the combustion of a unit value of fuels. One of the most important determinants of the heating value is the moisture content of the feedstock oil (Cecrle et al., 2012). The heating value is not specified in the biodiesel standards ASTM D6751 but is prescribed in EN 14213 with a minimum value of 35 MJ/kg (Rashid et al., 2009). **Figure 4.7** presents the calorific value of the fuels measured by Bomb Calorimeter. It decreases as the amount of biodiesel increases in the blends. This is mainly due to the higher oxygen content of biodiesel than that of diesel. Thus hydrogen and carbon contents in the blend decrease which is the source of thermal energy (Rao, 2011). It is shown in **Table 4.4** that, the calorific value of ranges from 38.412 MJ/kg (CBD100) to 44.874MJ/kg (CBD10) for COME biodiesel diesel blend, and from 39.794 MJ/kg (PBD100) to 44.982 MJ/kg (PBD10) for POME biodiesel diesel blends which are slightly lower than diesel (44.69 MJ/kg). The calorific value does not indicate in the ASTM D7467 standard; all fuel samples meet the EN14213 requirement for the calorific value (minimum 35 MJ/kg). Hence, lower blends of biodiesels (CBD10–CBD40), and (PBD10–PBD40) have higher calorific value than other blending. The heating value for COME biodiesel/diesel blends is approximately by (1.13–14.05%), and POME biodiesel/diesel blends by (1.03–10.95%) lower than the heating value of mineral diesel (44.69 MJ/kg), respectively.

4.3.1.6 *Effect of Blending on Cetane Number*

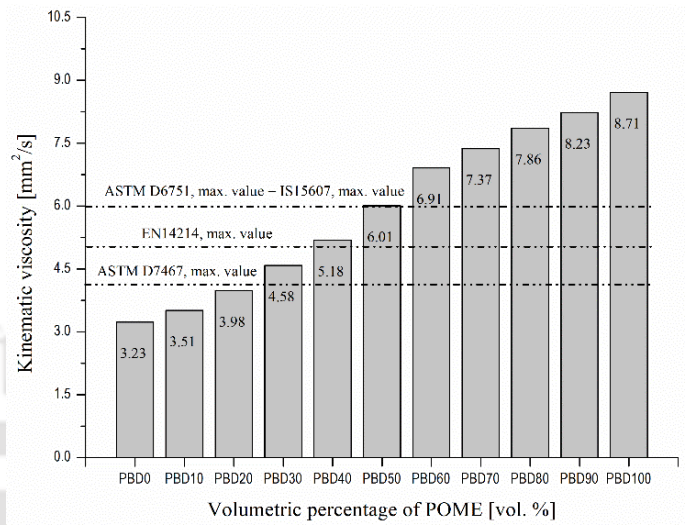
The Cetane number of the biodiesel is significantly high when compared to mineral diesel. **Figure 4.8** illustrates different Cetane number of the tested blends of fuels. It can be seen from the figure that the diesel fuel has the lowest Cetane number at 51°C while the blends of biodiesel CBD100 and PBD100 have the highest at 52.83°C and 60.84°C , respectively. The Cetane number is found to be increased when the percentage of biodiesel blend increase (**Figure 4.8**). This is because the Cetane number of biodiesel depends on the distribution of fatty acids in the original oil or fat. The longer the fatty acid carbon dioxide (CO₂) chains and the more saturated the molecules, the higher the Cetane number. The Cetane number all fuel samples meet the in the ASTM D7467 standard (min. 40) and the BIS requirement (minimum 51).

4.3.1.7 Effect of Chemical Bonds of Fuel on its Properties

The percentage of unsaturation in biodiesel is in reference to the fuel properties, engine combustion. The degree of unsaturation of an alkyl ester molecule is an indicator of the average number of double bonds present in its fatty acid chain, with a higher number of double bonds representing a higher degree of unsaturation. There exists a strong correlation between the degree of unsaturation and the properties of alkyl esters. The unsaturation in biodiesel affect the fuel properties. Increasing percentage of degree of unsaturation decreases the kinematic viscosity, improves the cold flow properties and increases moderately the heating value, increases density but also lowers the cetane number. The density of the test fuels increased with the degree of unsaturation. The density of biodiesel fuels increases with increase in percentage of unsaturation. An increase in density could be expected for every single percentage increase in unsaturation. The density increases with the increase in the number of double bonds, which means that themore unsaturated the originating oil, the higher the density of the derived methyl ester, and the greater the fuel mass that will be injected if a diesel-tuned engine is run on biodiesel. The density of biodiesel purely depends on the free space between the molecules, which in turn depends on the number of double bonds in the structure. Whereas, kinematic viscosity of biodiesel fuels decreases with increase in degree of unsaturation, due to the existence of double bonds reduces viscosity. It purely depends on the raw material and distribution of fatty acid ester composition. The cetane number of biodiesel fuels increases with chain length and decreases with increase in degree of unsaturation. Thus, each percentage increase in unsaturation could reduce in cetane number, which represents the ignitability of the fuel. Therefore, an increasing the percentage of unsaturation of biodiesel did not significantly affect oxygen content, heating value, and viscosity of biodiesel, which means it has insignificant effect of fuel atomization, but it may have a noticeable influence on combustion characteristics, via its effect on the cetane number. Since a higher degree of unsaturation of biodiesel fuels led to a longer ignition delay and, consequently, a more retarded start of combustion. This may also have attributed to the density variation of biodiesel due to the presence of double bonds. Ignition delay increases with increase in percentage of unsaturation and density. An increase in ignition delay may be predicted for an increase in each percentage of unsaturation.

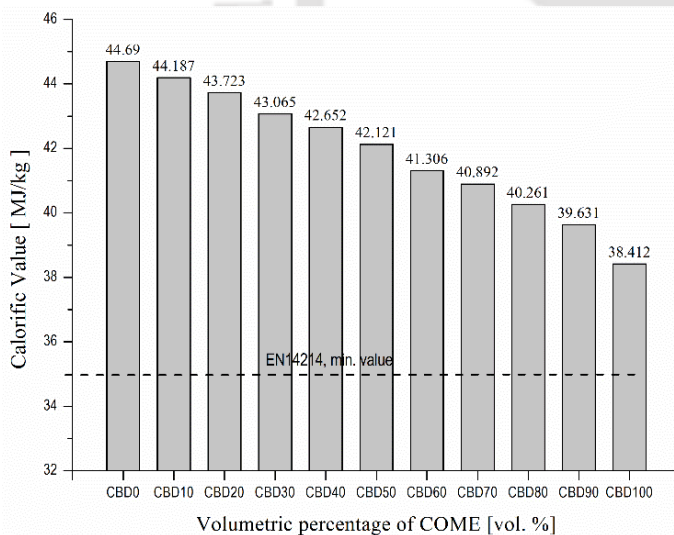


(a)

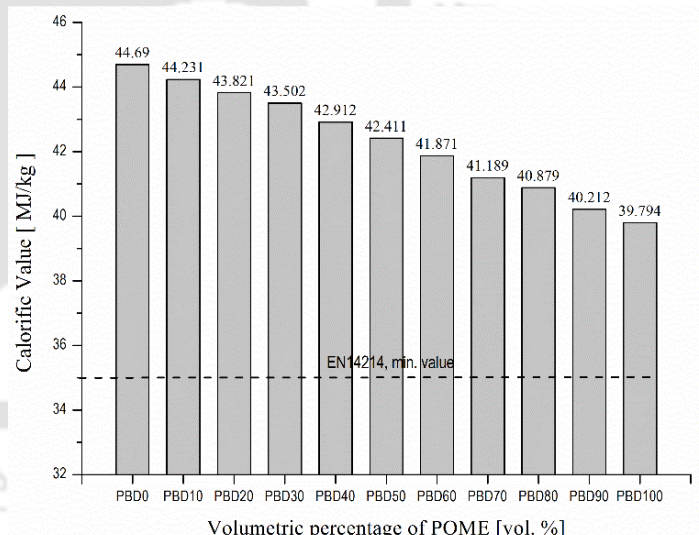


(b)

Figure 4.6: Blending effect on kinematic viscosity: (a) COME-diesel blends (b) POME-diesel blends.

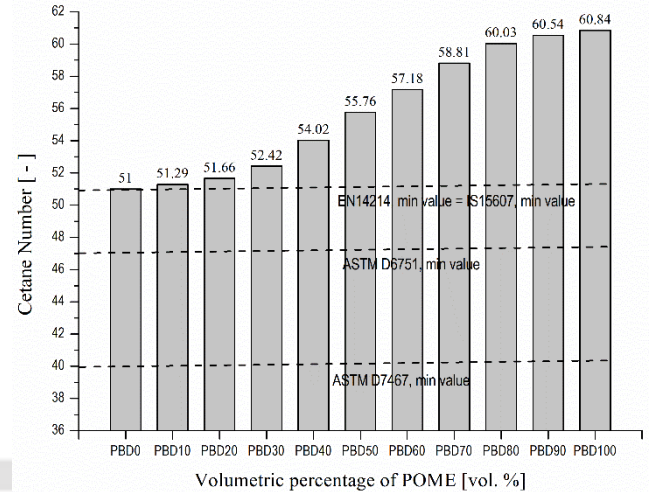
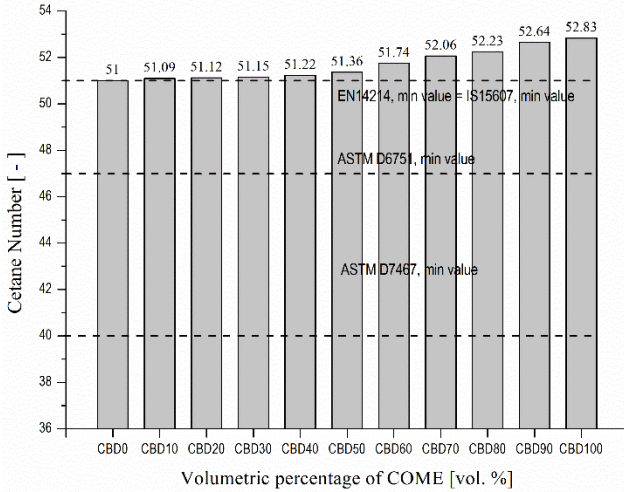


(a)



(b)

Figure 4.7: Blending effect on calorific value: (a) COME-diesel blends (b) POME-diesel blends.



(a)

(b)

Figure 4.8: Blending effect on cetane number: (a) COME-diesel blends (b) POME-diesel blends.

4.3.2 Engine Performance Analysis

In this study, engine performance was evaluated in terms of BP, BSFC and BTHE. The following section will discuss the results of these parameters. Blended fuel properties measurement and analysis performed in this study reveal that blended palm/castor biodiesel-diesel fuel meets the blended fuel standard ASTM D7467 specification up to 20% biodiesel ratio. However, these fuels are approved for use in unmodified diesel engines, after thorough investigation of their efficiency of in diesel engine characteristics. In this regards, tests were conducted using blended fuels 10%, 20%, 30% and 40% that may/may not meet the blended fuel standard requirements, in addition to mineral diesel fuel as baseline for comparison. **Table 4.5** presents engine performance and emission parameter results of (CBD0, CBD10, CBD20, CBD30 and CBD40), and (PBD0, PBD10, PBD20, PBD30 and PBD40) of blends of fuel.

Table 4.4: Testing properties of blended fuel: (a) COME biodiesel/diesel blends; (b) POME biodiesel/diesel blends.

Fuel property	Blend fuel code											Blended biodiesel standards(Handling and Guide, 2009; Silitonga et al., 2013)	
	CBD0	CBD10	CBD20	CBD30	CBD40	CBD50	CBD60	CBD70	CBD80	CBD90	CBD100	ASTM D7467	BIS
<i>(a) COME biodiesel/diesel blends</i>													
Kinematic viscosity at 40°C	3.23	4.01	5.18	9.58	12.18	14.01	15.81	16.67	18.06	19.73	21.91	1.9–4.1	2.0–4.62
Density at 27°C	838	842	851	857	864	876	889	906	914	931	943	820-860	820–860
Cloud point (CP), °C	-6	-4	-2	0	3	5	6	8	11	13	14	-	-
Pour point (PP), °C	-8	-5	-4	-2	1	2	4	6	8	10	12	-	3 max. winter
Flash point, °C	62	64	65	68	70	77	88	92	105	124	155	min. 52	min. 35
Fire point, °C	66	67	69	73	76	83	90	98	110	131	168	-	-
Calorific value, MJ/kg	44.69	44.874	44.817	44.236	43.681	42.393	41.378	40.835	40.366	39.878	38.412	-	-
Cetane number	51	51.09	51.12	51.15	51.22	51.36	51.74	52.06	52.23	52.64	52.83	min. 40	-
<i>(b) POME biodiesel/diesel blends</i>													
	PBD0	PBD10	PBD20	PBD30	PBD40	PBD50	PBD60	PBD70	PBD80	PBD90	PBD100	ASTM D7467	BIS
Kinematic viscosity at 40°C	3.23	3.51	3.98	4.58	5.18	6.01	7.21	7.67	8.06	8.53	8.71	1.9–4.1	2.0–4.1
Density at 27°C	838	841	843	862	872	889	901	912	928	939	946	820-860	820–860
Cloud point (CP), °C	-6	-5	-3	0	1	2	3	5	7	9	11	-	-
Pour point (PP), °C	-8	-6	-5	-3	-1	1	2	3	5	7	8	-	max. 3
Flash point, °C	62	75	89	135	172	194	212	228	245	260	265	min. 52	min. 35
Fire point, °C	71	78	93	146	185	199	227	232	257	268	272	-	-
Calorific value, MJ/kg	44.69	44.982	44.911	44.493	43.715	42.626	41.723	41.225	40.889	40.278	39.794	-	-
Cetane number	51	51.29	51.66	52.42	54.02	55.76	57.18	58.81	60.03	60.54	60.84	min. 40	-

Table 4.5: Comparison of engine performance and emission parameters of biodiesel/diesel blending testing.

Fuel property	COME-Diesel blend fuels					POME-Diesel blend fuels				
	CBD0	CBD10	CBD20	CBD30	CBD40	PBD0	PBD10	PBD20	PBD30	PBD40
Brake power, kW	3.5823	3.37019	3.31657	3.22981	3.10893	3.5823	3.4291	3.3867	3.3278	3.2956
BSFC, kg/kW.hr	0.26857	0.29658	0.37276	0.41889	0.45752	0.26857	0.28134	0.35356	0.38876	0.42662
BTHE, %	29.70685	26.54991	20.1432	18.40204	16.63701	29.70685	28.44672	22.67183	19.32171	18.03504
CO Emission, %	0.396875	0.36594	0.32531	0.28158	0.16865	0.39687	0.33924	0.27531	0.22158	0.12865
HC Emission, ppm	98	94	82	67	52	98	83	69	57	45
NOx Emission, ppm	398	404	414	426	443	398	443	462	489	521

4.3.2.1 Effect of Blending on Brake Power (bp)

Figure 4.9 shows the engine brake power for different blends of fuel at full engine load. It can be seen that brake power decreased marginally with increasing percentage fraction of biodiesel in blends. **Figure 4.9(a)** illustrates the brake power of COME biodiesel diesel blends of fuel (CBD0, CBD10, CBD20, CBD30 and CBD40) at full engine load. As a comparison, the brake power achieved for diesel fuel (CBD0) was about 5.92, 7.42, 9.84 and 13.21 % higher than that of the blended fuel CBD10, CBD20, CBD30 and CBD40 respectively, at the same engine conditions. Similarly, for POME biodiesel diesel blends of fuel of PBD10, PBD20, PBD30 and PBD40, the brake power was found lower by 4.27, 5.46, 7.10 and 8.0%, respectively as compared to diesel (PBD0) as shown in **Figure 4.9(b)**. The reason for the lower brake power of biodiesels compared to diesel can be attributed to their lower calorific values and higher viscosities. Both the calorific value and viscosity have an effect on the combustion. Additionally, uneven combustion characteristics of biodiesel fuel decreased the engine brake power (Muralidharan et al., 2011). Furthermore, it is seen that the average brake power reduction for CBD10–CBD30 in comparison to CBD0 fuel is 5–9%, whereas for PBD10–PBD30, it is 4–8%, respectively.

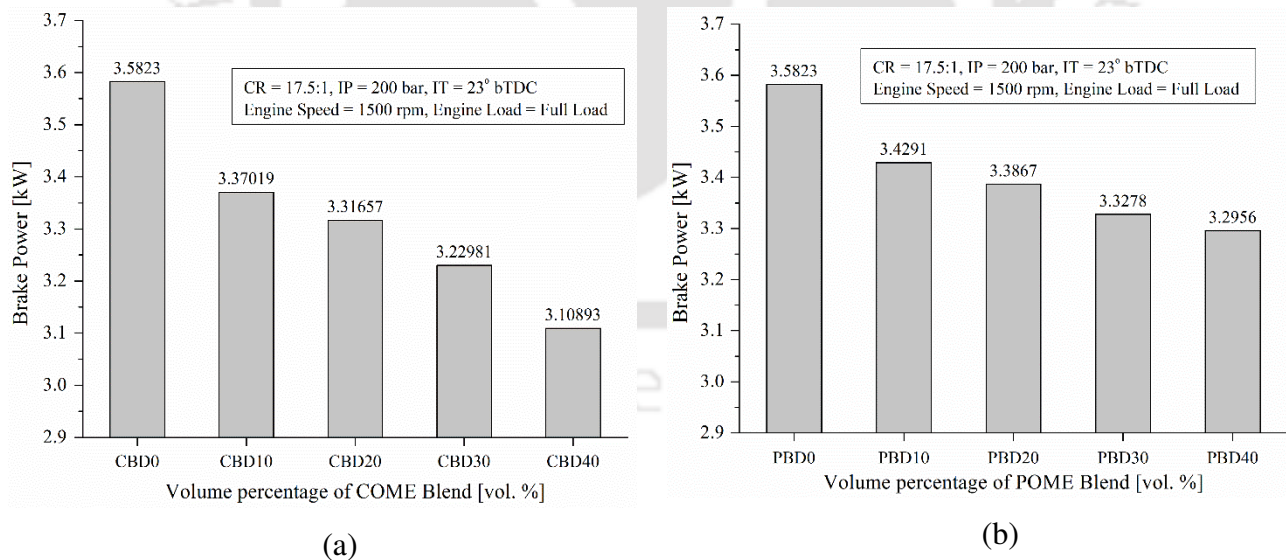


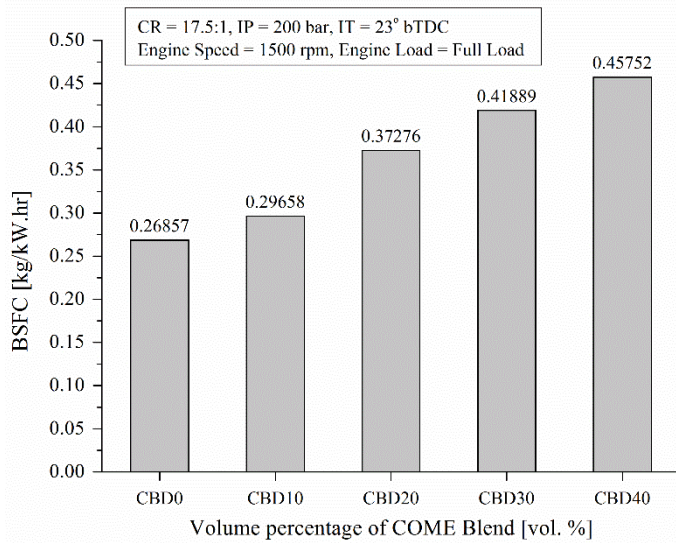
Figure 4.9: Effect of blending on brake power (BP): (a) COME-diesel blends, (b) POME-diesel blends.

4.3.2.2 *Effect of Blending on Brake Specific Fuel Consumption (BSFC)*

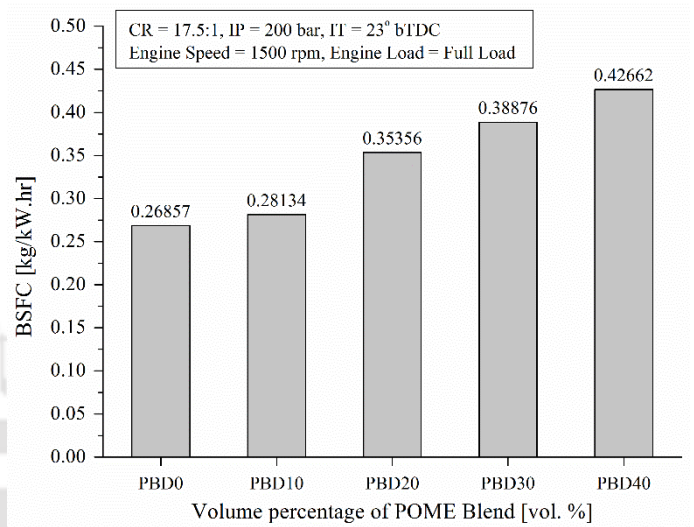
The BSFC is defined as the fuel flow rate per unit power output. It can be considered as a measure of the engine efficiency in using the supplied fuel to produce work. The BSFC is one of the most important parameters to evaluate engine performance with various fuels and it can be calculated based on the engine brake power and fuel mass flow rate for each speed. The BSFC of diesel engine depends on the relationship among volumetric fuel injection system, fuel density, viscosity and lower heating value (Qi et al., 2010). **Figure 4.10(a–b)** shows the variation of BSFC for all samples at full load engine running conditions. It is seen that the BSFC for biodiesel-blended fuels is higher compared to diesel. The minimum BSFC kept for diesel, which increased with increasing biodiesel ratios in the blended fuel. The results for BSFC shown in **Figure 4.10(a)**, proved that there is a significant difference among the fuels. As a comparison, the BSFC for diesel fuel (CBD0) was about 10.43 %, 38.79 %, 55.97 % and 70.35 % lower than that of the blended fuel CBD10, CBD20, CBD30 and CBD40, respectively, whereas, the BSFC for diesel fuel (PBD0) was about 4.75 %, 31.64 %, 44.75 % and 58.85 % lower than that of the blended fuel PBD10, PBD20, PBD30 and PBD40, respectively (**Figure 4.10-b**). This difference in BSFC with diesel and blended fuel is due to the high density value of biodiesel blended fuel compared to mineral diesel. The average BSFC for CBD0, CBD10, and CBD20 is 0.26857 kg/kW.hr, 0.29658 kg/kW.hr, and 0.37276 kg/kW.hr, respectively. while, for PBD0, PBD10 and PBD20 of was 0.26857 kg/kW.hr, 0.28134 kg/kW.hr, and 0.35356 kg/kW.hr, respectively. The reason for the higher BSFC of biodiesels can be attributed to the combined effects of the relative fuel density, viscosity and heating value of the blends (Chauhan et al., 2012). Biodiesel fuel is delivered into the engine on a volumetric basis per stroke; thus, larger quantities of biodiesel are fed into the engine. Therefore, to produce the same power, more biodiesel fuel is needed because biodiesel has a lower calorific value compared to diesel fuel (Lin et al., 2009; Tsolakis et al., 2007). Among the blend CBD10 and PBD10 are lowest at full engine load which is normally the optimal for any diesel engine. Hence, for BSFC point of view CBD10 and PBD10 may be advantages. This is because of the lower fuel flow rate due to high density of the blends. Higher proportion of castor and palm oil in the blends increase the viscosity which in turn increased the BSFC due to poor atomization of fuel.

4.3.2.3 *Effect of Blending on Brake Thermal Efficiency (BTHE)*

The BTHE is the ratio of the thermal power available in the fuel to the power that the engine delivers to the crankshaft. It is an indicator of the operation with the test fuel. This parameter is a better evaluation than fuel consumption for the performance of different fuels, besides heating value. Since thermal efficiency is normalized to the fuel heating value, it depends heavily on the manner in which the energy is converted. The BTHE is slightly higher with increasing biodiesel ratios in the blended fuel as shown in **Figure 4.11(a–b)**. Accordingly, the average reductions of BTHE of the blended fuel CBD10, CBD20 and CBD30 and CBD40 are 11 %, 32%, 38 %, and 44 % respectively, which is lower than that of the mineral diesel (CBD0) under the same engine conditions. Similarly, for PBD10, PBD20, PBD30 and PBD40 were decreased by 4.24%, 23.96%, 34.96% and 39.29% with respect to diesel (PBD0). This difference may be attributed to the high kinematic viscosity of the blended fuel compared to mineral diesel, which affects the fuel vaporization and combustion process. Thus, the efficiency of CBD10 and PBD10 are close to diesel. This is due to the more amount of diesel presence in blended fuel. The amount of diesel is decreased in the CBD20, PBD20, CBD30, PBD30, CBD40 and PBD40 blends, respectively, which is recorded loss in efficiency. In maximum load, for CBD10 and PBD10 blends the efficiency are 26.6% and 28 % and for diesel is 29.7%.

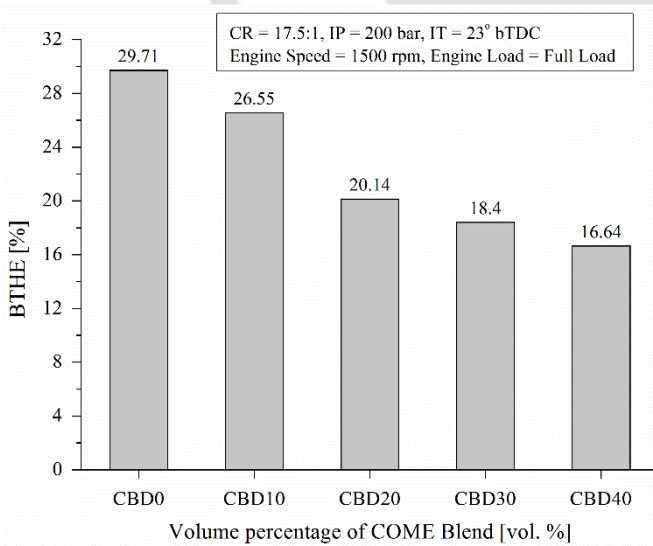


(a)

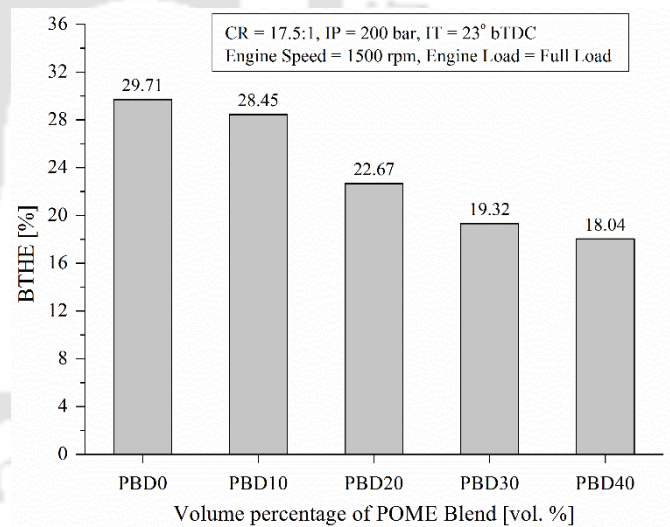


(b)

Figure 4.10: Effect of blending on BSFC: (a) COME-diesel blends, (b) POME-diesel blends.



(a)



(b)

Figure 4.11: Effect of blending on BTHE: (a) COME-diesel blends, (b) POME-diesel blends.

4.3.3 Effect of COME and POME Biodiesels on Engine Knock

Engine knocking is the metallic ping sound generated from the engine. In compression ignition engines, the fuel is sprayed into cylinder over a period of time. However, if the ignition delay is longer, more fuel will be accumulated into the cylinder. When the fuel ignites, the instantaneous pressure buildup can lead to jamming of piston and violent gas vibrations, thus causing premature wear of them. This undesirable type of combustion in the engine causes knocking. Many biodiesels have been tried as fuel for the compression ignition engine. Major drawback of using the biodiesel in compression ignition engine was its higher viscosity, higher density, lower cetane number, lower calorific value and poor volatility. Viscosity of the fuel leads to poor atomization causes the incomplete combustions. High viscosity of the fuel leads to cloaking of the filters and fuel injectors and also increases the pumping power. Higher viscosity of fuel leads to longer ignition delay, rise in peak pressure very close to TDC, and creates knocking. Most of the alternative oil like palm oil biodiesel, castor oil biodiesel has relatively higher cetane number than mineral diesel fuel which define the ignition quality of the fuel. The fuel, which has higher cetane number, supports for good combustion than the lower cetane number fuel. The ignition delay is the time interval from the start of nozzle valve lift to the start of rapid pressure rise. It is observed that the ignition delays of palm and castor biodiesels are lower than that of diesel and decrease with an increase in the biodiesel in the blend. Increase in CN causes a shorter ID which results in less fuel being injected during the premix burn and more during the diffusion burn portion, thus reducing the cylinder pressure rise, which may result in lower cylinder temperatures and reduce engine knock. These characteristics lead to a complete combustion of biodiesel fuel with lower exhaust emissions compared to mineral diesel. Hence, the peak pressure of biodiesel or its blends takes place definitely after TDC for safe and efficient operation. Otherwise, a peak pressure occurring very close to TDC or before that causes severe engine knock, and thus affects engine durability.

4.3.4 Analysis of Engine Emission

The emissions analysis has been conducted only at full engine load (1500 rpm). The variation of NO_x, HC and CO emissions for COME biodiesel diesel blend (CBD0, CBD10, CBD20, CBD30 and CBD40), and for POME biodiesel diesel blend (PBD0, PBD10, PBD20, PBD30 and PBD40) are shown in **Figure 4.12** to **Figure 4.14**.

4.3.4.1 Effect of Blending on Carbon Monoxide (CO) emission

Carbon monoxide is formed during the combustion process with rich air-fuel mixtures regions and when there is insufficient oxygen to fully burn all the carbon in the fuel to CO_2 (Kalam et al., 2005). **Figure 4.12(a–b)** shows the results of CO emission at full engine load. The average reduction in CO emission for COME blend of fuels CBD10, CBD20, CBD30 and CBD40 compared to CBD0 are 7.8%, 18.0%, 29.1% and 57.5%, whereas for POME blend of fuels PBD10, PBD20, PBD30 and PBD40 were 14.5%, 30.6%, 44.2% and 67.6% respectively. It can be seen that CO emission decreased with increasing biodiesel percentages in the blends. This is because biodiesel contains higher oxygen than diesel fuel which helps to complete the combustion (Chauhan et al., 2012). Thus, increased biodiesel percentage fraction caused to decreased the CO emission in a diesel engine.

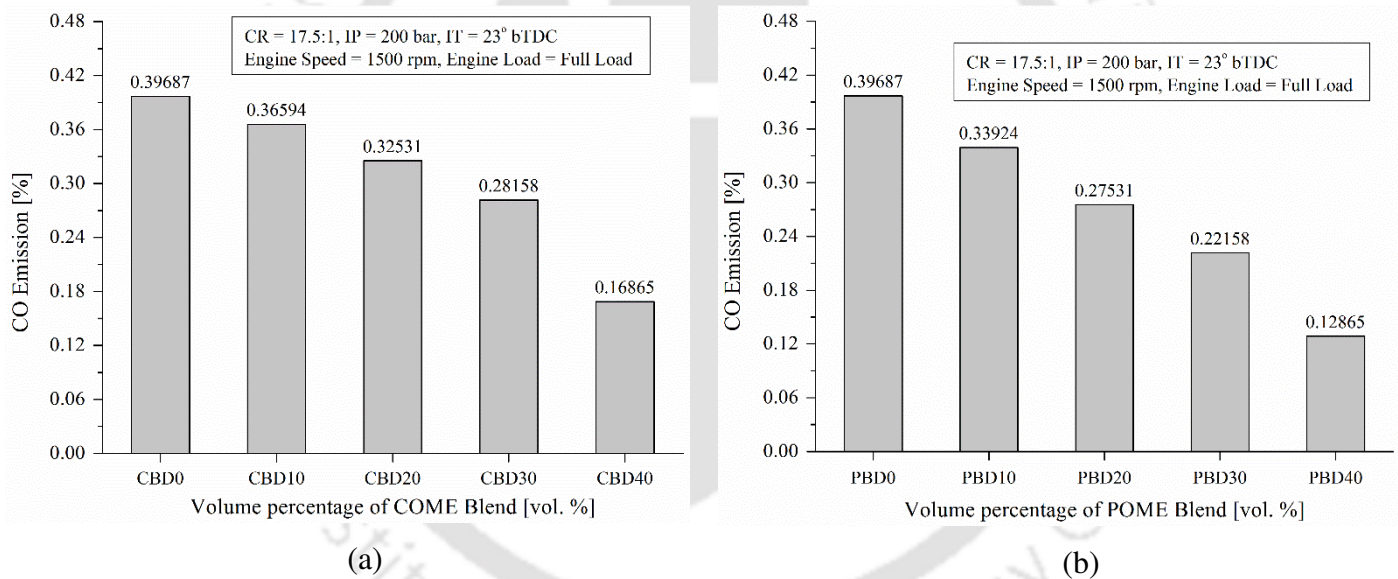


Figure 4.12: Blending effect on CO emission : (a) COME-diesel blends, (b) POME-diesel blends.

4.3.4.2 Effect of Blending on Unburnt Hydrocarbon (HC) Emission

Unburned HC results from the incomplete combustion of fuel and flame quenching (Kalam et al., 2005). The variation in HC emissions with respect to biodiesel blends at full engine load is shown in **Figure 4.13(a–b)**. The HC emission of diesel fuel was higher than the blends of fuel. HC content decreases gradually with the blending of oils. This is because of the higher oxygen content and

higher cetane number of biodiesel, which helps to complete the combustion and thus reduce HC emission (Lapuerta et al., 2008). The average percentage reductions of HC in CBD10, CBD20, CBD30 and CBD40 were 4.08%, 16.32%, 31.63% and 46.94%, respectively as compared to BD0, whereas, PBD10, PBD20, PBD30 and PBD40 by 15.31%, 29.59%, 41.84% and 54.08%, respectively as compared to diesel (PBD0). Increasing biodiesel ratios in blended fuel lowered hydrocarbon emissions. All blends have shown lower HC emissions after about full load. This may be due to higher oxygen content as well as the higher cetane number.

4.3.4.3 Effect of Blending on Oxide of Nitrogen (NO_x) Emission

Figure 4.14(a–b) shows the variation in NO_x emissions with biodiesel diesel blends of at full engine load operating conditions. It has been observed that, NO_x emissions for all blends are higher as compared to diesel. This is probably due to higher combustion temperature in the engine cylinder with increasing blends. It is also observed that with increase in the percentage biodiesel blends there is a trend of increasing NO_x emissions. Overall increased percentage of oxides of nitrogen gases in CBD10, CBD20, CBD30 and CB40 of COME blend are by 1.5%, 4.0%, 7.0% and 11.31%. However, for POME biodiesel blends of PBD10, PBD20, PBD30 and PBD40 was by 11.3%, 16.1%, 22.8% and 30.9%, respectively as compared to diesel (PBD0). This can be attributed to the lean air/fuel ratio, as biodiesel contains 10–12% more oxygen in its molecular structure. Many researchers reported that oxygenate fuel blends can cause an increase in NO_x emission. Normally, complete combustion causes higher combustion temperature, which results in higher NO_x formation (Devan and Mahalakshmi, 2009). In addition, the higher viscosity of the blend of fuel leads to a bigger droplet size and shorter ignition delay. Hence NO_x emission is increased (Kalam et al., 2011). On the other hand, NO_x emission is strongly related to the higher peak combustion temperature. The test results shows that, the NO_x emission for CBD10–PBD30 blends of COME was close to diesel fuel (CBD0), while the blends of POME biodiesel only PBD10 and PBD20 has low NO_x emission and close to diesel fuel.

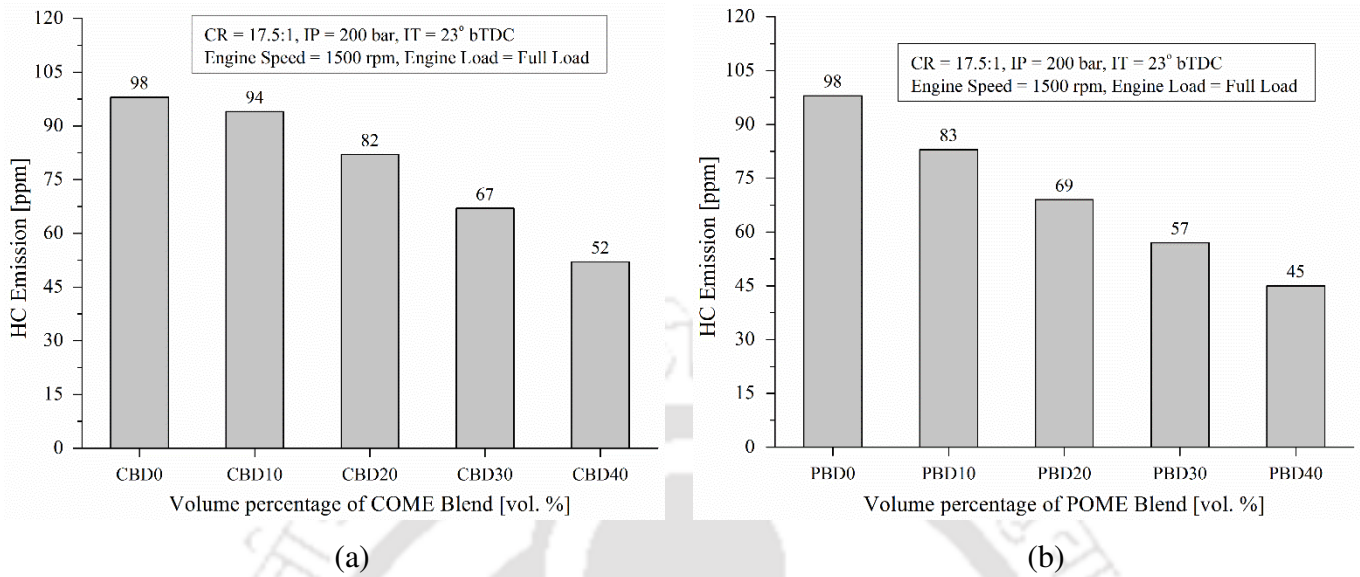


Figure 4.13: Blending effect on HC emission: (a) COME-diesel blends, (b) POME-diesel blends.

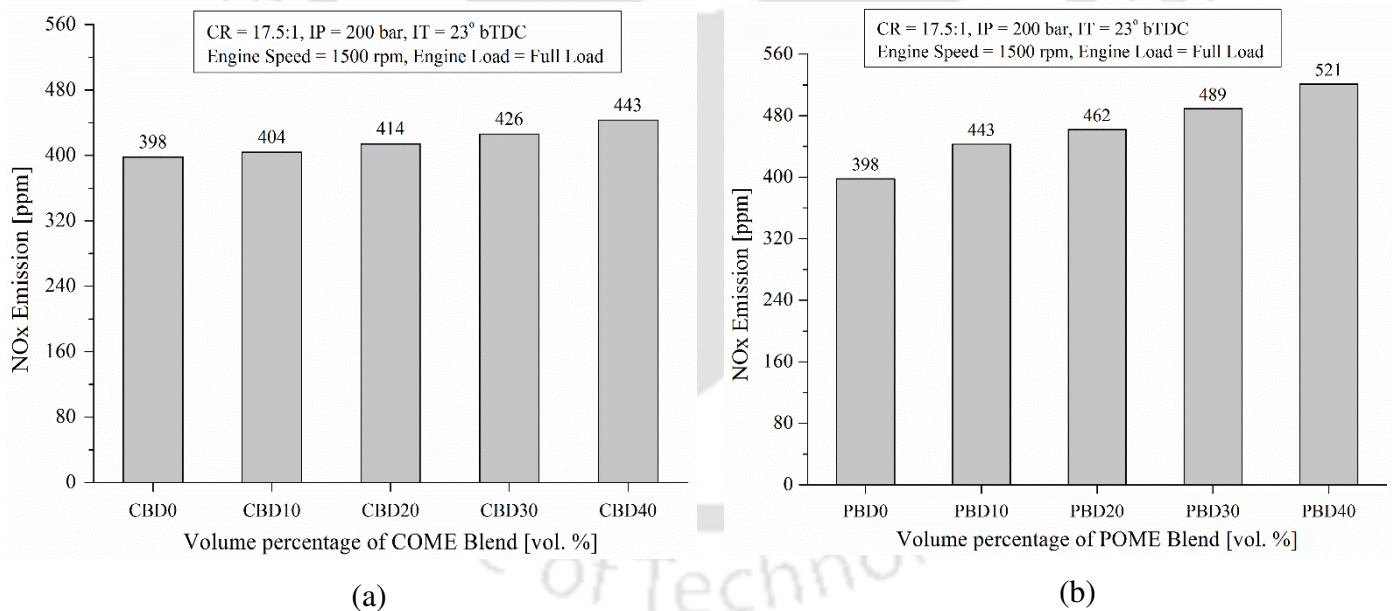


Figure 4.14: Blending effect on NOx emission: (a) COME-diesel blends, (b) POME-diesel blends.

4.4 Uncertainty Analysis

Any experiment is not free from error. Error may occur due to the instrument's inaccuracy, inappropriate calibration, human inadequacy etc. Therefore, uncertainty analysis is mandatory for any experimental work. The sequential perturbation technique (Kline and McClintock, 1953;

Moffat, 1982) is followed to acquire the uncertainties of independent and performance parameters (Appendix - D). The uncertainty analysis was carried out at standard engine operating conditions (CR 17.5, fuel IP 200 bar, and IT 23° bTDC at 1500 rpm). Some of these are air flow rate (1.1 %), liquid fuel flow rate (0.1%), engine load (0.1%), engine speed (1.3%), LHV of liquid fuel (1.0%) etc. Because of these facts, the calculated accuracy of the performance and combustion study for both COME blend (CBD10), POME blend and diesel run engine are found within $\pm 3.6\%$. However, the accuracy of the emissions is found within $\pm 2.3\%$.

4.5 Summary

All of the measured fuel properties of the biodiesel (POME & COME) met the international standards of ASTM D 6751, EN14214 and IS15607 with the exception of density and kinematic viscosity. Thus, neat biodiesel may require making minor modifications to the engine to avoid maintenance and performance problems. Therefore, the high viscosity and density of the methyl ester has to be further reduced by blending with diesel fuel up to BD100 to see the effect of blending on its properties to satisfy within the limits for biodiesel standard specifications. When biodiesel is blended with diesel the characteristics of the new fuel changes as per the volumetric percentage of biodiesel in a mixture of biodiesel and diesel. The blending ratio will be optimum when the blended fuel will satisfy the different blended biodiesel standards as specified by different countries (ASTM D7467 and BIS).

- The first objective of the study was to characterized how the key fuel properties change when conventional diesel fuels blend with castor oil biodiesel and palm oil biodiesel. The variations of the fuel properties with the volumetric percentage for the POME biodiesel-diesel blends and COME biodiesel-diesel blends were determined. According to the results, the density, viscosity, flash point and pour point of the blended fuel increase with the rise of biodiesel concentrations in the fuel blend. The POME-diesel blends and COME-diesel and POME-diesel were able to satisfy all of blended biodiesel specifications except for their inferior kinematic viscosity and density. The pour point of the blended fuel increases as the concentration of biodiesel increases in the blend with a lower difference in the cloud point of the blends. A significant linear reduction in blended fuel calorific values is seen with increasing biodiesel ratio in the blend, with a lower heating value for neat castor oil biodiesel (CBD100) and palm oil biodiesel (PBD100), which is about 14.05% and 10.95% lower than mineral diesel (BD0),

respectively. The fuel property results showed that the blended fuel CBD20 and PBD20 has a pour point of approximately less than 0°C. Furthermore, the densities are 843 kg/m³ and 851 kg/m³, which are both meet blended biodiesel standard specifications, whereas kinematic viscosity for CBD20 (5.18 mm²/s) is out of limits blended biodiesel standards, but CBD10 (4.01 mm²/s) and PBD20 (3.98 mm²/s), which meet blended fuel standard specifications. On the other hand, the reduction in the heating value CBD20 and PBD20 was about 2.16% and 1.94%, respectively compared to diesel fuel. The blending ratio of the COME is recommended up to 10% volume (CBD0) and POME up to 20% volume (PBD10–PBD20), which is remarkable beneficial and meet the blended fuel standard (ASTM D7467 and BIS). Other blends can't be considered as feasible alternative substitution diesel fuel for diesel engine without any modification.

- From the engine characterization analysis, it was found that the performance of the biodiesel blends PBD10 and PBD20 of POME biodiesel blends were comparable to that of diesel (PBD0) rather than PBD30 and PBD40 fuels. For COME biodiesel blends, only CBD10 fuels relatively close to diesel fuel. Engine tests demonstrated that increasing biodiesel ratios up to 20% in the blended fuel reduced the engine brake power by about 7.4% and 5.5%, respectively for POME and COME biodiesel blend, and increased the brake specific fuel consumption by 24.04% and 27.95%, respectively. However, there was significant difference for the engine brake thermal efficiency 31.05% and 47.52%, with the blended fuels compared to the mineral diesel. The use of PBD20 as a partial diesel fuel substitute produced lower HC (29.6%) for POME blend, and (16.3%) for COME blend (CBD20); and CO emissions by 30.6% for PBD20, and 18.0% for BD20 blend, respectively. But slightly higher NO_x by 16.1% for PBD20 blend, and 4.0% for BD20 blends of emission compared to diesel.

Accordingly, the blending ratio of the POME biodiesel/diesel blended fuel up to 20% (PB20) whereas COME biodiesel/diesel blended fuel up to 10% (CBD10) is recommended and could replace diesel in diesel engines for getting better performance as a comparable fuel to mineral diesel with smoother engine operation. It would also help in achieving energy economy without major modification. Therefore, it can be concluded that low blend ratios POME and COME biodiesel can be considered as a promising source of biodiesel production in India.

CHAPTER-5

Results and Discussion: Preheated Biodiesel Run Engine

Overview

Many investigators have studied the emissions in diesel engines fueled with neat biodiesel. It has been usually reported that there are reductions in CO, HC, while NO_x increases. The oxygen content in biodiesel is the main factor for reducing emissions and increasing NO_x as a result of better combustion. Conversely, it has been revealed that the use of biodiesel leads to a slight reduction in the brake power, torque, and BSFC is marginally higher compared to diesel. These changes can be attributed to the lower heating value, higher viscosity and density of biodiesel compared to diesel. The viscosity of fuels has important effects of fuel droplet formation, atomization, vaporization and mixing process, thus influencing the emissions and performance parameters of the engine. The intention of this study is to investigate the effects of the viscosity and density of POME) and COME, on the performance and exhaust emissions of a diesel engine. The properties of preheated POME and COME were evaluated based on biodiesel standards (ASTM D6751, EN 14214 and IS15607). A heat exchanger was fabricated based on the indigenous design for heating the fuel with waste heat of exhaust gases of the engine. An alternate fuel supply system was attached with the heat exchanger to preheat the biodiesel. The developed preheating arrangement was integrated to the engine for the completion of experimental set up. First, POME and COME were preheated at different temperatures, and its properties were determined. Then biodiesel optimal preheating temperature which provides better engine performance parameters with lower the exhaust emissions were determined. Later, performance and emissions parameters of a diesel engine fueled with a biodiesel, namely POME and COME, subjected to preheating at different temperatures have been investigated. The tests were carried out at full load conditions in a VCR DI CI engine. Before supplied to the engine, POME and COME were preheated to nine different temperatures, namely (54–138 °C) with 12 °C increments.

5.1 Preface

Biodiesel has higher kinematic viscosity as well as density and lower heating value compared to petroleum based fuels (Hoekman et al., 2012). The higher viscosity and density can affect fuel flow rate, cause poor fuel atomization and create improper homogeneity in charge during the combustion process (Nayak and Mishra, 2017). It results in incomplete combustion and inconsistent engine performance and emission parameters for diesel engines resulting failures (Gao et al., 2009; Jazair et al., 2011). Hence the direct use of biodiesel oil in a compression ignition engine is almost impossible due to poor fuel injection characteristics. Thus, the fuel properties of biodiesel have to be improved adequately before using directly in a diesel engine (Aksoy et al., 2009; Hoekman et al., 2012). Presently, blending is widely used as one of the methods for improving fuel properties (i.e. kinematic viscosity, density and heating value) of biodiesel (Atabani et al., 2012). However, our recent analyses showed that, low blending ratios (20% volume biodiesel) are considered as a feasible fuel. Many previous studies also indicated that, low blended biodiesels are used as a replacement of diesel fuel in a diesel engine instead of using a high-levels of blends or neat biodiesel (100% volume biodiesel), as it reduced the emission levels along with increase the engine performance (Naik and Balakrishna, 2017). However, diesel engine run with blended methyl ester (biodiesel) ratios greater than 20% volume, described extra drop in exhaust emissions (CO and HC). Nevertheless, they are worse in the performance in terms of BSFC and BTHE (Mohan et al., 2014). To overcome the existing problems of neat biodiesel (100% volume) in a diesel engine, it is essential to optimize the fuel properties by heating process. A few studies have been reported that, heating of biodiesel is the simple, economical and competent technique for optimizing the problems (Khalid et al., 2017). Fuel preheating is one of the operative techniques to decrease the density and viscosity of biodiesel and its blend, which will progressively decrease as the fuel preheating temperature increases (Sharma et al., 2014). The studies show that preheating biodiesel improves the fuel injection characteristics (atomization and vaporization rate) by dropping the viscosity and the density of biodiesel. The important engine performance and emissions parameters were also improved using heating (Gumus et al., 2012). Aksoy et al. (2009), investigated the effect of fuel preheating on fuel properties of biodiesels and their blends. Tests were conducted at the temperature ranges from 0 to 100 °C in steps of 10 °C. The results indicated that, the viscosity of biodiesel gets closure to diesel with increase of preheating temperatures. Abedin et al. (2014), studied the effect of preheating on palm oil (biodiesel) on fuel injection

characteristics (spray and atomization) subsequently on engine performance and emission analysis. They found that, the kinematic viscosity and fuel spray characteristics improved. Yilmaz and Morton (2011), evaluated diesel engine performance and emission parameters under the effects of heating vegetable oils. The tests are conducted at elevated temperature and the base findings were compared with standard diesel fuel. The result indicated that, emission concentration of CO, HC, and NO_x was improved with heating compared to unheated biodiesel and became competitive with diesel fuel. Nasim et al. (2013), executed experimental investigation on engine performance analysis of compression ignition engine fueled with preheated Jatropha oil. They found that, the effect of preheating Jatropha oil significantly decreased the viscosity of the oil. The results also exposed that, the brake specific fuel consumption decreased with increasing fuel-preheating temperature of the oil. Pradhan et al. (2014), utilized the waste exhaust gases for heated Jatropha oil and run in a diesel engine. Studies revealed that, preheating Jatropha curcas oil caused to decrease its density and kinematic viscosity. The ignition delay and the BSFC of diesel engine were found improved with preheating.

The main objective of this experimental study is to investigate the effects of preheated neat biodiesel on viscosity and density, and thus influences to the performance and emissions of a diesel engine. It is expected that this work will provide the knowledge of the effects fuel preheating on its properties. Heating is the easiest solution to bring lower the kinematic viscosity and density of biodiesel, thus enhancing engine characteristics of diesel engine.

5.2 Materials and Methodology

In this study a thorough review has been done on fuel preheating of techniques. The exhaust heat recovered from a diesel engine is used to preheat biodiesel in order to use it as an alternative for conventional diesel fuel. For this purpose, a helical coil tube heat exchanger was fabricated for heating the fuel with waste heat of engine exhaust gases. The heat exchanger has been shown in **Figure 3.10**. An alternate fuel supply system (like biodiesel storage tank, fuel lines and burette) were fixed with a heat exchanger to preheat the biodiesel oil. The developed preheating arrangement was attached to the engine for the completion of experimental set up. Then, the effect of fuel preheating using biodiesel as fuel in a diesel engine is investigated. Experiments are carried out at the rated speed of 1500 rev/min with a standard engine manufacture operating setting (CR = 17.5, fuel IP = 200 bar, and fuel IT = 23° bTDC) at full load operating conditions (refer the test

matrix-**Table 3.3**). The biodiesel is heated to 54 °C, 66 °C, 78 °C, 90 °C, 102 °C, 114 °C, 126 °C and 138 °C using engine waste heat exhaust gases. Experiments are conducted at all temperatures. Engine performance and emissions parameters are analyzed. It is recommending for the optimal fuel preheating temperature which improved the important fuel properties of biodiesel (kinematic viscosity and density) with better performance parameters with a lower exhaust emission of diesel engine.

5.3 Results and Discussion

5.3.1 Fuel Characterization

5.3.1.1 Test Fuel Properties

The test fuel properties mentioned in **Figure 5.1** are considered as the desired properties of a fuel for its application in diesel engine. The test fuel need to have better fuel characteristics such as higher calorific value, low moisture content, low ash content, moderate combustion temperature and moderate velocity of combustion. The quality of the fuel is directly interconnected with the performance and emission parameters of a diesel engine (Çelikten et al., 2012). Hence, there values need to be in the range of most widely known biodiesel standard like ASTM D6751, EN14214 and IS15607 to define the limits and to meet the standard diesel engine parameters. The elemental analysis of existing fuel properties of POME (PBD100) and COME (CBD100) and diesel fuel are listed in **Table 4.2**. The kinematic viscosity of PBD100 and CBD100 were approximately 6.78 and 2.69 times that of diesel, density was nearly 12.5% and 12.9% higher, the flash point and cetane number met the lower limits fuel standards, as reported in references (Knothe, 2006). The measured calorific values are close to diesel fuel. The density and viscosity value of PBD100 and CBD100 are shown higher and beyond international standard values. Thus, the properties of each biodiesel needs improvement to meet minimum standard limits which are widely known ASTM D6751, EN14214 and IS15607 for a diesel engine application to avoid fuel atomization and spray characteristics problems. In this regard, oil properties are aimed for improvement by using preheating technique. It is one of the suitable method to optimize the fuel properties and solve the poor fuel spray characteristics that results poor engine performance and higher emission parameters (Mustaffa et al., 2014b).

Table 5.1: The properties of test fuels.

Properties	Test fuels			Biodiesel at elevated inlet temperature (°C)					
	Diesel	CBD100	PBD100	102		114		126	
				PCBD100	PPBD100	PCBD100	PPBD100	PCBD100	PPBD100
Kinematic viscosity, (mm ² /s)	3.23 at 40°C	21.91 at 40°C	8.71 at 40°C	8.78	4.92	7.89	3.47	6.23	3.29
Density, (kg/m ³)	838 at 27°C	943 at 27°C	946 at 27°C	876	881	865	868	859	864
Flash point, °C	63	155	265	159	268	176	289	183	287
Pour point, °C	0	4	5	10	8	9	6	7	5
Calorific value, MJ/kg	44.69	38.41	39.79	39.02	39.99	39.62	40.63	39.81	41.20
Cetane number, (minimum value)	51	52.83	60.84	52.89	61.81	53.96	63.71	53.07	62.99

CBD100-neat COME biodiesel, PBD100-neat POME biodiesel, PCBD100-preheated neat COME biodiesel, PPBD100-preheated POME biodiesel.

5.3.1.2 Effect of Preheating on the Fuel Property of Biodiesel

The kinematic viscosity and density of COME and POME at different preheating temperature from 54 °C to 138 °C in increments of 12°C were measured. They have been shown in **Figure 5.1**. It is indicated that, the kinematic viscosity and density are reduced gradually as the fuel preheating temperature increases respectively from 54 °C to 138 °C. Similar works were also reported in literature (Khalid et al., 2017). It is seen in **Figure 5.1(a)**, at 114 °C fuel temperature, the kinematic viscosity of PBD100 and CBD100 were decreased rapidly to (3.47 and 7.89 mm²/s) from (8.71 and 21.91 mm²/s), respectively at 40°C. The density decreases from (946 and 943 kg/m³) at 27°C to (868 and 865 kg/m³) at 114 °C. These values meet within ASTM D6751, EN14214 and IS15607 standards. The average percentage reduction of kinematic viscosity and density for PBD100 and CBD100 were, 60.2 and 63.99%, and 8.2 and 8.3%, respectively as compared the intimal property value of unheated biodiesel conditions. At 114 °C temperature, the kinematic viscosity and density values for PBD100 (POME) was found close to diesel fuel, while for and CBD100 (COME) slightly higher needs further improvement. The preheated PBD100 and CBD100 beyond 114 °C, in fuel mixture started to form vapor and make vapor lock in the fuel line, this may be the oxygenated nature of biodiesel. It may also cause excessive fuel leakage owing to lower fuel kinematic viscosity. It is seen in **Figure 5.2(a)**, that, when the temperature is increased to 126 °C, the BSFC values for all biodiesels are gradually increases, this may be due to excessive leakages of lower kinematic viscosity of preheated biodiesel and therefore the temperature of heating

biodiesel up to 114 °C was practical. Hence, among the various preheating temperature heating the PBD100 and CBD100 oil studied, the optimal temperature 114 °C brings down the viscosity and density of neat PBD100 close to a diesel fuel. It could affect the diesel engine performance and emission parameters. Referring to **Table 5.1**, the measured fuel properties are validated with literature data (Debnath et al., 2013a). It is seen in, the flash point for each biodiesel increases with fuel preheating temperature of 114 °C with respect its reference value (from 155 to 176 °C for CBD100 and 265 to 289 °C for PBD100 biodiesel). Like other properties, flash point of biodiesels is higher compared to diesel. The higher value of flash point is an advantage for biodiesel because it becomes safe for handling, storages, and transportation. The calorific values and cetane number of each biodiesel increased with fuel preheating temperature (**Table 5.1**).

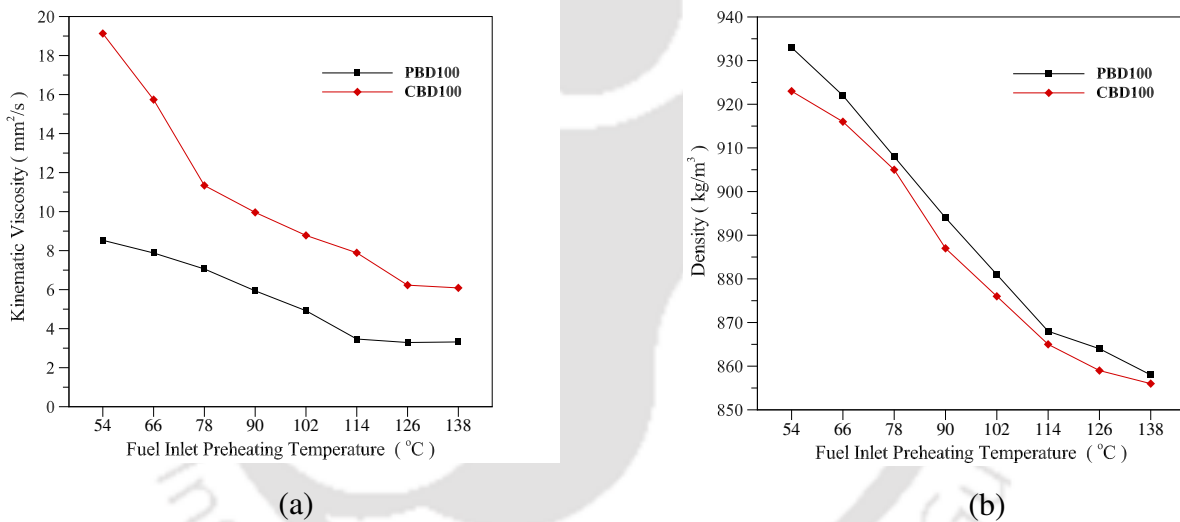


Figure 5.1: Effect of fuel preheating temperature on biodiesels: (a) kinematic viscosity; (b) density.

5.3.1.3 Effect of Preheating on the Injected Fuel Quantity and Energy Input

The main problems with the use of neat biodiesel in direct injection compression ignition engines are higher viscosity and density. The fuel properties have the strong impact on fuel injection, mainly characterized by atomization. Fuels with high viscosity tend to form larger droplets on injection which can cause poor combustion and emissions while the low viscosity fuels may not able to supply sufficient quantities to fill the pumping chamber, thereby, loss in engine power.

Increased density tends to degrade atomization rates, they have a significant effect on the atomization quality of the spray injectors, with subsequent impacts on the efficiency of the combustion and emissions. All diesel injection systems meter the fuel on a volume basis, so that fuel density affects the mass of fuel injected. These fuel properties affect the atomization of biodiesel on fuel injection characteristics which influencing engine operating parameters of a diesel engine. Hence, it is essential to improve the fuel properties for better combustion of the biodiesel oils. To get the similar physical properties to mineral diesel fuel from the diesel engine operation, single injection quantity of methyl ester biodiesel oil should be increased. For increase the biodiesel injection quantity, the followings effective ways are worth for consideration; preheating biodiesel, the increase of fuel injection pressure, the increase of nozzle hole and orifice diameter, and the nozzle tip converting. Among these ways, the preheating biodiesel at different fuel inlet temperature is easier than other method for bring down the viscosity and density which enhances better fuel injection, improves mixture formation, thereby minimize the troubles due to poor fuel droplet formation and atomization, allowing better injection of fuel and increase in the heating value as the proportion of diesel. The preheating reduces the fuel's viscosity which helps in better fuel flow and improved atomization. Also the preheating of biodiesel leads to improve vaporization characteristic and hence the improved combustion can be accomplished. Hence preheating is a viable solution for in biodiesel. Since, the spray penetration of injected biodiesel found to be enhanced, resulting in increased the injection quantity of fuel and spray area. Compared with the case of without preheating biodiesel, the injection quantity of preheated biodiesel (mg/cycle) will be higher with a fixed duration of time, since the amount of biodiesel supplied to the injection hole can be increases due to a lower viscosity of preheated biodiesel. This promotes mixture formation and distributes larger amount of fuel between sprays, thus creates good spray atomization and exhibits a greater amount of fuel-air premixing prepared for combustion. It seems that increment in fuel inlet temperature also affect the spray tip penetration of the fuel and can make spray tip penetration shorten. The injected heat value means the heating value by the single injection of fuel. It is calculated to multiply the single injection quantity with density and low heating value of preheated biodiesel at each elevated preheated inlet temperatures indicated in **Table 5.1**. It is expected that the heating value of preheated biodiesel injected into the constant volume chamber will be increased with preheating temperatures, and at 114 °C elevated fuel inlet temperature, it is approached to a similar heat value with that of diesel injection case

during normal operating condition. In addition, the lower heat value of biodiesel also increased with a heating process as indicated in **Table 5.1**. The combined results together with the increased injection quantity of biodiesel fuel, can be attributed to the increased of the heat value of biodiesel (kJ/cycle) closes to diesel operation.

5.3.1.4 Uncertainty Analysis of Fuel Properties of Biodiesel

The kinematic viscosity measurement was carried out for each sample (54-138 °C) in steps of 12 °C according to ASTM standard D445. Redwood Viscometer is used to measure the dynamic viscosity of biodiesel (uncertainty of the viscosity measurement was $\pm 0.2\%$), whereas, the standard hydrometer method is used to measure the density of biodiesel at different fuel inlet temperatures. A glass hydrometer with scale subdivision 0.5 kg/m³ (uncertainty of the density measurements was ± 0.5 kg/m³).

5.3.2 Effect of Fuel Preheating on Engine performance parameters

The performance parameters considered are BSFC and BTHE. These engine parameters are evaluated with neat POME (PBD100) and COME (CBD100) as fuel in a diesel engine were thoroughly studied at full engine load. Preheating of neat biodiesel was done from temperature of 54 °C –138 °C with increments of 12 °C. For all fuel inlet temperatures, the performance of engine was evaluated and compared. The main intention is to show the significance of preheating fuel for maximize the engine performance at lower emissions, and to investigate the optimal fuel preheating temperatures which can provide the best performance and emission parameters.

5.3.2.1 Brake Specific Fuel Consumption (BSFC)

The variation of BSFC for PBD100 and CBD100 biodiesel with a varying fuel inlet temperature corresponding to the constant engine load (100%) and speed of 1500 rpm is displayed in **Figure 5.2(a)**. It is realized that, increase a fuel inlet temperature caused to drop the BSFC of biodiesels up to 114 °C. Beyond this temperature the BSFC start to increase slightly. The reported decrease in BSFC for preheated PBD100 and CBD100 is mainly because of the drop in viscosity with increase the fuel temperature. Once the kinematic viscosity is dropped, the fuel becomes vaporized, finer and thus could improve the combustion process by using the oxygen present in biodiesel. The minimum, BSFC (0.32281 and 0.35404 kg/kW.hr) for POME and COME was obtained at maximum/optimal preheating temperature of 114 °C. The maximum BSFC (0.54542

and 0.66179 kg/kW.hr), respectively was seen at the minimum fuel preheating temperature of 54 °C. **Table 5.1** showed that, as the preheating fuel temperature at 114 °C, the energy content (heating value) of each preheated biodiesels was found increased. This contributes the lowest BSFC for maximum fuel temperature at 114 °C when compared to the minimum temperature (54°C). Lower specific fuel consumption with better performance in power output and efficiency are clearly most desirable with lesser emission (Rahim et al., 2012). The BSFC of CBD100 higher than PBD100 over varying fuel preheating temperatures, this is may be because of higher kinematic viscosity the oil contributes to the relatively higher BSFC in the diesel engine operation. Any fuel temperature beyond 114 °C, the BSFC for each preheated oils was found slightly increased. This may be due to the extreme outflow of preheated oils from the fuel pump and injector nozzle tips owing to very lower kinematic viscosity of the oil.

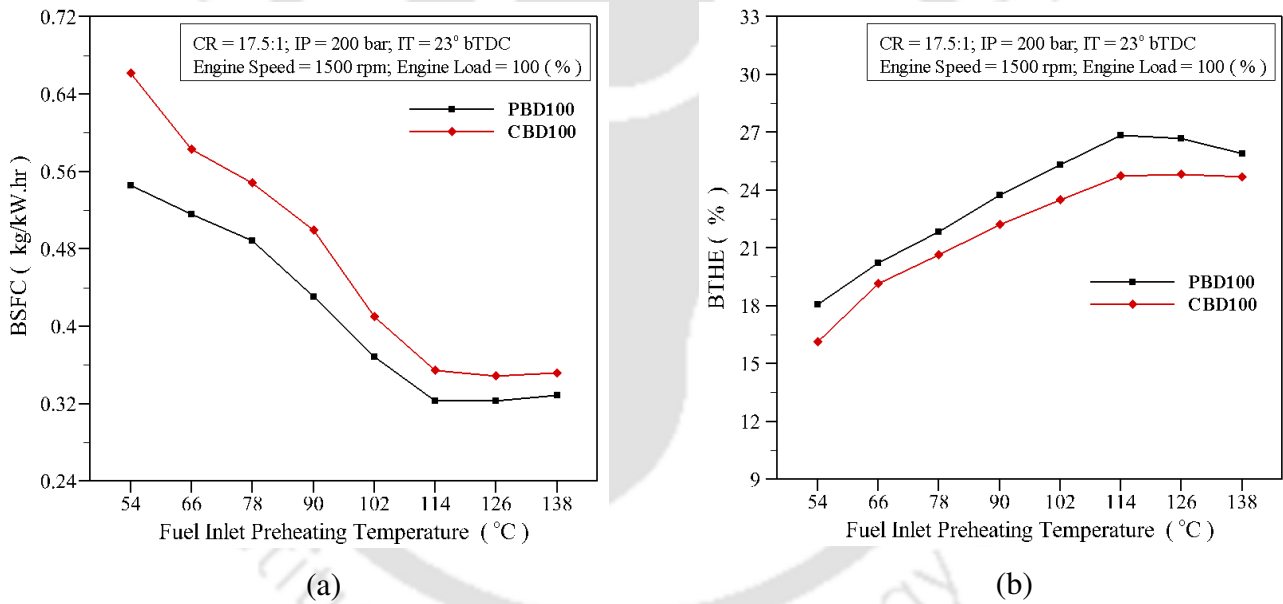


Figure 5.2: Effect of fuel preheating temperature on biodiesels: (a) BSFC; (b) BTHE.

5.3.2.2 Brake Thermal Efficiency (BTHE)

Figure 5.2(b) displays the BTHE for PBD100 and CBD100 with fuel inlet temperature at rated load (100%). The trends are opposite to that of the results of BSFC, with the increase in inlet fuel temperature. BTHE is increased due to the advancement in ignition process caused by the reduction in viscosity and improved volatility of biodiesels by means of preheating processes. The maximum, BTHE for PBD100 and CBD100 was found at 114 °C fuel temperature. With respect

to the minimum fuel temperature of 54 °C, the average increases in BTHE were obtained as, 48.4% and 53.2% respectively with PBD100 and CBD100. This is because of a better fuel atomization, vaporization and mixing rate of injected fuel achieved with a decrease of the viscosity of biodiesel with heating process. The results are similar to the literatures (Abedin et al., 2014; Qi et al., 2014). The test data indicated that preheating increases the thermal efficiency preheated PBD100 and CBD100 biodiesels at 100% of engine load. At 114 °C fuel preheating temperature, the performance indicators of edible biodiesel (PBD100) seems a reasonable match with respect to that of non-edible biodiesels (CBD100). Having a potential biodiesel fuel in future, PBD100 and CBD100 offers more reasonable fuel properties at par with diesel.

5.3.3 Effect of Fuel Preheating on Engine Exhaust Emissions

A gas analyzer is incorporated to measure exhaust gas emissions (CO, CO₂, HC, and NO_x). They have been studied for all preheated (54 –138 °C) with increments of 12 °C. Similar comparison was made for emissions of preheating fuel for reduction the emissions and to find out the optimal fuel preheating temperatures which can provide the best emission parameters as it was made in performance results.

5.3.3.1 Carbon Monoxide (CO) Emissions

The CO emission is an indication of incomplete combustion. The formation of CO emission mainly depends upon the physical and chemical properties of the fuel used. **Figure 5.3(a)** showed the CO emission of diesel engine using PBD100 and CBD100 with varying fuel inlet temperatures (54 °C–138 °C) at full engine load. The CO emission for biodiesels decrease with increasing fuel temperature up to 114 °C. The amount of the CO emission of PBD100 and CBD100 biodiesels decrease with heating. The reason is attributed to its reduced viscosity, density, and increase in rate of evaporation due to preheating (Venkanna and Venkataramana Reddy, 2013). The CO emission using heated PBD100 was found lower compared to heated CBD100. The decrease in CO emission for POME is attributed due to its high cetane number of 60.84 than that of CBD100 (**Table 5.1**). It is observed that, the CO emission of PBD100 (102°C) and PBD100 (114°C) dropped by 60.68% and 85.47%, respectively as compared to PBD100 (54 °C). Similarly, the CO emission of CBD100 (102 °C) and CBD100 (114 °C) dropped by 34.14% and 48.44%, respectively as compared to CBD100 (54 °C). However, the CO emission of PBD100(126°C) and CBD100 (126

°C), showed slightly increase by 17.21% and 23.53%, respectively compared with engine at 114 °C operating conditions for both biodiesels. This is due to lack of air (oxygen) required to make a homogenous mixture because of extra supply of vaporized fuel in combustion chamber. The energy conversion was incomplete for CBD100 (126 °C) and PBD100 (126 °C) caused with the increase in CO emission. When the fuel preheating temperature increased further to 138 °C, the level of CO emission of both biodiesel shown increased significantly due to poor mixture formations.

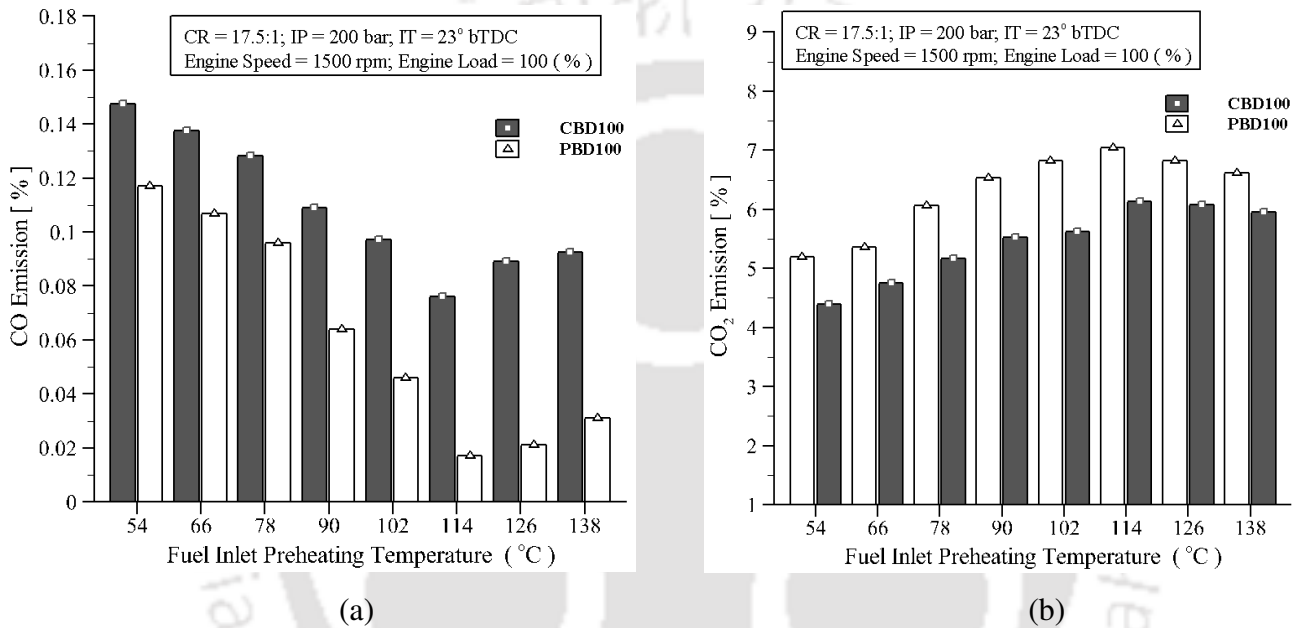


Figure 5.3: Effect of fuel preheating temperature on biodiesels: (a) CO emission; (b) CO₂ emission.

5.3.3.2 Carbon Dioxide (CO₂) Emissions

Carbon dioxide (CO₂) is the major emission contributing to greenhouse gas effect. The variation of CO₂ emissions with fuel preheating temperature for PBD100 and CBD100 is shown in the **Figure 5.3(b)**. The CO₂ emission was slightly improved using preheating for all biodiesels. Hence, higher amount of CO₂ emission of biodiesel in exhaust emission is an indication of the complete combustion of fuel. It is observed that, the CO₂ emission PBD100 and CBD100 biodiesel at 102°C and 114 °C fuel preheating temperatures, were increased by (28.2 and 39.6%) and (31.6 and 35.6%), respectively as compared to 54 °C temperature. This may be due to improvements fuel injection characteristics of PBD100 and CBD100 with heating (Yilmaz and Morton, 2011). While

further increased the preheating temperature towards 126 °C, the percentage of CO₂ emissions of both biodiesel shown decreasing trend, this may be due to the presence of fuel rich mixture due to lack of oxygen resulting from engine operation.

5.3.3.3 *Unburnt Hydrocarbon (HC) Emissions*

The HC emission of PBD100 and CBD100 with different fuel preheating temperatures at full load is presented in **Figure 5.4(a)**. It is showed that the preheated biodiesel lowers the levels of HC emissions. The reason is attributed due to improved spray pattern (due to its lower viscosity) and high rate of evaporation (due to preheating biodiesel), which leads to efficient combustion. It is seen that, the HC emission of PBD100 and CBD100 was lower at 114 °C, and it was decreased by 65.4% and 59.1%, respectively as compared to 54 °C. The improved atomization and mixing rate of running fuel with heating, results in more burned hydrocarbons during burning process and thus improves the mixture formation (Yilmaz and Morton, 2011). When the preheated temperature is beyond 114°C, it was marginally increased. This is due to the presence of fuel rich mixture due to lack of oxygen resulting from engine operation. Shortage of air (oxygen) leads to poor combustion and higher HC emissions.

5.3.3.4 *Oxides of Nitrogen (NOx) Emissions*

The variation of NO_x emission with fuel preheating temperature of different biodiesels is shown in **Figure 5.4(b)**. The formation of NO_x emissions due to the peak flame temperature and excess oxygen within the combustion chamber (Heywood, 1988). It is observed that, the NO_x emission increases for higher preheating temperature. This may be attributed to the rise in the ignition gas temperature with higher fuel preheating temperature (Gumus et al., 2012; Yilmaz and Morton, 2011). The NO_x emissions of PBD100 and CBD100 increased by average (44.33 and 58.85%) and (33.72% and 44.57%) at 102 and 114 °C, respectively compared to fuel preheating temperature at 54°C. The increase in NO_x emissions with CBD100 may be attributed to various reasons, such as improved fuel spray characteristics, better combustion of biodiesel due to its high oxygen content and higher temperatures in the cylinder as a result of preheating. This shows that the formation of NO_x is very sensitive to temperature and it increases with increase in the preheating temperature of biodiesel. This is because preheated fuel premixes more rapidly with oxygen molecules present in air resulting in the formation of increased NO_x emissions. It can be seen from **Figure 5.4(b)**,

PBD100 (114 °C) possesses the highest NO_x emission of 879 ppm whereas CBD100 (114 °C) was 799 ppm at full load. While, NO_x emission is slightly decreased when preheating temperature is increased from 126 °C to 138 °C, this may be due to a lack of oxygen for homogeneous mixture.

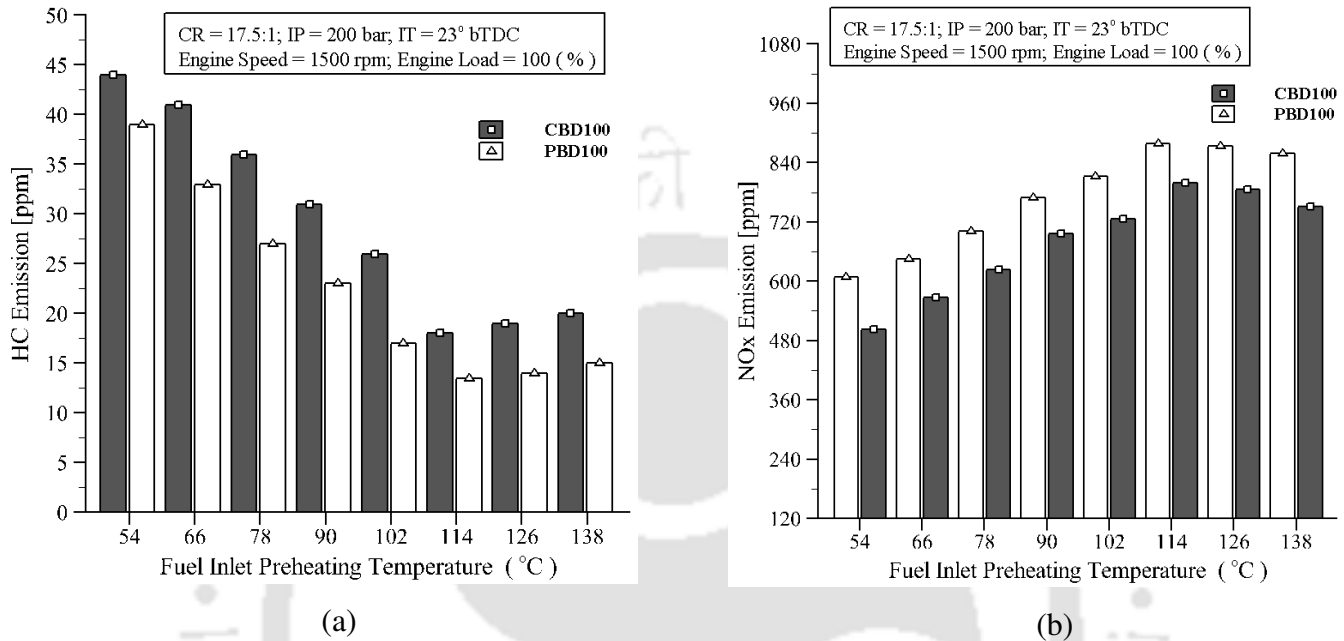


Figure 5.4: Effect of fuel preheating temperature on biodiesels: (a) HC emission, (b) NO_x emission.

5.4 Summary

The viscosity and density of two biodiesels (POME and COME) were reduced by preheating it before supplied to the test engine. After the fuel properties of each biodiesel with respect to fuel inlet temperature has been determined, various performance parameters and exhaust emissions of the engine fueled with biodiesel preheated at different temperatures were investigated and compared the variations with each fuel temperature. The result showed that, the kinematic viscosity and density of biodiesel gradually decreases as the fuel inlet temperature increase from 54 to 138 °C. According to the test results, heating of neat Palm oil biodiesel to a temperature above 102 °C brings its viscosity within ASTM D6751, EN14214 and IS1507 limit, that is, 6 mm²/s, whereas for Castor oil biodiesel the minimum preheating temperature was 114 °C. These effect cause improvements the engine performance parameters with a reductions of exhaust emissions. This is due to the reduction in viscosity, density and improved atomization, fuel-air

mixture formation and increase combustion efficiency of biodiesel as the fuel inlet temperature increases. From the properties of neat POME and COME biodiesel, and engine test results, it was observed that fuel inlet temperature of 114 °C is an optimal preheating temperature for neat biodiesel and can be substituted for diesel without any engine modification. Particularly, PBD100 (114 °C) and CBD100 (114 °C) yielded with a high improvement in the BTHE with significant reduction CO and HC exhaust emissions. However, for higher value of fuel inlet temperature above 114 °C, performance was observed to be marginally inferior, because of excessive fuel leakage (i.e. lower fuel viscosity). The upper limit of fuel inlet temperature tested was 114 °C. Thus, fuel inlet temperature of 114 °C is recommended as an optimal preheating temperature for neat POME/COME oil biodiesel. The experimental conclusions of this investigation can be summarized as follows:

- Kinematic viscosity and density of neat POME (PBD100) and COME (CBD100) are higher than those of diesel fuel. The heating value of PBD100 and CBD100 is slightly lower while its flash point and pour point is higher than diesel fuel.
- Preheating of the PBD100 and CBD100 caused a considerable decrease in its kinematic viscosity and density, thus causing them to approach the values of biodiesel standards and diesel fuel.
- Noteworthy, enhancements in engine performance (BSFC and BTHE) using preheated PBD100 and CBD100 is observed. It have been noticed, PBD100 (114 °C) and CBD100 (114 °C) yielded with a high improvement in the BSFC and BTHE. From the results, it is summarized that PBD100 and CBD100 can be used as replacement of diesel fuel in a diesel engine by decreasing its viscosity and density to international standards biodiesel limits by preheating it to a temperature of 114 °C.
- The use of preheated PBD100 and CBD100 usually yielded a significant decrease in CO and HC emissions, while NO_x emissions were increased due to higher combustion temperatures caused by preheating and oxygen content of POME/COME biodiesel.
- Regarding the variations in the performance parameters and exhaust emissions except NO_x emissions of the engine, PBD100 (114 °C) and CBD100 (114 °C) is found to be the moderate fuel among other preheated biodiesels with reference to 54 °C fuel preheating temperature. however, further studies can be carried out to decrease NO_x emissions.

CHAPTER-6

Results and Discussion: Preheated Biodiesel Blends Run Engine

Overview

Blending of biodiesel with diesel is the most common method for enhancing the biodiesel property and overcoming the flow problems. Biodiesel can blend with diesel and used as a fuel for CI engine. This blend is approved as a fuel for the existing CI engines at low blending ratio up to 20% biodiesel according to the fuel standard. They are used as a commercial fuel in many countries. However, at high blending ratios problems related to fuel properties are worsening. This is mainly due to the increases of the biodiesel percentage in the blend. Thus, high blends ratio of biodiesel cannot be used directly without bringing its properties closer to diesel. Mainly viscosity reduction is sufficient to improve its flow and atomization properties. In this study palm oil methyl ester (POME) which is edible vegetable oil, having high viscosity and low volatility has been used as a fuel in CI engine. Viscosity of POME is reduced by blending with diesel fuel along with preheating through waste heat of engine's exhaust gases, that requires heating mechanisms, which is easily fabricated and used the existing CI engine. The POME biodiesel was preheated in various blend ratios in the engine at elevated fuel preheating temperature of 114°C and the properties were evaluated. Several tests were conducted with diesel and preheated POME blends, initially diesel and preheated POME blends were characterized and then engine test run was done with preheated POME blends. The evaluation of blended fuel property is an important indicator for the maximum blending ratio of biodiesel that can meet the fuel specification requirements (ASTM D7467). Experiment has been conducted with a VCR DI CI engine to obtain engine characteristics with various preheated blend ratios (PBD0, PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100) at varying engine loads. Since biodiesel is oxygenated fuel, it offers potential reduction in CO and HC emissions. However, enhancements in NO_x emissions were noticed. The application of exhaust gas recirculation (EGR) partially resolves the aforementioned problems of the blend biodiesel run in a diesel engine. An experimental investigation was conducted to observe the effect of EGR on the NO_x emissions with varying EGR rate (EGR0%–EGR40%) with 10% increments for various biodiesel blend ratios to give an insight into the study of engine performance and emissions.

6.1 Preface

Numerous methods were introduced to advance the fuel properties of biodiesel, namely blending biodiesel with mineral diesel fuel, usage of chemical (additives), and heating biodiesel (Alonso et al., 2008; Su et al., 2011). It is reported that direct implementation all of these methods to biodiesel are not fully promising and practical. Till date, blending of biodiesel is a widely used methods for reducing the kinematic viscosity and density of biodiesel and applied as a fuel in compression ignition engine under the ASTM blended fuel standard (Knothe, 2010). However, the selection of maximum possible percentage of biodiesel ratio depends on the biodiesel feedstock and their fuel properties differ with respect to feedstock. Hence, the determination of right blend ratio is the most important criteria for utilizing biodiesel in a diesel engine. A few earlier studies indicated that, the maximum blend ratio of biodiesel currently limited to blends of 20 % or less as commercial fuels for many countries in existing diesel engine operate without engine modification (Atabani et al., 2012). Conversely, at higher percentage of blend difficulties associated to degradation of fuel performance (Smith et al., 2010). Therefore, important fuel properties of blends of biodiesel need modification in order to make improvement for utilization of higher blend ratios of biodiesel in diesel engines. Thus, modifying fuel properties of biodiesel through heating method at specific temperature before blending with diesel improved the fuel performance of blends of biodiesel run as a fuel in a diesel engine without modification engine or fuel injection system (Khiraiya Krunal et al., 2013; Martin et al., 2017; Rambabu et al., 2013). There are limited research studies for use of preheated biodiesel–diesel blend in a diesel engine. Rahim et al. (2012), investigated the effect of fuel temperature on the performance parameters and combustion characteristics of diesel engine fueled with biodiesel blends of fuel at varying engine speed. The result showed that, preheating biodiesel at higher temperature causes shorter ignition delay (early start of combustion) thus resulting increased in-cylinder gas pressure. Preheating also increased the energy content of fuel which results better combustion and lower the brake specific fuel consumption of the diesel engine. Dinesha and Mohanan (2012), worked on investigation on engine performance and emission parameters of diesel engine running with preheated pongamia methyl ester as fuel. The result indicated that, preheating improves the viscosity and density of biodiesel. Subsequently, engine running with preheated and blends of biodiesel significantly improves in the brake thermal efficiency and UBHC emission and smoke opacity. The fuel preheating of methyl ester biodiesel at optimal temperature can improve overall engine performance parameters with reductions

emissions of CO and HC emission at expense of NO_x emission and can be substitute a diesel fuel without any engine modification. Further, the effect of fuel preheating and blend ratios on engine performance and emission evaluation of diesel engine operated with blend mixtures has been investigated (Patel and Shrivastava, 2016). It is indicated that, 80% preheated biodiesel- diesel blend and 20% ethanol can offer reasonably performance improvements with reduction of CO, HC and NO_x emissions at varying load conditions as compared diesel. The effect of exhaust gas recirculation (EGR) on performance and emission characteristics of diesel engine fueled with 20% biodiesel was investigated (Dhana Raju and Kishore, 2018). The effect of EGR on NO_x emission of diesel engine running with different biodiesel-diesel blends of fuel was examined by (Saravanan, 2015). It was reported that, the NO_x and smoke emissions were significantly reduced for blends with reasonable drop in engine performance and slight increase exhaust emission (CO and HC) as compared to diesel fuel.

In the present study an attempt has been made to enhance the possibility of using a high biodiesel blends with an objective to maximize the renewable biodiesel fraction in the fuel using a combined methods of heating and blending in order to make blend ratios suitable as a fuel for use in a CI engine. Then, evaluate the performance, combustion characteristics and emission parameters with new alternative fuel are evaluated and compared in to diesel fuel. In this investigation, EGR was also implemented to see the effect of EGR on NO_x exhaust emission of diesel engine running with preheated biodiesel/diesel blends.

6.2 Material and Methods

6.2.1 Preheated Biodiesel/Diesel Blend Fuels

In this study, palm oil methyl ester (POME) biodiesel has been blended and preheated before going through properties and performances test to analyze the quality of POME biodiesel blend as an ignition fuel. Five preheated biodiesel blends were prepared by mixing preheated biodiesel with diesel at blending ratios of 20%, 40%, 60%, 80% and 100% by volume basis, which corresponded to PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 blends, respectively as shown in **Figure 6.1**. The properties of all tests fuels were measured following ASTM test methods and are shown in **Table 6.1**. Provision was made for preheating of biodiesel to the required levels (114 °C) before blending with diesel, so that its viscosity and density biodiesel was equalized to that of diesel fuel at room temperature. Then, the characterization of the different properties of preheated POME

biodiesel blends have been determined and compared with blended biodiesel standard (ASTM D7467 and BIS). During experimentation, all the blends were stirred well using a mixture to ensure homogeneity just before fueling. Then, experiments were carried out with different blends of preheated biodiesel from palm oil methyl ester mixed with diesel in concentrations of 20%, 40%, 60%, 80% and 100% to examine engine performance parameters, combustion characteristics and emission levels of a VCR DI compression ignition engine. The collected data were analyzed for various parameters such as BSFC, BTHE, EGT, volumetric efficiency, peak cylinder pressure (PCP), heat release rate (HRR), ignition delay (ID), CO, HC, CO₂ and NO_x emissions. To make further analysis for this study, diesel fuel also has been tested to make the comparison between diesel fuels and preheated biodiesel blends result. It is expected that this work will provide the knowledge of the effects of fuel properties due to the variant in biodiesel blending ratio on performance, combustion and emission characteristics of a diesel engine.

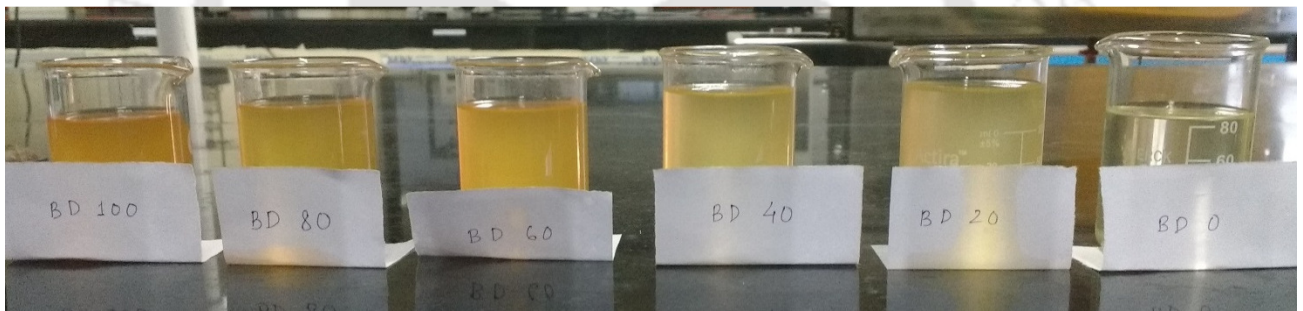


Figure 6.1: Samples of biodiesel blends for property test.

6.2.2 Facilities and Test Procedures

Tests were performed in a VCR DI, water cooled, constant-speed (1500 rpm), single-cylinder, compression ignition engine. specifications of the engine are given in **Table 3.1**. The schematic arrangement of experimental set up including the preheating mechanisms are shown in **Figure 3.15**. The setup consists EGR system, fuel tanks with a fuel switching control valves, one for normal diesel fuel (PBD0) and the other for biodiesel, and fuel filter “F” is fixed at the biodiesel supply line (before preheater) for removing impurities from high viscous biodiesel and to minimize clogging of fuel supply line (refer **Figure 3.15**). The test engine is interfaced with exhaust gas

analyzer (Make: AVL India, Model: DIGAS 444) for measuring important exhaust emission parameters samples (CO, HC and NO_x).

Table 6.1: Properties of diesel, unheated POME biodiesel and preheated biodiesel blends.

Fuel blend		Kinematic viscosity (mm ² /s)	Density (kg/m ³)	Pour point (°C)	Flash point (°C)	Calorific value, MJ/kg
Diesel (PBD0)		3.23 at 40 °C	838 at 27 °C	-8	63	44.69
POME Biodiesel (PBD100)		8.71 at 40 °C	946 at 27 °C	5	265	38.29
PPBD20 at 114 °C		3.38	851	-1	103	44.12
PPBD40 at 114 °C		3.76	859	2	141	43.32
PPBD60 at 114 °C		4.08	863	4	181	42.67
PPBD80 at 114 °C		4.86	869	6	226	41.78
PPBD100 at 114 °C		5.37	876	8	289	40.12
Blended biodiesel Requirements	ASTM D7467	1.9-4.1	820-860	-	min. 52	-
	BIS	2.0-4.62	820-860	max. 3	min. 35	-

PBD0–neat diesel fuel, PBD100–neat POME biodiesel, PPBD–preheated POME biodiesel

The engine was first run with diesel fuel which forms the reference fuel for this study and baseline data was recorded. The engine always ran for 15 min at each load condition to allow for stabilization before the readings were recorded. Each test was repeated three times to ensure repeatability. The engine was always allowed to run for some time before changing the blends in order for it to consume the fuel that remained in the fuel system during the previous trail. After the characterization various preheated biodiesel blends fuel properties, experiments were performed using these blends for investigation of engine performance, combustion and emission characteristics in a compression ignition engine as per text matrix (**Table 3.3**). The engine was first tested with varying engine load from no-load to full load (0%–100%) with 20% increments. Experiment was also conducted with variation of EGR rates (0%, 10%, 20%, 30% and 40%) with the same preheated biodiesel blend of oils. Finally, the optimized preheated biodiesel blends were compared with diesel fuel for performance and emission characteristics. For recirculation of the exhaust gas, appropriate pumping was done. No insulation on the pipe line was provided therefore allowing the recirculated exhaust gases to partially cool down. The schematic diagram of the EGR system is shown in **Figure 3.15**. The quantity of EGR can be regulated by a control valve “gv3” installed in the EGR loop. An air box was provided in EGR loop to dampen the fluctuations of the pulsating exhaust. An orifice was installed in the EGR loop to measure the flow rate of exhaust

gas. Sufficient mixing of incoming air and recirculated exhaust gas is ensured inside the mixing chamber before it gets induced into the combustion chamber.

6.3 Result and Discussion

In the present study several tests were conducted with preheated biodiesel blends. Initially diesel and preheated POME biodiesel/diesel blends were characterized and then engine test run was done with diesel (PBD0) and various blend oils (PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100).

6.3.1 Characterization of Preheated POME Biodiesel Blends

The preheated POME biodiesel was further blended with diesel at different volumetric ratios to study the of the kinematic viscosity, density, flash point, cetane number, pour point and calorific value of the preheated POME biodiesel/diesel blends. Most properties of the preheated POME biodiesel “PBD100 (114 °C), including the kinematic viscosity, density and flash point, were slightly higher, but close to those of diesel fuel (refer **Table 5.1**). Thus, the preheated palm oil methyl ester (POME) has been blended with diesel and the variation of different properties of blended fuel properties at varying blending ratio are predicted experimentally. **Figure 6.2(a–b)** shows variation of kinematic viscosity and density with increase in percentage blending of POME biodiesel diesel fuel. The POME biodiesel are preheated for the temperature of 114 °C before blending with diesel in different percentage fractions of blend ratios. All of these properties were evaluated in the Laboratory. Preheated POME biodiesel oil and diesel were characterized and results have been shown in **Table 6.1**. Blending of diesel with preheated POME biodiesel can cause some improvement in some properties such as kinematic viscosity, density and calorific value. However, the flash point and cetane number decrease as the percentage of diesel increases. The measured fuel properties have been compared and adopted for a certain range of the biodiesel blending ratios standard requirements of ASTM D7467 and BIS.

6.3.1.1 Effect of Fuel Preheating and Blending on Kinematic Viscosity

Figure 6.2(a) shows variation of kinematic viscosity with increase in percentage blending of POME biodiesel diesel fuel. The kinematic viscosity of PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 are 3.38, 3.76, 4.08, 4.86 and 5.37 mm²/s, respectively. The kinematic viscosity of PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 blends falls from 5.37 to 3.38% at the same temperature, and for PPBD60 blend it is 53.16% less than unheated PBD100 (8.71 mm²/s). The

results reveals that the kinematic viscosity of each blends of fuel significantly decreased with heating of biodiesel at elevated temperature. However, the blended fuel viscosity still meets the blended fuel standard requirements ASTM D7467 for up to 60% biodiesel blending ratio (PPBD60). The viscosity of the blended fuel PPBD60 is 4.08 mm²/s, which is higher than that of diesel fuel (3.23 mm²/s) by about 1.26 times. However, the blended fuel viscosity for up to 100% (5.37 mm²/s) satisfies the limits given in the specification of biodiesel standard requirements of ASTM D6751, EN14214 and IS15607, which is 6.0 mm²/s and it is about 1.66 times higher than the fossil diesel at room temperature.

6.3.1.2 *Effect of Fuel Preheating and Blending on Density*

Figure 6.2(b) shows the variation of density with volumetric percentage of preheated POME biodiesel-diesel blend. It is observed that the density increases linearly with an increase in the blending ratio due to an increase in the volumetric percentage of biodiesel in the blend ratios as the density of POME biodiesel is higher than diesel. Accordingly, the density of the blended fuel PPBD60 is 1.5% lower than that of PPBD100 due to the effect of preheating biodiesel and blending with diesel. It was observed in **Figure 6.2(b)** that, the density of the preheated blends decreased marginally for all the tested fuels. The density of PPBD20 and PPBD40 were within the blended fuel standard requirements ASTM D7467 and BIS, in ranges of 851–859 kg/m³, while the blends PPBD60, PPBD80 and PPBD100 are out of the blended fuel standard requirements, but only conformed to the ASTM D 6751, EN14214 AND IS15607 standard specification when heated to temperatures of 114°C to lower the density to the specified maximum of 900 kg/m³.

6.3.1.3 *Effect of Fuel Preheating and Blending on Calorific Value*

Calorific value is an important parameter in the selection of a fuel. It is the amount of heat released during the complete combustion of one gram of fuel. This value determines the chemical energy of the fuel which specify the combustion efficiency. **Table 6.1** shows that the heating value of the preheated POME biodiesel-diesel blend decreased with a higher volumetric percentage of the biodiesel. The minimum heating value measured was 40.12 MJ/kg for PPBD100, which satisfies the requirements of the EN 14213 Standard. The heating value for POME biodiesel is approximately 14.32% lower than the heating value of mineral diesel (44.69 MJ/kg). **Figure 6.3** shows variation of calorific value with preheated POME biodiesel/diesel blending ratio. In contrast to density and kinematic viscosity, it shows a decrease in calorific value of PPBD20, PPBD40,

PPBD60, PPBD80 and PPBD100 with increasing biodiesel ratios in blended fuel (44.12, 43.32, 42.67, 41.78 and 40.12 MJ/kg). The decrease in calorific value is due to the increase in biodiesel in the mixture of biodiesel-diesel blend as the calorific value of biodiesel is lower than diesel. Increase in calorific value is expected to improve the engine combustion efficiency, thus reduce the BSFC of the engine.

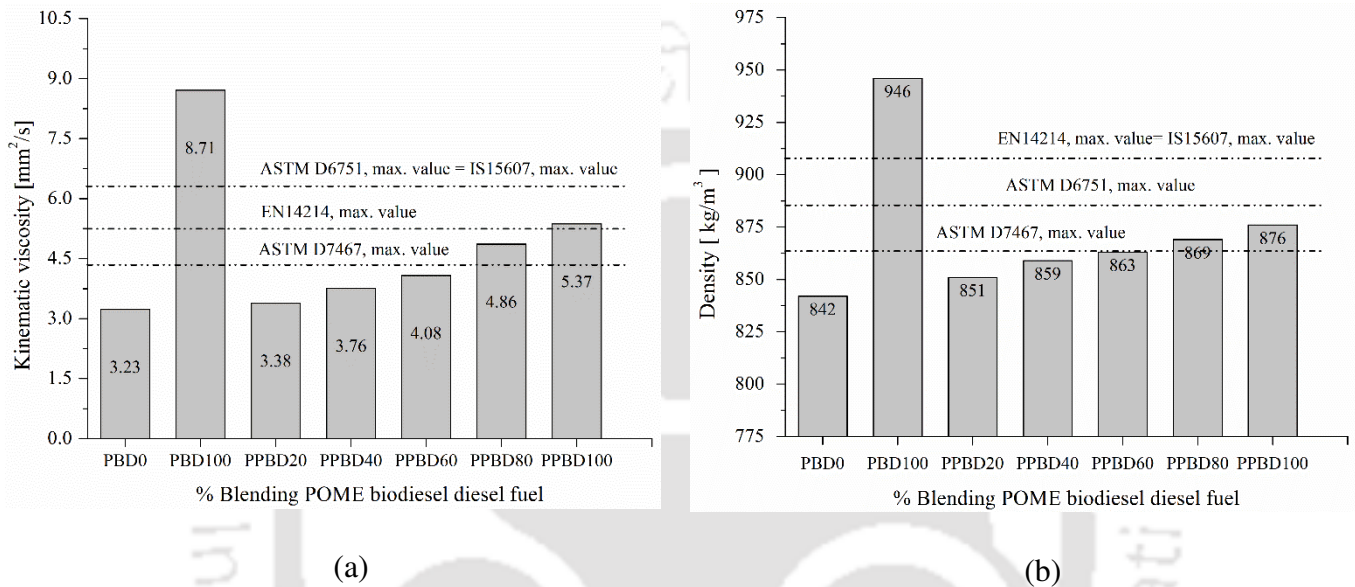


Figure 6.2: Variation of fuel property with percentage of preheated biodiesel blends: (a) kinematic viscosity; (b) density.

6.3.1.4 Effect of Fuel Preheating and Blending on Flash Point

The flash point is the minimum temperature at which a fuel emits enough vapour to get ignited. The higher the flash point, the higher the temperature required to ignite the fuel. On the one hand, it is better to lower the flash point for combustion purposes. However, the higher flash point means the fuel is safer to transport. Generally, higher flash point is preferred. **Table 6.1** presents the variation of flash point with preheated biodiesel blending ratio. It varies from one blend to another. Like pour point, it also shows an increasing tendency with increase in blending ratio as the flash point is higher for POME biodiesel.

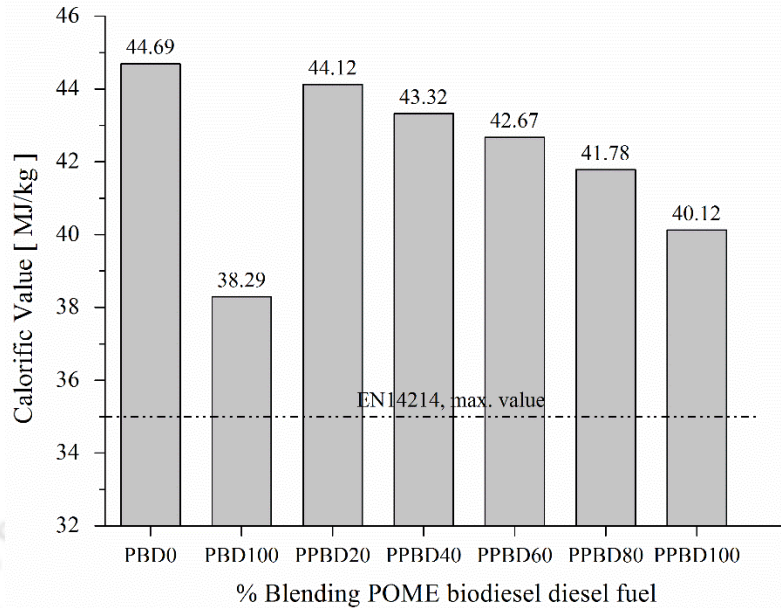


Figure 6.3: Variation of calorific value with percentage of preheated POME biodiesel blends.

6.3.1.5 Optimum Preheated Blending Ratios

When preheated biodiesel is blended with diesel the characteristics of the new fuel changes as per the volumetric percentage of biodiesel in a mixture of biodiesel and diesel. The blending ratio will be optimum when the blended fuel will satisfy the different standards as specified by the different countries. Accordingly, an optimum blending ratio was recommended for the POME at the ratios of 20% to 60% according to the properties of the POME–diesel blends. The optimal blends (PPBD20–PPBD60) would have the kinematic viscosity of 3.38 mm²/s to 4.08 mm²/s and density of 851 kg/m³ to 863 kg/m³ which comply with the limits of the blended biodiesel standard specifications. The calorific value (43.12 MJ/kg –41.67 MJ/kg), flash point (103 °C–181°C) and pour point (-1°C to 4 °C) are above the lower limit (35 MJ/kg, 52 °C and 3 max.), respectively. So an optimum blending ratio of up to 60% blending (up to PPBD60) has been recommended by compliance of the preheated biodiesel-diesel blend specification to the specified standards, which can be used in diesel engines without modification and without sacrificing much power loss.

6.3.2 Effect of Preheated Biodiesel Blends on Engine Performance Parameters

The study of the engine was carried out with diesel fuel for the baseline data and then test run was done on various blends (PPBD20–PPBD100) with 20% increments. This tests were conducted in

two phases for each performance parameters (BSFC, BTHE, EGT, and volumetric efficiency). In the first phase, tests were carried out at varying engine loads (0%–100%) with 20% increments without EGR rate, and results of various preheated biodiesel blends were compared with diesel. In second phase, similar experiments were performed at varying EGR rates (10%–40%) in steps of 10% under full engine load, and the results were compared to with no-EGR operating conditions. Accordingly, an optimum blending ratios were recommended for the preheated POME biodiesel blends, for the considerable utilization of high blend ratios of preheated POME biodiesel as a fuel in a diesel engine. The results of the experiments are presented in the following sub-sections.

6.3.2.1 Effect of Preheated Biodiesel Blends on BSFC

Figure 6.4(a) shows the effect of preheated POME biodiesel/diesel blends on BSFC as a function of engine load without EGR. It is seen in BSFC all tested fuels gradually decreases with increasing engine load in a similar trend as mentioned in references (Chauhan et al., 2010; Rambabu et al., 2013). It is mainly due to better combustion characteristic due to improved fuel-air mixing rate at high engine loads. At full load and with respect to PPBD100, the average reduction of BSFC for PPBD20, PPBD40, PPBD60 and PPBD80 was found to be 32.74 %, 28.79 %, 17.78 % and 10.14 %, respectively. It can be seen BSFC increases with increase in preheated POME biodiesel quantity in the blend as more amount of blends are required to produce the same amount of power by the engine due to the less energy content of POME biodiesel when compared to diesel fuel. It clearly indicates that PPBD20–PPBD60 offered better BSFC compared to other tested fuels. With respect to pure diesel (PBD0), the BSFC is seen to be 7.8%, 11.1% and 21.5% higher for PPBD20, PPBD40 and PPBD60.

The effect of increasing EGR rates on BSFC for preheated POME biodiesel/diesel blends at full engine load is presented in **Figure 6.4(b)**. BSFC generally increases for all preheated POME biodiesel /diesel blends at increasing EGR rates with the consumption being more for PPBD60 blends (Kegl et al., 2013; Qi et al., 2009). This is because of the reduction in oxygen availability in cylinder for combustion, which leads the reduction in in-cylinder temperature due to the application of EGR leading to incomplete combustion. It is indicated that BSFC for PPBD100 is higher for every EGR rates as compared to other tested fuels. This is because of reduction in in-cylinder gas temperature due to higher oil viscosity leading to incomplete combustion. However, BSFC for PPBD20 to PPBD60 are seen to be decreased as compared to other test fuels. Thus, it

offers improved fuel properties, increased in-cylinder temperature, better air-fuel mixing process and combustion characteristic.

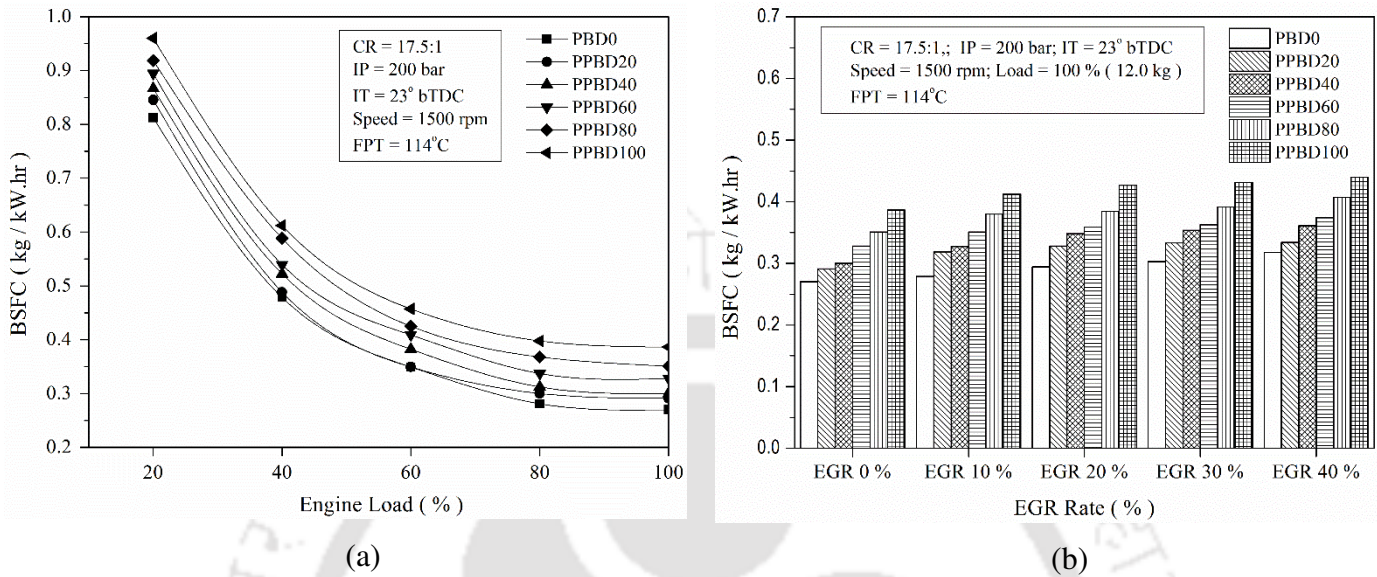


Figure 6.4: Variation of BSFC: (a) With loads (no-EGR); (b) With EGR rates at full load.

6.3.2.2 Effect of Preheated Biodiesel Blends on BTHE

By referring to **Figure 6.5(a)**, the BTHE, show opposite trends with respect to BSFC. It shows the effect of preheated POME biodiesel/diesel blends on brake thermal efficiency as a function of engine load without EGR. It can be seen that BTHE increase with engine load but decreases slightly from 90% to 100% loads. This slight drop could be due to the low excess air ratio at high engine loads that worsened the combustion (Heywood, 1988). It can be seen that preheated POME biodiesel/diesel blends have lesser BTHE than diesel. Increasing preheated POME biodiesel fraction in the blend caused a drop in BTHE due to the less energy content of POME biodiesel when compared to diesel fuel. A maximum drop of BTHE for PPBD20, PPBD40, PPBD60 and PPBD80 were 22.67%, 17.86%, 12.2% and 5.2%, respectively as compared to PPBD100 at full engine load. Thus, it is understood that, BTHE for POME was significantly improved with a combined effects of heating and blending with diesel fuel. Relating to neat diesel (PBD0), the BTHE of PPBD20, PPBD40 and PPBD60 was lower 2.16%, 6.0% and 10.5%. Hence, PPBD20, PPBD40 and PPBD60 test fuel offers the most efficient BTHE as compared with other tested fuels.

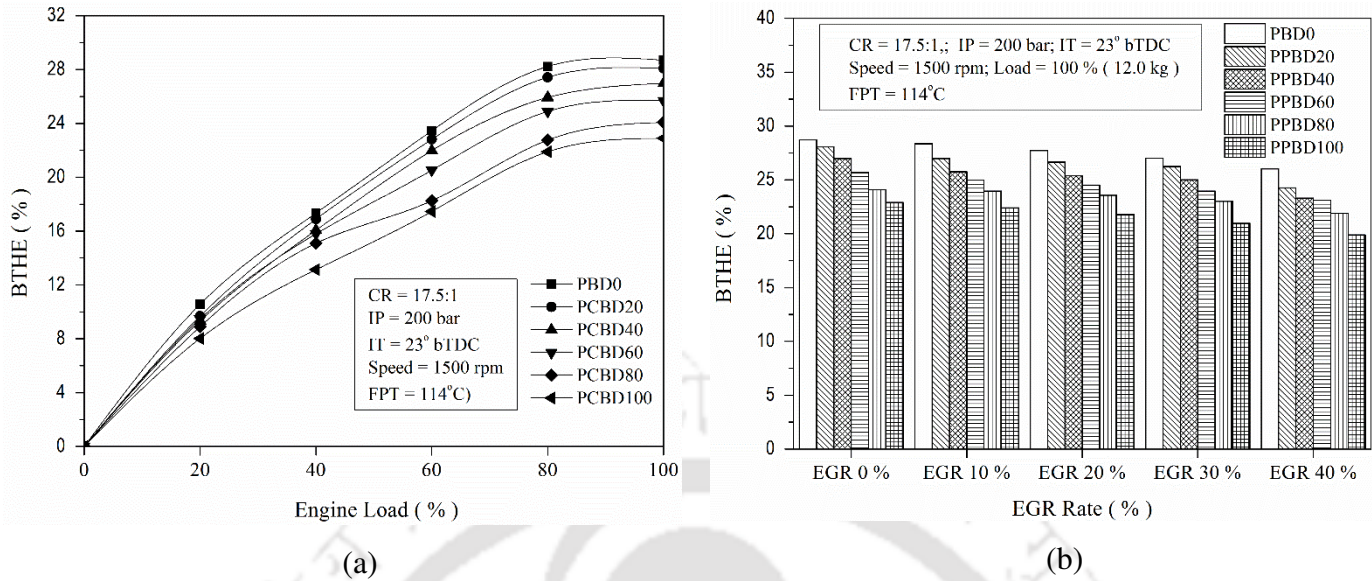


Figure 6.5: Variation of BTHE: (a) With load (no-EGR), (b) With EGR rates at full load.

The effect of increasing EGR rates on preheated POME biodiesel/diesel blends at full engine load (12 kg) was presented in **Figure 6.5(b)**. It can be seen that BTHE decreases very slightly for all preheated POME biodiesel/diesel blends at increasing EGR rates (Agarwal et al., 2011; Saravanan, 2015). Increase in EGR impedes the normal combustion process and reduces the burning rate. The increase of the percentage of exhaust gas recirculated to the engine threatens the normal engine combustion process because it decreases in-cylinder gas temperature along with oxygen deficiency. The average percentage decrements of BTHE for PPBD20, PPBD40 and PPBD60 test fuel at 30 % EGR rate were 6.53%, 7.36% and 6.75%, respectively as compared to no-EGR rate (0%EGR). However, at 40% EGR rate the BTHE for different blends of fuels are found decreased significantly as compared to no-EGR rate, thus BTHE at 30% EGR rate for PPBD20–PPBD60 test fuel found to be the optimum as compared to no-EGR rate studied in these investigations.

6.3.2.3 Effect of Preheated Biodiesel Blends on EGT

Another performance indicator of the fuel quality during combustion process is the EGT. **Figure 6.6(a)** shows the variation of exhaust gas temperature as function of engine load under no-EGR. It can be seen that exhaust gas temperature increases with increasing preheated POME biodiesel content in the blend. It essentially means fuel energy has been efficiently used at higher loads

because of improved oil quality and the presence of oxygen content in biodiesel. It is seen that all the test fuels of POME have higher EGT as compared diesel. The higher value of EGT implies increased in-cylinder gas temperature and thus ensures efficient combustion characteristics of the fuel. Since the biodiesels have higher Cetane numbers, the premixing time reduces and combustion phasing moves earlier towards the compression stroke. In addition, the occurrence of higher oxygen content in POME facilitates better combustion and causes EGT to be higher. The earlier experimental findings do highlight similar results (Pradhan et al., 2014). In general, the preheating ensures higher EGT but with lower blending ratio, the EGT drops with heating. At full load condition, the overall increase of EGT for PPBD100, PPBD80, PPBD60, PPBD40 and PPBD20 were, 20.54 %, 16.84%, 12.93 %, 9.9% and 8.03 %, respectively, as compared to diesel (PBD0) even though it has higher calorific value.

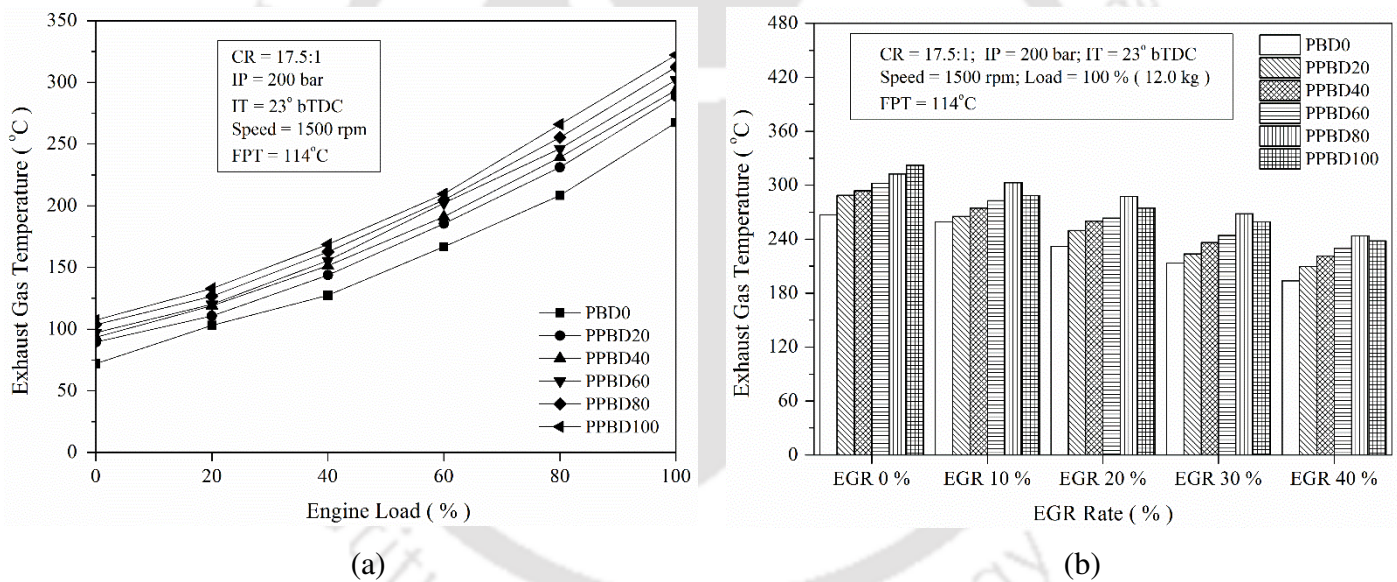


Figure 6.6: Variations EGT (a) With load (no-EGR), (b) With EGR at full load.

Figure 6.6(b) shows effect of increasing EGR percentages on exhaust gas temperature for all preheated biodiesel/diesel blends at full engine load. It can be seen that the exhaust gas temperature slightly decreases in EGR rates for all blends as a result of oxygen in combustion chamber and reduction in peak combustion temperature brought about by the increase of EGR conditions (Agarwal et al., 2011; Saravanan, 2015). At EGR 40%, the overall reduction of EGT is about 25% for all test fuels as compared to EGR0% (without EGR).

6.3.2.4 *Effect of Preheated Biodiesel Blends on Volumetric Efficiency*

Figure 6.7(a) illustrate the effect of preheated POME biodiesel/diesel blends on the volumetric efficiency with variation of loads under no-EGR. For all fuels, it drops slightly with increase of loads (Rambabu et al., 2013). This is because of increased in-cylinder gas temperature at higher engine load leading to higher gas pressure in the cylinder. Hence, it restricts the air quantity at inlet of the engine cylinder resulting slightly drop of the volumetric efficiency of the engine with increasing load. The effect of preheating POME helps to improve volumetric efficiency of diesel engine. The heated POME blend fuels causes improvement in fuel injection characteristics and combustion efficiency leading to increased amount of air intake. Due to this fact, there is a slight increase of volumetric efficiency, but it is still somehow lower than the mineral diesel fuel. At 100% loading condition, the average value of volumetric efficiency is about 80 % for all test fuels including diesel. The variations of volumetric efficiency for different fuels with EGR rates at full engine loading condition is shown in **Figure 6.7(b)**. All the test fuels show marginal drop (2.5 %) with increasing the EGR rates which is in line with reported literatures (Agarwal et al., 2011). It is mainly due to the reduction of amount of intake air mass flow into engine cylinder for combustion and supply of exhaust gases to in the engine cylinder through EGR system.

6.3.3 **Effect of Preheated Biodiesel Blends on Combustion Characteristics**

The “peak cylinder pressure (PCP) and ignition delay (ID) are two important parameters that needs to be evaluated for test fuels while looking into engine combustion behavior at different loads (0%–100%) with 20% increments without EGR rate. Then, with variation EGR rates (10%–40%) in steps of 10% under full engine load and the results were compared to with no-EGR (0%) operating conditions. Both PCP and ID indicate the fuel efficiency during combustion and they are inversely related i.e. higher ID implies lower PCP and vice versa. The combustion characteristics results of the experiments are presented in the following sub-sections.

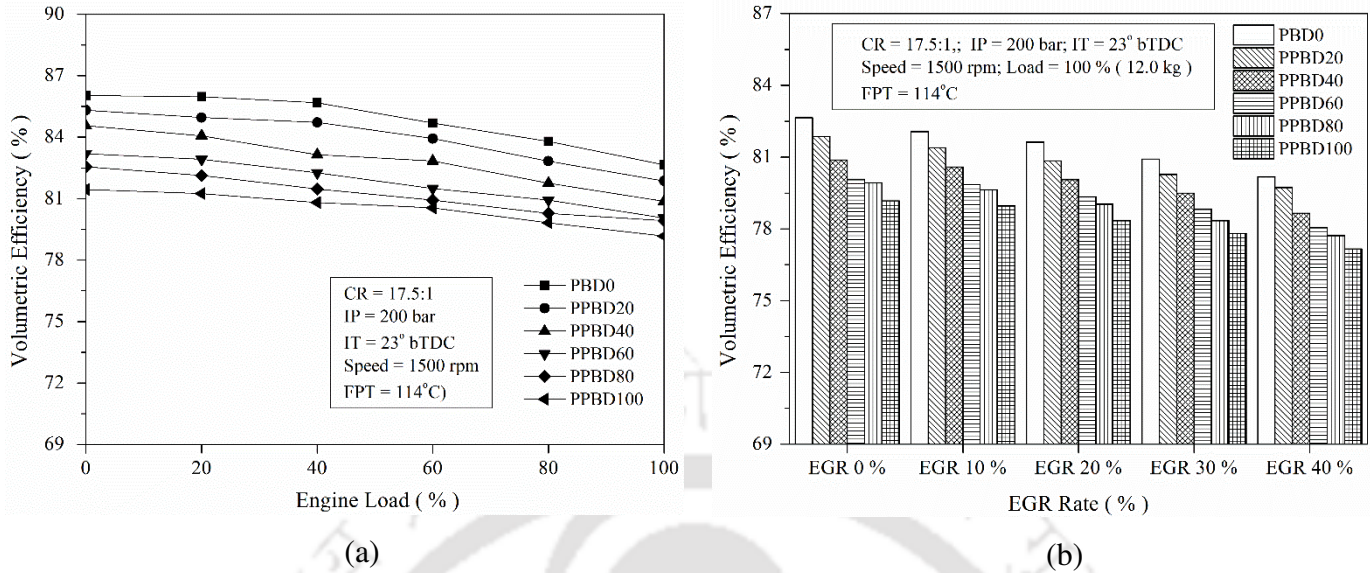


Figure 6.7: Variation of volumetric efficiency for fuels: (a) With loads (no-EGR), (b) With EGR rates at full load.

6.3.3.1 Effect of Preheated Biodiesel Blends on PCP

Figure 6.8(a) demonstrate the variation of peak cylinder pressure (PCP) for different test fuels with engine load without considering EGR rate. It is seen that the PCP all tested fuels were increased with increasing load (Rambabu et al., 2013). At lower loads up to 40%, the PCP values of preheated biodiesel (PPBD100) are highly 8.9% as compared to diesel (PBD0). All test fuels have identical values (71 bar) of PCP at higher loads. It is because of a shorter ID for biodiesels, thereby injection timing (IT) is advanced because of a higher bulk modulus and density of POME. Even the preheated POME (high density) has lower ID, the oxygenated nature of the fuel improves ignitability and produced maximum cylinder gas temperature even within the existing delay period (Qi et al., 2009). The PCP for diesel fuel is less at lower engine load due to a longer ID as compared to biodiesel and occurred in working stroke far from the position of top dead center (TDC). However, at higher load, PCP occurred at near to TDC at same crank angle position with biodiesel. It is realized in **Figure 6.9(a)** that, when the load increases, the ID for test fuels (PPBD100, PPBD80, PPBD60 and PPBD40) is shortened that lead to start of ignition before TDC and PCP increases more rapidly. The preheating and blending POME improved the fuel properties of biodiesel resulting marginal increase of ID compared to preheated PPBD100.

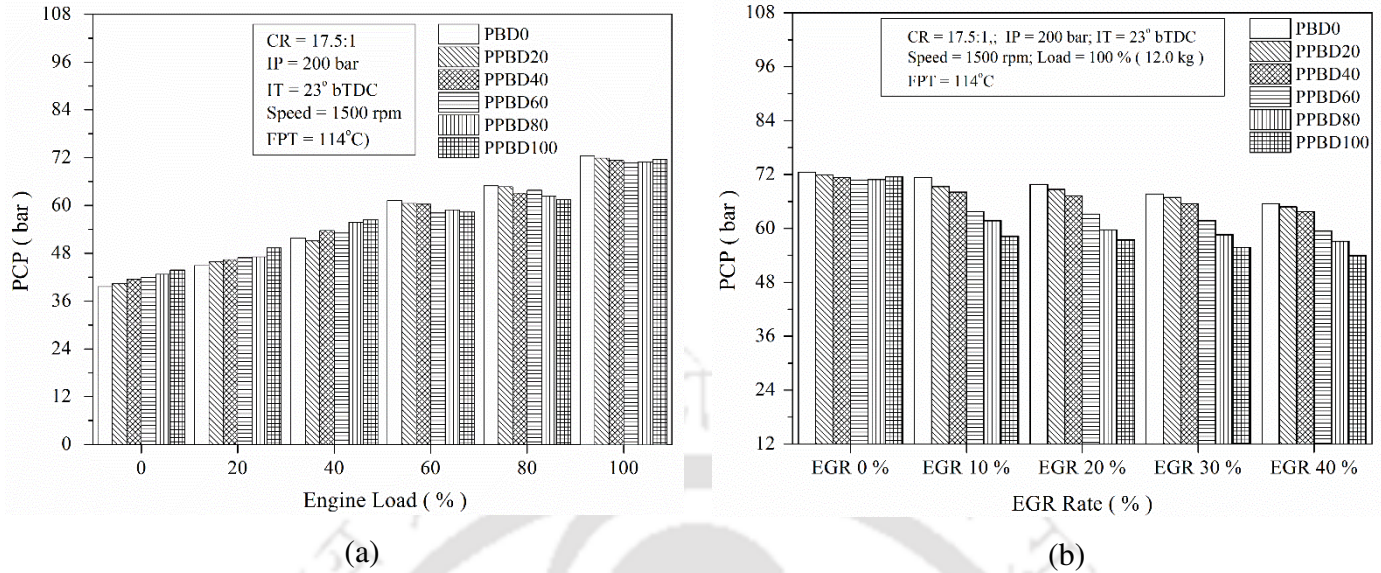


Figure 6.8: Variations of PCP for fuels: (a) With loads (no-EGR), (b) With EGR rates at full load.

The effect of EGR rate on PCP for different test fuels at full load is shown in **Figure 6.8(b)**. The PCP for all fuels decreases with higher EGR rate. This is because of the shortage of oxygen in the combustion chamber due to the occurrence of higher CO_2 in the exhaust gas. These gases can simply dilute the intake air availability in the combustion chamber and affect the air-fuel mixing process and combustion characteristics. With respect to diesel (PBD0), the test fuel PPBD20 to PPBD60 at EGR30% rate, the drop in PCP is 1.04%, 3.11% and 8.73%, respectively with the same 30% of EGR rate, and (6.95%, 8.13% and 12.73%) as compared to the engine operated without EGR (no-EGR).

6.3.3.2 Effect of Preheated Biodiesel Blends on ID

In a diesel engine, the time lag between fuel injection and combustion initiation is called “ignition delay” (Heywood, 1988). **Figure 6.9(a)** shows the variation of ignition delay (ID) for all preheated POME biodiesel/diesel blends with reference to diesel for the entire load spectrum of the engine with no-EGR. It can be seen that ID gets longer with increasing preheated POME biodiesel content in the blend fuel. The increase is obviously due to the decrease in Cetane number of diesel brought by the addition of preheated POME biodiesel. It is also seen that the ID for all tested fuels decreases with increasing load. This is a result of the increase in-cylinder gas temperature with increasing load, ensuring a shorter ID for the tested fuels. For PPBD100, the IDs were shorter compared

to other test fuels over the operating loading conditions (Rajasekar and Selvi, 2014). In this case, the higher Cetane number of POME plays a vital role on affecting the chemical delay period for ignition compared to other tested fuels properties. At full load, the IDs for preheated POME (PPBD100) and diesel fuel (PBD0) was 13° CA and 18° CA respectively. The combined effect of fuel preheating and blending caused to decrease the ignition delay as compared to preheated neat POME biodiesel “PPBD100” (Figure 6.9-a). At full engine load, the IDs for PPBD20, PPBD40, PPBD60 and PPBD80 were less by 5.56%, 11.11%, 16.67% and 22.22%, respectively as compared to diesel fuel (PBD0). This is due to a decrement of kinematic viscosity and density of POME with the effect of heating and blending.

Figure 6.9(b) illustrates the effect of increasing EGR rates on the ignition delay of preheated POME biodiesel/diesel blends at full engine load. The variation of IDs for test fuels shows increasing trends with EGR rate. This due to the supply of more exhaust gas to the engine cylinder that decreases the quantity oxygen needed for combustion. Hence, ID increases causing to drop in combustion efficiency. One of the important effects of EGR is prolongation of ID (Chen et al., 2014). Hence, increase in EGR rates increased the ignition delay for all blends. With respect to the test fuel PPBD20, PPBD40 and PPBD60 at EGR30% rate, the drop in ID is 9.52%, 14.2% and 14.3% as compared diesel (PBD0) with same EGR rate, and increased by 11.76%, 12.5% and 20% as compared to the engine operated without EGR.

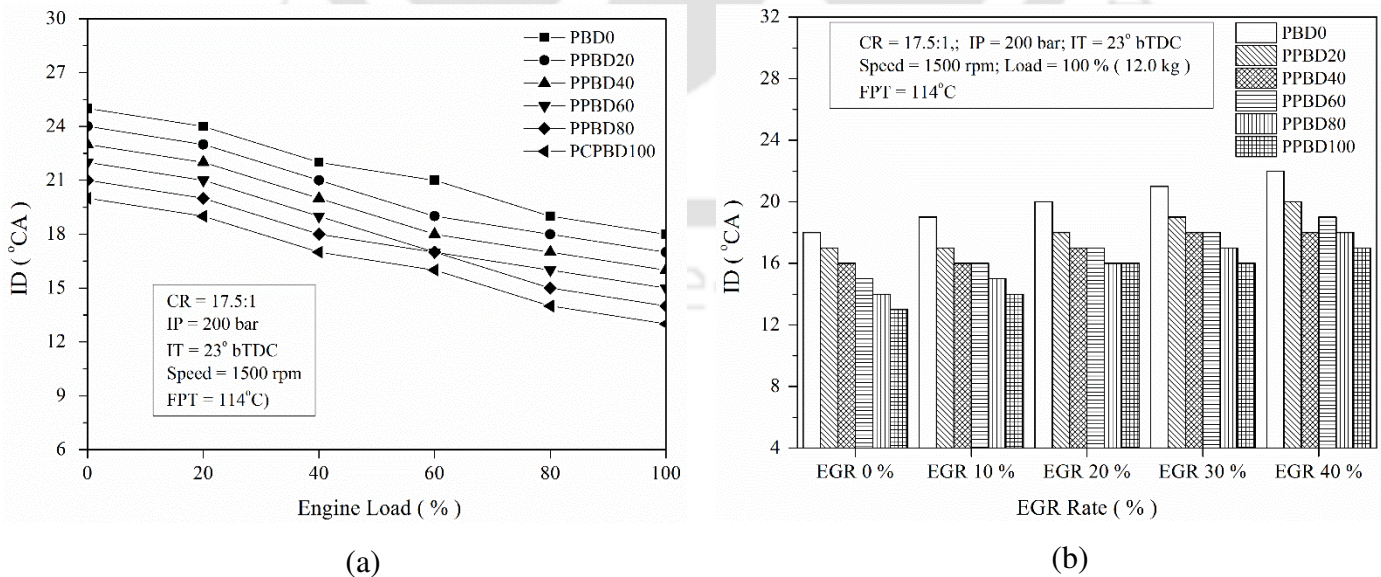


Figure 6.9: Variations of ID for fuels: (a) With loads (no-EGR), (b) With EGR rates at full load.

6.3.4 Effect of Preheated Biodiesel Blends on Engine Emission Levels

In this section, important engine emission elements for example carbon monoxide (CO), hydrocarbon (HC) and oxide of nitrogen (NO_x) had been studied for diesel (PBD0), all preheated POME biodiesel blends (PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100). It is important to note that the significant regulation of NO_x emissions can be achieved with controlled EGR rate. Similar experimental operating conditions was followed for emissions as it was made in performance and combustion characteristics, and comparison of the results. Finally, the optimized EGR rates can be recommend, which can be applied to diesel engine without sacrificing its efficiency, fuel economy for NO_x reduction.

6.3.4.1 Effect of Preheated Biodiesel Blends on CO Emission

The CO emission in exhaust gas is an indication of incomplete combustion due to deficiency of oxygen available in the combustion process. **Figure 6.10(a)** displays the carbon monoxide (CO) for test fuels with load without considering EGR. The CO emissions were higher at low load with gradually drop with increasing load (Yilmaz and Morton, 2011). This is due to high in-cylinder temperatures with increase load tending to make the fuel-air mixture leaner so that combustion is complete. It can also be seen that CO emission decreases with increasing preheated POME biodiesel content in the blend. The high oxygen content and lower carbon content (less Carbon/Hydrogen ratio) in biodiesel which acts as combustion promoter inside the cylinder and causes complete combustion. This sufficient amount of inherited oxygen in biodiesel which encourages in oxidation of carbon in the fuel throughout combustion plays important role for lesser CO emissions (Buyukkaya, 2010). At full load condition, the overall percentage decrease of CO emission for PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 was 5.32%, 15.52%, 27.19%, 36.53% and 46.65%, respectively as compared to diesel (PBD0). It is seen in **Figure 6.10(a)** that the engine consumes more amount fuel to generate appropriate engine power beyond 80% load. In this instance, only limited oxygen available for combustion in engine cylinder. This makes poor mixture in cylinder and may be the cause of increased CO, as it could not be oxidized to CO₂. The similar results were also reported in references (Hira and Das, 2016).

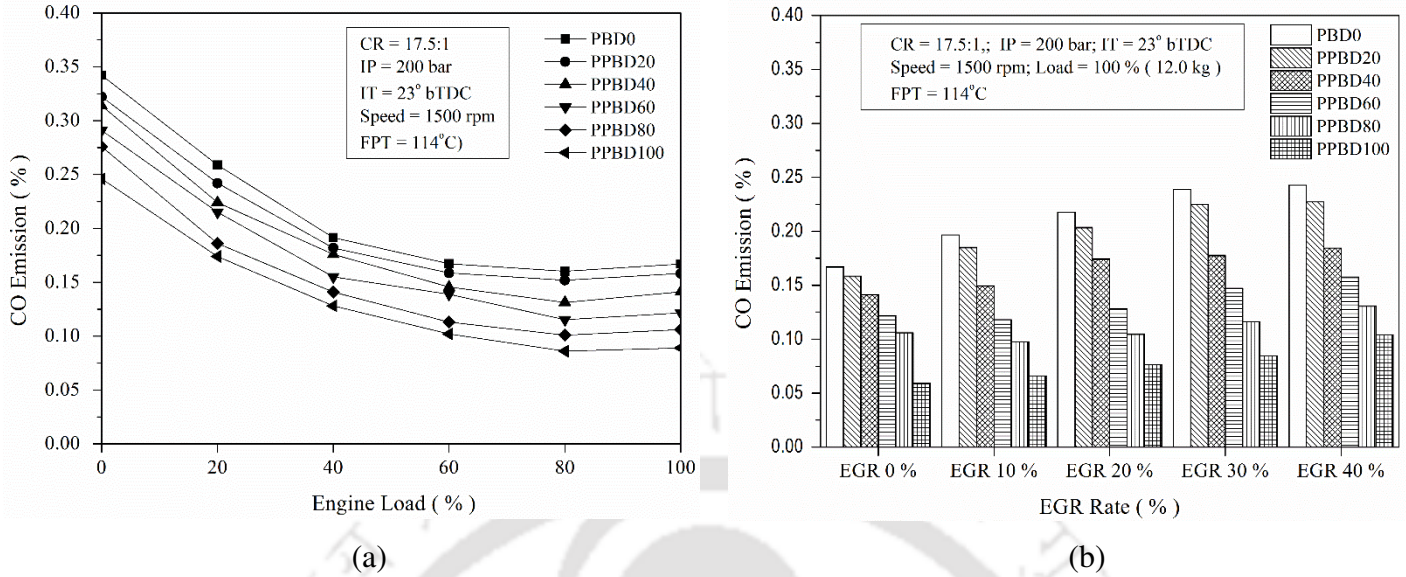


Figure 6.10: Variation of CO emission for fuels: (a) With load (no-EGR), (b) With EGR rates at full load.

Introduction of EGR prevents CO oxidation due to lower oxygen concentration and as a result, CO emission increases slightly with increasing EGR rates (Gupta, 2012; Rajasekar and Selvi, 2014), as shown in **Figure 6.10(b)**. This is because of a shortage of air (oxygen) content in the engine cylinder for burning. The exhaust gas dilutes with the intake air-fuel mixture and make a heterogeneous mixture. It becomes a challenging task to burn completely. CO emission of PPBD100 is less than other blends, since amount of oxygen availability increase with increasing POME biodiesel content in the blend. It is seen that, at EGR rate of 40% running condition, the amount of CO emission for all tested fuel was higher and worsening trend. Hence, it is recommended operate the engine with EGR rate less than 30% (maximum averaged increased by 22.4%).

6.3.4.2 Effect of Preheated Biodiesel Blends HC Emission

The variation of unburned hydrocarbon (HC) for different test fuels with load (no-EGR) is presented in **Figure 6.11(a)**. The amount of HC all fuels gradually drop with increasing load (Yilmaz and Morton, 2011). The HC for preheated neat POME fuel has lower values compared to other tested fuels over the load range. This may be a shorter ignition delay and early combustion attributed because of the higher Cetane number of biodiesel. The presence of more oxygen content in the biodiesel also contributing to have a better combustion, and this results in lowering HC

emissions. The results and trends are supported by available literatures (Samsukumar et al., 2015). At full engine load, the average reduction of HC emission for PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 was 2.32%, 6.97%, 16.28%, 25.58% and 34.88% with respect to neat diesel fuel (PBD0), respectively.

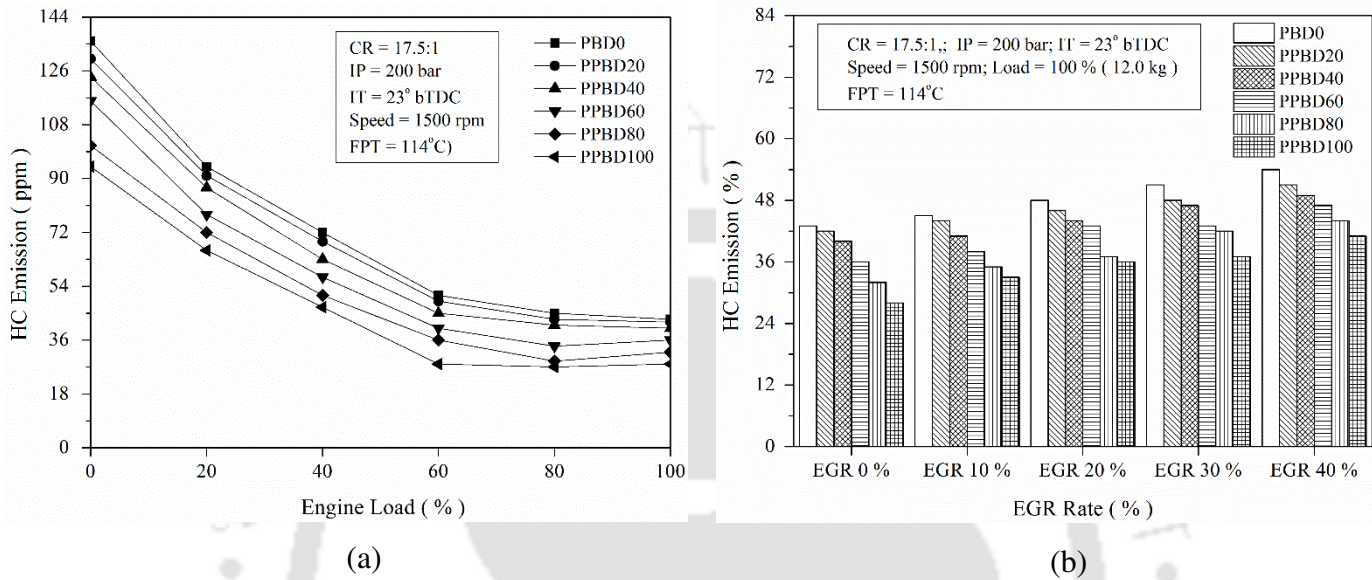


Figure 6.11: Variation of HC emission for fuels: (a) With load (no-EGR), (b) With EGR rates at full load.

The effect of various EGR percentages on HC emissions for all preheated POME biodiesel blends at full load is shown in **Figure 6.11(b)**. HC emission increases slightly with increasing EGR rates (Gupta, 2012; Saravanan, 2015). Similar reasoning can be made in this case as it was done for CO emissions. The air-fuel mixture is diluted and form heterogeneous mixture at higher EGR rate, which cannot combust fully. With respect to the case of without EGR (no-EGR) consideration, the average HC emission decreased for PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 with EGR rate of 30% was about 14.28%, 17.5%, 19.44%, 31.25 % and 32.14%, and increased by 2.08% as related to diesel fuel (PBD0) the engine operated with same EGR rate 30 % (refer **Figure 6.11-b**).

6.3.4.3 Effect of Preheated Biodiesel Blends on NOx Emission

The main drawback of biodiesel usages in diesel engine is the release of high amount NOx emission in exhaust gas, as a result of the attendance of more oxygen in combustion chamber. The

oxygenated nature of biodiesel obtained from vegetable oils is the source of the oxygen contents. **Figure 6.12(a)** illustrates the effect of preheated biodiesel/diesel blends on NO_x emission at various load condition without EGR. NO_x emission generally increased with engine load for all blends similar to diesel fuel due to increase in-cylinder gas temperature at higher engine loads (Yilmaz and Morton, 2011). It can be inferred that, at low/medium load conditions, NO_x emissions lowered for all blends with higher percentages of preheated POME biodiesel in the blend when compared to diesel. At full engine loading condition, the overall increased of NO_x emission with PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 was 12.04%, 21.91%, 25.92%, 31.77% and 36.29% higher compared to diesel (PBD0).

The major concern that limits the biodiesel usage in a diesel engine is the NO_x formation. Under these circumstances, the effectiveness of EGR is more promising as it can significantly decrease the NO_x emissions. The reduction in the engine in-cylinder gas temperature, is a key to reduce level of NO_x emission in the exhaust gas. The effect of various EGR percentages on NO_x emissions for all preheated POME biodiesel/diesel blends at full engine load is shown in **Figure 6.12(b)**. Increasing EGR rates decreases the flam temperature and oxygen concentration leading lower NO_x emissions (Gupta, 2012; Heywood, 1988; Saravanan, 2015). As seen earlier, NO_x emissions are generally higher for preheated POME biodiesel/diesel blends for any loads compared to diesel. The introduction of EGR, helps for reductions levels of NO_x emission for all EGR rates tested. At maximum EGR rate of 40 % considered in present investigation, the overall percentage of reduction NO_x emission for PBD0, PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 test fuels were noted to be, 91.14 %, 90.29 %, 86.55 %, 83.93 %, 76.27 % and 75.09 %, respectively, as compared to the case with engine running without EGR (EGR0%) at full load. However, at this EGR rate engine operation, the engine performance parameters (such as BTHE, BSFC) was seen worsen. Thus, the EGR should be limited up to 30 % EGR rate while using the biodiesel. At EGR rate of 30%, the NO_x emission for all test fuels of PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 dropped by 69.1%, 70%, 68.86%, 64.14%, 60.41% and 55.46%, respectively compared with without EGR (no-EGR).

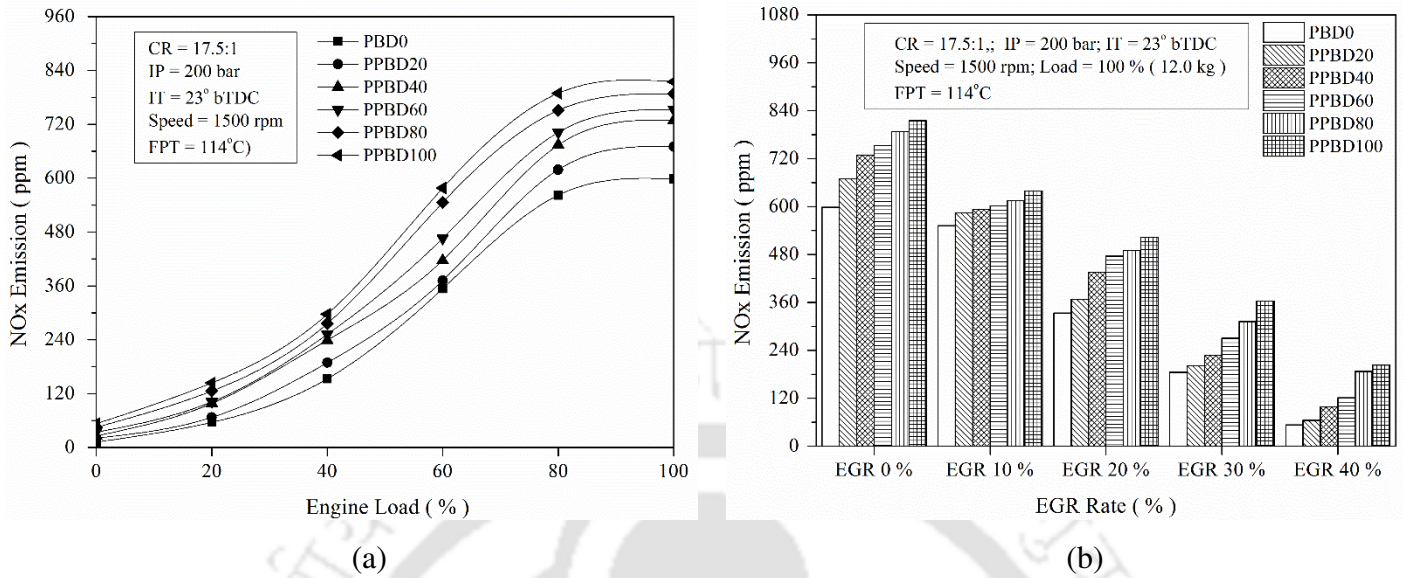


Figure 6.12: Variation of NOx emission for of fuels: (a) With load (no-EGR), (b) With EGR rates at full load.

6.3.5 Comparative Overall Engine Performance and Emission Analysis

The most preferred fuel sample among the considered biodiesel-diesel blends of fuel, preheated blended biodiesel-diesel fuel can be obtained by comparing their performance in terms of engine performance and emission parameter, and the related combustion behavior with respect to neat mineral diesel fuel. However, all these parameters are expressed/measured in different units and their values are also widely apart. Therefore, for comparative evaluation it is necessary to bring down these parameters to a common platform by non-dimensioning them. In the present study, as diesel is the base fuel for all of the selected unheated/preheated biodiesel-diesel blends of fuel samples, each of the performance and emission parameters for a particular sample is being non-dimensionlized by dividing the corresponding value for neat diesel. Mathematically, i^{th} parameter of a particular sample can be expressed as:

$$\beta_{i \text{ particular sample}} = \frac{i_{\text{particular}}}{i_{\text{diesel}}} \quad (6.1)$$

Where, β_i is known as the non-dimensionilizing parameter for the i^{th} parameter of a particular sample. For example, the non-dimensional form of BSFC for PPBD20 blends are expressed as;

$$(\beta_{PPBD20})_{BSFC} = BSFC_{PPBD20} / BSFC_{PBD0} ; (\beta_{PPBD20})_{BSFC} = BSFC_{PPBD20,EGR10\%} / BSFC_{PBD0,EGR0\%} ,$$

respectively. Accordingly, β values of all the considered performance (BTHE, EGT) and emission parameters (CO, HC and NOx) for each of the considered fuels are evaluated. The variations in the evaluated β values for preheated biodiesel blends with diesel samples (PBD0, PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100) at full load are presented graphically in **Figure 6.13** and **Figure 6.14** for engine performance and emission parameters, respectively. **Table 6.2** and **Table 6.3** are illustrated, the variations in the evaluated β values for preheated biodiesel blends. It is observed that the BSFC for PPBD20, PPBD40 PPBD60 and PPBD80 are relatively higher than neat diesel and found by 10.08%, 16.87%, 19.84% and 29.45%, respectively. Similarly, at EGR30% rate, the BTHE for PPBD20, PPBD40 PPBD60 and PPBD80 are lower than diesel fuel by 2.78%, 7.41%, 11.28% and 14.82%, respectively. The NOx emission for PPBD20, PPBD40 PPBD60 and PPBD80 are reduced than diesel fuel by 9.65%, 22.70%, 45.94% and 68.65%, respectively. Increasing the percentage of EGR rates, the CO and HC emission for different blends of fuel was higher than diesel fuel without EGR rate (EGR0%). The EGT is observed to be decreasing with increase in the percentage EGR rates of preheated biodiesel in blend ratio, which indicates poor fuel combustion.

6.4 Uncertainty Analysis

Any experiment is not free from error. Error may occur due to the instrument's inaccuracy, inappropriate calibration, human inadequacy etc. Therefore, uncertainty analysis is mandatory for any experimental work. The sequential perturbation technique (Kline and McClintock, 1953) is followed to acquire the uncertainties of independent and performance parameters (Appendix - D). Some of these are air flow rate (1.1%), liquid fuel flow rate (0.1%), engine load (0.1%), engine speed (1.3%), LHV of liquid fuel (1.0%) etc. Because of these facts, the calculated accuracy of the performance and combustion study for both PPBD20 and diesel run engine are found within $\pm 3.6\%$. However, the accuracy of the emissions is found within $\pm 2.3\%$.

Table 6.2: Performance and emission parameters of blended fuels at full load.

Parameter	Preheated Biodiesel blends					
	PBD0	PPBD20	PPBD40	PPBD60	PPBD80	PPBD100
β_{BSFC}	1	1.07772	1.110815	1.214597	1.298837	1.430601
β_{BTHE}	1	0.97836	0.939992	0.894852	0.839107	0.797513
β_{EGT}	1	1.08029	1.099019	1.12933	1.168426	1.205371
β_{CO}	1	0.946773	0.844809	0.728056	0.634654	0.533469
β_{HC}	1	0.976744	0.930232	0.837209	0.744186	0.651163
β_{NOx}	1	1.120401	1.219063	1.259197	1.317725	1.362876

Table 6.3: Performance and emission parameters of blended fuels at EGR30% with respect to no-EGR.

Parameter	Preheated biodiesel-diesel blends					
	EGR0%	PPBD20 @ EGR30%	PPBD40 @ EGR30%	PPBD60 @ EGR30%	PPBD80 @ EGR30%	PPBD80 @ EGR30%
β_{BSFC}	1	1.144206	1.178629	1.105345	1.116494	1.115777
β_{BTHE}	1	0.934623	0.926384	0.932434	0.954773	0.915905
β_{EGT}	1	0.774518	0.804504	0.808665	0.859053	0.805211
β_{CO}	1	1.421995	1.258257	1.210773	1.097453	1.432657
β_{HC}	1	1.142857	1.175	1.194444	1.3125	1.321429
β_{NOx}	1	0.3	0.311385	0.358566	0.395939	0.445399

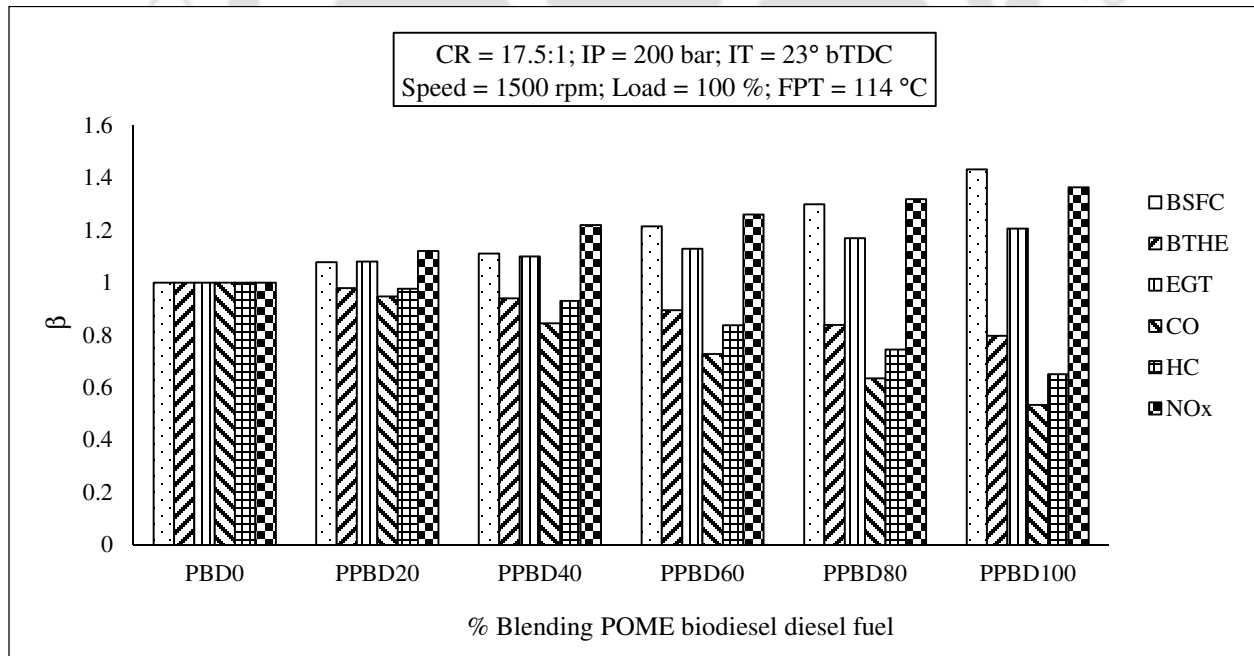


Figure 6.13: Comparative engine performance and emission parameters analysis for different test fuels.

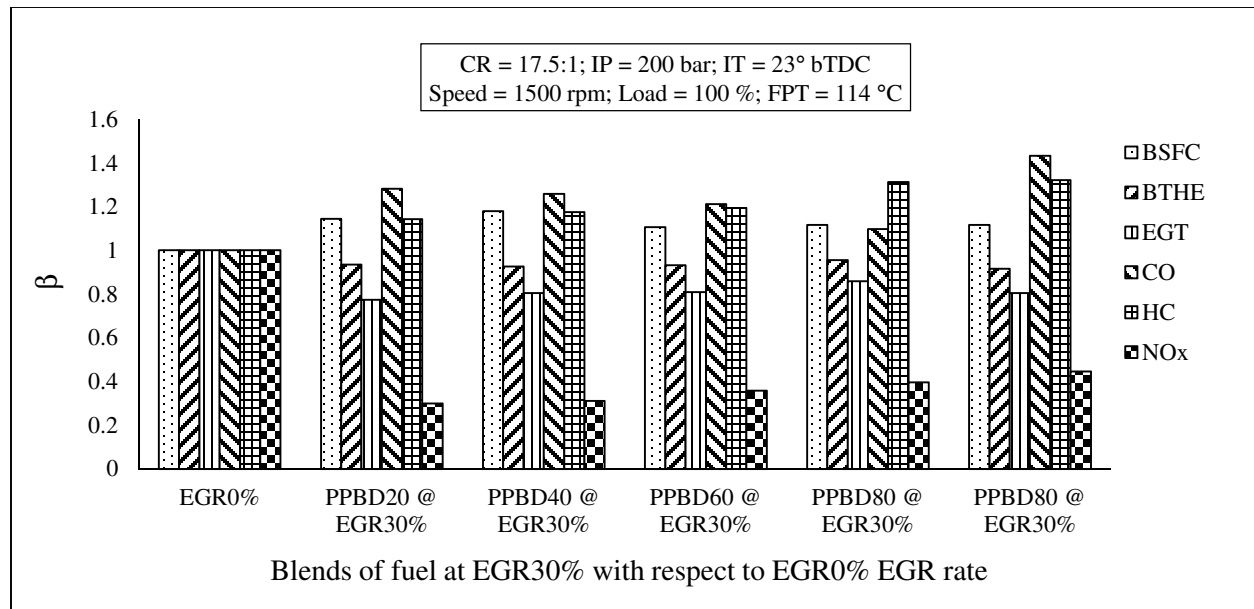


Figure 6.14: Comparative engine performance and emission parameters analysis for different test fuels with EGR30% with respect to no-EGR rate (EGR0%).

6.5 Summary

The effect of using preheated POME biodiesel/diesel blends up to 100% in proportion on the performance, combustion and emission characteristics of preheated POME biodiesel/diesel blends were investigated and compared with baseline diesel fuel in a DI diesel engine. The effect of increasing EGR rates up to 40% was also investigated. EGR was intended to reduce the high NOx emissions that are prevalent during full engine load. The following points were drawn from the experimental study:

- Combined effect of heating and blending significantly reduces the kinematic viscosity of biodiesel, while the density drops linearly. The blends of fuel (PPBD20–PPBD60) at 114°C, property values of blended biodiesel meet standard specification. The average reduction of viscosity for blends was found to be 53.2–61.2% as compared to initial values 8.71 mm²/s at 40 °C, whereas the density drops by 8.8–10% with respect to reference values at 27 °C (838 kg/m³).
- It is possible to operate a DI diesel engine with up to 60% preheated POME biodiesel/diesel blends with and without EGR. The operation required no modifications to be made to the

engine and no visible damage to the engine parts was observed. However, to recommend it as a regular fuel, long-term durability tests are required.

- BTHE decreases for all preheated POME biodiesel/diesel blends at increasing EGR rates with the loss being more for PPBD100 blends. BSFC also increased for preheated POME biodiesel/diesel blends with increase in EGR rates.
- Ignition delay gets longer for all preheated POME biodiesel/diesel blends at all EGR rates tested, while the peak cylinder pressure slightly decreased.
- The combination of medium EGR rates (EGR30%) and blends up to 60% Preheated POME biodiesel content can simultaneously reduce NO_x emissions with a little drop in engine performance.
- CO and HC emissions increased with increasing EGR rates for all preheated POME biodiesel/diesel blends.
- With respect to utilization of test fuels in the diesel engine, the biodiesel blended with diesel (PPBD20–PPBD60) with EGR rate of 30% offers best alternative for replacement of diesel. The average percentage increase of BSFC by 10.5–17.8%, BTHE decreased by 6.5–7.4%, EGT decreased by 19.1–22.6%, volumetric efficiency decreased marginally by 1.6–1.9%. PCP decreased by 6.9–12.7%, ID increased by 11.7–20%. CO increased by 21.1–42.2%, HC increased by 14.3–17.5%, NO_x decreased by 64.1–70% at full load operated conditions.

In general, preheated palm oil methyl ester biodiesel/diesel blends (up to PPBD60) are an excellent alternative fuels that can find potential use in diesel engines under EGR rate of 30% is an effective solution for a high level of NO_x reduction with a minor affects engine performance, and has the capability to offer both energy security and environmental safety.

CHAPTER-7

Results and Discussion: Preheated Biodiesel Blends with Intake Air Preheating Mode Run Engine

Overview

Preheated biodiesel-diesel blends show some mixed results on NO_x, CO and HC emissions in the in previous experimental works, and the reason for such contradictory outcomes is that preheated biodiesel–diesel mixing ratios and perhaps ambient air temperature would tremendously change the results. Thus, it is greatly important to know performance and emissions of CI engines as functions of intake air temperature and preheated biodiesel blend ratio and understand how these two parameters relate to each other in terms of engine performance and emissions. So for achieving better efficiency with emission controls of NO_x, and CO and HC, reduction the intake air temperature plays a very important role. Hence, an intake air preheater is used in diesel engine applications for quick, reliable, and environmentally friendly operation. Intake air preheating is a technique to increase the temperature of inlet air by utilizing the thermal energy of exhaust gas with a prime objective of increasing the thermal efficiency of the engine. In this regards, intake air preheater was designed for heating the air using waste heat of exhaust gases of the engine. An exhaust pipe line and flow control valve was attached to the experimental setup with prepared heat exchanger to preheat the intake air. Then, the performance, combustion and emissions characteristics of a diesel engine fueled with preheated POME biodiesel–diesel blends (PBD0, PPBD20, PPBD40 and PPBD60), subjected to intake air preheating at different temperatures (54–138 °C) with 12 °C increments have been investigated. Tests were carried out at full load conditions in a VCR DI CI engine. Preheating intake air saves the fuel that would otherwise require heating the combustion air. Intake air preheating up to elevated temperature subsequently reduces CO and HC emissions. Preheating not only ensures performance enhancement and emission control but also improves the cold start condition.

7.1 Preface

Latest studies show significant differences in fuel consumption and exhaust emissions depending on the temperature of the intake air. It is well accepted that the combustion and emissions characteristics highly rely on the oxygen availability and fuel properties. The excess of oxygen resulted in very lean mixture that lead to the instable combustion and misfire. In fact, lean mixture tends to promote a longer ignition delay and a slower burning rate resulted in longer combustion duration. This condition creates longer time for heat transfer from combustion to the end gas that lead to the knocking phenomenon. However, the engine operates with deficiency of oxygen resulted in rich mixture that leads to the higher unburned gaseous fuel due to shorter ignition delay and faster burning rate. Therefore, it is vital to make sure the charged air entered into the combustion chamber was sufficient of oxygen in order to promote complete combustion. The previous researches also showed that the air intake temperature has a significant role to increase the combustion efficiency, stability and reduced exhaust emissions (Abdullah et al., 2015; Mamat et al., 2009; Mamat et al., 2013). Few previous studies on diesel engine showed that, fuel economy can be improved through several strategies. Intake air preheating is one of the possible methods to promote higher combustion efficiency and improved combustion stability. Preheating intake air is important to ensure higher amount of fuel involves in the combustion process. Besides that, air intake temperature will also control the exhaust emissions and combustion process (Abdullah et al., 2015). Limited studies have been carried out in a diesel engine related to intake air preheating for performance and emission parameter analysis (Kumar et al., 2013; Pan et al., 2015; Paykani et al., 2012). They proved that, intake air preheating causes lower ignition delay and improves the combustion rate, which is responsible for higher engine brake thermal efficiency and better fuel economy. Uniform or better combustion is occurred due to preheating of intake air, which also causes lower unburned hydrocarbon (HC) and carbon monoxide (CO) emission but NO_x formation slightly increases with increase in temperature. However, there are no extensive studies on a combined effect of fuel and intake air preheating on a diesel engine, it is hard to make a conclusion regarding diesel engine performance and combustion analysis about a combined result in past literature.

The biodiesel and intake air has been preheated with the help of waste heat exhaust gases from the engine. The biodiesel was preheated to evaluated temperature before blend with diesel fuel in a different percentage fractions of biodiesel to decrease kinematic viscosity and provide a better

fuel atomization and combustion efficiency which results emission control. Preheated blended fuel ensures proper atomization and combustion of the air-fuel mixture while preheating the inlet air temperature causes lower ignition delay leading to increased combustion efficiency. In a diesel engine only air is supplied into the cylinder during the suction stroke and during the compression stroke the intake air getting compressed, and at the end of compression stroke, fuel is injected so that combustion takes place inside the cylinder. As the temperature of air at the inlet to engine is low there is no complete combustion of the fuels. To ensure the complete combustion of the fuel the temperature of the air at the inlet should be improved and this can be done by means of an intake air preheater at the inlet of the air flow to the engine.

The aim of this paper is to investigate the combined effect of fuel and intake air preheating on diesel engine characteristics with palm oil methyl ester (POME) biodiesel–diesel blends of PPBD20, PPBD40 and PPBD60. One of the targets of this work is to have uniform or better combustion and lower the diesel emissions (CO and HC) with the newly designed system without deteriorating engine performance. Fuel was evaluated within the engine twist with intake air temperature at 33 °C, 41°C, 49 °C and 61°C respectively at 90% load conditions. The results of diesel engine characteristics (performance, combustion and emission characteristics) with the newly designed system were compared with that diesel (PBD0), with and without preheat intake air. The POME biodiesel was preheated to evaluated temperature of 114 °C before blend with diesel fuel.

7.2 Materials and methodology

7.2.1 Experimental Facilities

Tests were performed considering a modified experimental setup demonstrated in **Figure 3.13**. **Table 3.1** shows the specification of the engine. For preheating intake air, we have developed a novel shell and tube counter flow type heat exchanger which is located at the air inlet, and it preheat the combustion air to the required temperature for ignition of fuel. In the modified experimental set up, along with intake preheater, a helical coiled type heat exchanger is also integrated for heating the biodiesel (refer **Figure 3.13**). The detailed technical specifications each preheater are given in **Table E.1** (Appendix-E). Both the heat exchangers are indigenously designed with an assumption to meet the requirements with specific intended purpose. While designing a fuel and

intake air preheater, the laws, which govern this process, should be well understood and thus should be used in this design, construction, testing and operation of the equipment. The intake air and biodiesel were preheated by engine exhaust gas temperature, when it is passed through a newly designed preheating systems. An exhaust gas analyzer (AVL-444 DI-GAS, India) was used for emissions of HC, CO, CO₂ and NO_x measurement. Finally, the experiments with all the fuel samples in the diesel engine are conducted to study combined impact of air and fuel preheating on the engine performance, combustion and emission characteristics under stable conditions of operation at constant speed (1500 rpm) with default CR 17.5, 200 bar and 23° bTDC of fuel IP and IT, and are shown in test matrix **Table 3.3**. For each test (PBD0, PPBD20, PPBD40, and PPBD60), important observations such as, mass flow rate of fuel, mass flow rate of air, temperatures, temperatures at various locations and emission measurements (CO, HC, CO₂, and NO_x) were made. The best operating condition for each blends of was calculated/measured (BSFC, BTHE, cylinder pressure, heat release rate, PCP and ID, and finally the optimum condition (intake air preheating temperature) for the operation of CI engine was determined.

7.2.2 Methodology

While executing the tests, the supply of exhaust gas to both preheating device (fuel and intake air preheaters) is regulated in combination to a manual control gate valves (gv2 and gv3) so that preheat temperatures in both devices are properly controlled (refer **Figure 3.13**). The operation of fuel preheater (Hx1) is controlled manually by two valves “gv1 & gv2” and both are regulated in combination to control exhaust gas flow rate so that the preheat temperature is properly set. When the engine starts, the valve (gv2) is fully open while valve (gv1) is closed partially to restrict exhaust flow to the atmosphere. It allows major quantities of the engine exhaust gases to come in contact with the surface of helical coil. Once the required temperature is displayed in a digital temperature gauge instrument, the opening of valves (gv2) is manually decreased. Subsequently, increasing the opening angle of valve (gv1) allows more exhaust gas flow to atmosphere such that the exhaust gas supply get restricted in order to maintain constant temperature of fuel. When the temperature drops below the necessary value, the opening of the valve for “gv2” is increased and for “gv1”, it is decreased to execute other heating process. This POME (biodiesel) with elevated temperature then flows through secondary fuel pipe (made of the same material as that of fuel injection line) and then connected to the fuel blend metered glass burette. The blending of different percentage fractions of biodiesel with diesel was done manually based on the percentage in volume

requirements of preheated POME using a biodiesel and diesel fuel flow control two-way valves (v1 and v2) as shown in **Figure 3.13**. Once proper blend ratio achieved, both valves, (v1 and v2) were closed and the engine gets fuel from the metered glass burette through valve “v3”.

Similarly, an intake air preheater (shell and tube type) device, (Hx2) has been arranged in the modified experimental setup as shown in **Figure 3.13**. It is not fixed directly in exhaust gas line rather adequate distance is maintained closure to intake manifold. The heating device is kept, as close as possible to the inlet openings of the cylinders. This arrangement assists the intake air combustion is taken from the atmosphere through an air filter (refer **Figure 3.13**). The filtered air is passed through the copper tubes in the intake line surrounded with the exhaust gases in the shell. This arrangement guarantees fast heating of the intake air as well as increased robustness and simplified mounting. The operation of preheater is similar to that of a fuel-preheater, for which the flow exhaust gas temperature to intake air preheater is controlled manually by using two valves (gv1 and gv3) as shown in **Figure 3.13**. While executing the tests, both are regulated in combination to control exhaust gas flow rate so that preheat temperatures in both devices are properly controlled. The operating intake air preheating temperatures (from atmospheric temperature of 33 °C to 41°C, 49 °C and 61°C) and fuel preheating temperature (114 °C), were decided based on optimal engine operations at constant 90% (10.8 kg) of engine load. The experiments were carried out with a standard operating conditions (CR 17.5, fuel IP of 200 bar, fuel IT of 23° bTDC) at 90% engine load without EGR. It may be emphasized here that a diesel engine has better fuel combustion and the effective conversion of chemical energy into useful work. In most of diesel engine, in the range of 80–90% of full load, the engine has the maximum thermal efficiency, but increasing to 100% of full load, the thermal brake efficiency decreases because the engine reaches the smoke limit (Najafi et al., 2018). All diesel engines attain best performance including specific fuel consumption and thermal efficiency at rated power. For a constant speed for four stroke diesel engines operated in the range of 80-90% load, the general tendency is that specific fuel consumption (fuel consumption per unit brake power) decreases efficiently. Beyond this load (> 90%), it gradually increases, thereby, the effectiveness to convert the chemical energy contents of fuel into useful work drops leading to decrease of mechanical efficiency and increase of radiation losses (un accountable losses) and frictional losses. That is why, in this experimental study, the full load (100%) engine loading operating condition was not considered so as to avoid the exhaustion and failure of the engine as a precautionary measure.

7.3 Results and Discussion

The results of the experiments are presented in the form of performance analysis, combustion analysis, and emission analysis in the following sub-sections.

7.3.1 Effect of Fuel and Intake Air Preheating on Engine Performance Parameters

Important performance parameters analyzed in the experiments are BSFC, BTHE and EGT. Each parameter is described separately in following sections.

7.3.1.1 Brake Specific Fuel Consumption (BSFC)

Figure 7.1(a) shows that, the BSFCs for different test fuels decreases with increasing intake air preheating. It is mainly due to shorter ignition delay and improved combustion efficiency which is in line with reported data (Yilmaz, 2012b). It is being found that specific fuel consumption drastically reduced by pre heating the inlet air as compared to no preheating attachment. This indicates reduced mass of fuel consumption for developing a kW of power. For 90% load, BSFC is attained minimum for all the test fuels (PBD0, PPBD20, PPBD40, and PPBD60) at 61°C intake air temperature i.e. 0.24884, 0.25652, 0.277048 and 0.28631 kg/kW.hr, respectively. The BSFC is seen maximum at 33 °C (i.e. without intake air preheating) with the value being 0.266345, 0.27832, 0.28934 and 0.30612 kg/kW.hr, respectively. The average percentage of drop in BSFC for all the test fuels at intake air preheating (61 °C) is seen to be the order of 5.54–7.83%, when compared to the conventional engine without preheating intake air at 33 °C.

7.3.1.2 Brake Thermal Efficiency (BTHE)

The trends of BTHE for various test fuels with intake air preheating temperatures are opposite to that of the graphs of BSFC. Preheating the intake air increases thermal efficiency of the engine, which causes better vaporization and mixing of air fuel mixture. **Figure 7.1(b)** displays the BTHE with increasing intake air preheating temperature. It is seen that, BTHE for PBD0, PPBD20, PPBD40, and PPBD60 inducted fuels is increased with preheating intake air, which is in line with reported data (Kumar et al., 2013). For 90% load, BTHE is attained maximum for all the test fuels (PBD0, PPBD20, PPBD40, and PPBD60) at 61°C intake air temperature i.e. 31.93%, 30.87%, 29.44% and 28.76%, respectively. The BTHE is seen minimum at 33°C (i.e. without intake air preheating) with the value being 29.12%, 28.68%, 27.81% and 26.95%, respectively. The average

percentage of increase in BTHE for all the test fuels at intake air preheating (61°C) is seen to be in the range of 5.87-9.63%, when compared to the conventional engine without preheating intake air at 33 °C.

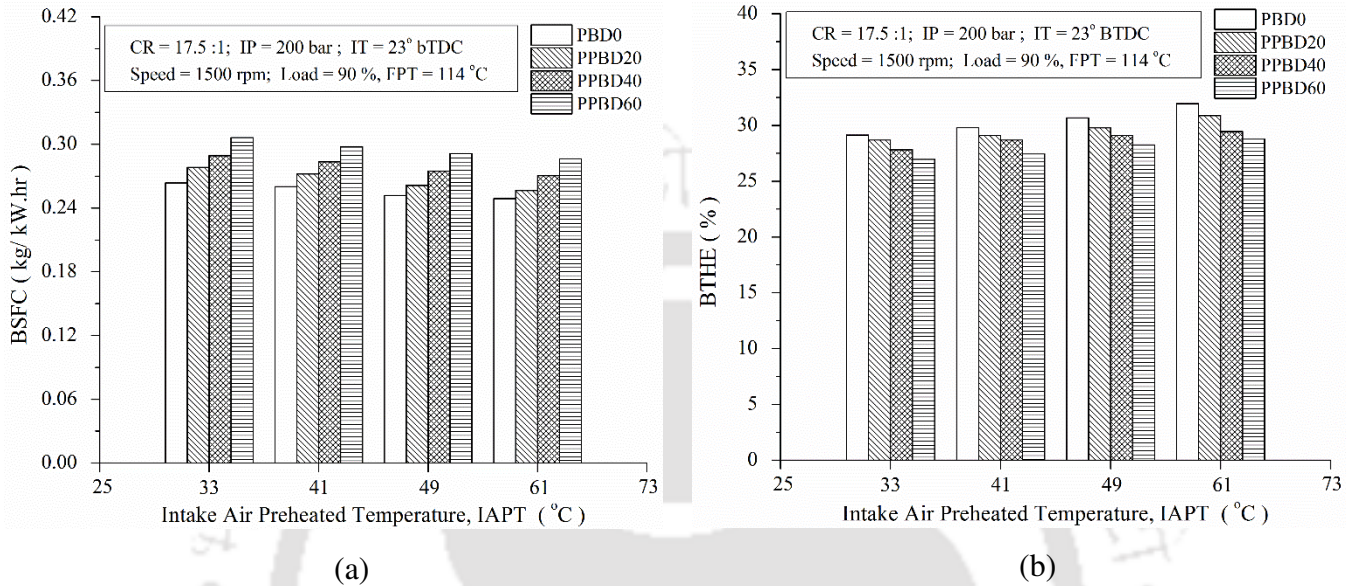


Figure 7.1: Variation engine parameters with intake air preheated temperature: (a) BSFC, (b) BTHE.

7.3.2 Effect of Fuel and Intake Air Preheating on Engine Combustion Characteristics

The primary purpose of preheating intake air is to improve combustion efficiency with shorter ignition delay. Combustion characteristics of diesel engine running with diesel fuel and three preheated biodiesel blends of fuels are explained in this section.

7.3.2.1 Cylinder Pressure

The variations of cylinder gas pressure with crank angle for all tested fuels at 90% of engine load, are shown **Figure 7.2(a)**. Cylinder pressure characterizes the ability of the fuel to burn with proper mixing with air. The pressure waves in the cylinder during combustion shows sudden rise in pressure indicating the effects engine noise with biodiesel operations. The results show that the peak cylinder pressure of the engine running with biodiesel is slightly higher than the engine running with diesel. The main cause for higher peak in-cylinder pressure in the CI engine running with biodiesel is because of the advanced combustion process initiated by easy flow-ability of bio-

diesel due to the physical properties of biodiesel. In addition, the presence of oxygen molecule in biodiesel, the hydrocarbons achieve complete combustion resulting in higher in-cylinder pressure (Gumus, 2010). The cylinder pressure of preheated biodiesel blends of fuel (PPBD20, PPBD40 and PPBD60) follows a similar trend to that of diesel (PBD0). It is evident that higher percentage fractions of biodiesel exhibited higher in-cylinder pressure as compared to diesel baseline (PBD0). The peak in-cylinder pressure of PPBD60 fuel is higher than that of baseline diesel. This is due to higher cetane number that tends to lower ignition delay period of biodiesel resulting improved combustion. It can be seen from **Figure 7.2(a)** that the peak cylinder pressure occurred in the range of 5-8° CA aTDC for all test fuels. It resembles the reported data in the literature (Rao et al., 2008). The effect of fuel preheating (POME at 114 °C) and blend with diesel fuel has offered maximum peak cylinder pressure at 90% load conditions. This is due to a better fuel atomization and improved combustion process.

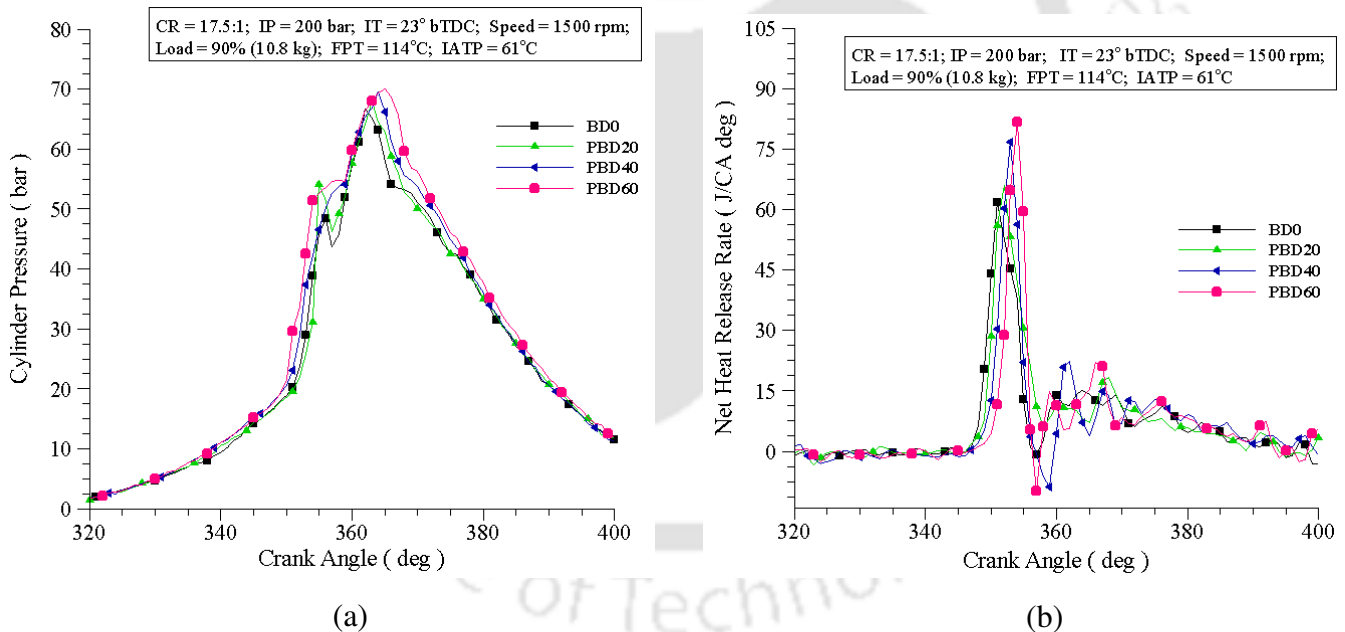


Figure 7.2: Comparisons of combustion parameters with crank angle for test: (a) Cylinder pressure, (b) Net heat release rate.

7.3.2.2 Heat Release Rate

Figure 7.2(b) shows the heat release rate (HRR) with crank angle for diesel engine running with different test fuels at 90% load conditions. It can be seen that the engine running with biodiesel

blends of fuel have a higher peak in the heat release rate diagram with respect to the baseline diesel fuel. This phenomenon can be explained on the basis of the presence of the oxygen molecule in biodiesel fuel that results in the air-mixed fuel in the cylinder to burn completely and increase the heat release rate. It can be observed that the NHRR curve shifts towards right with increase in volume fraction of biodiesel in the test fuel, which indicates shorter ignition delay. In the same figure, the heat release rate initially goes in negative due to endothermic reaction of the charge and it becomes positive when combustion starts. It is obvious that, engine running with different blends of fuel (preheated and blend of POME) caused to increases in peak heat release rate at 90% load condition. This is due to the reduction in viscosity and density with heating and blending. When the viscosity is lowered, the fuel gets atomized finer and enhance the combustion process, especially the diffusion combustion by capitalizing the oxygen presence in POME.

7.3.2.3 *Peak Cylinder Pressure*

Figure 7.3(a) shows the peak cylinder pressure (PCP) for different test fuels with intake air preheating. It showed that the PCP all preheated biodiesel blends was slightly higher than diesel. The result showed that increased percentage of biodiesel in blend ratio caused to increase the PCP of the engine (Kumar and Dixit, 2014). These may be due to a higher cetane number of biodiesel that shorten ignition delay and advance injection timing than diesel fuel. It results early combustion of biodiesel blends fuels and its PCP attain a maximum. The more oxygen content in POME help to burn more carbon fuel in combustion chamber that leads to improve combustion and increased PCP (Kumar and Dixit, 2014). In addition, preheating intake air causes higher PCP of all tested fuels. It is observed in **Figure 7.3(a)** that the PCP for all tested fuels increases at higher intake air preheating temperature. Thus, the air molecules in combustion chamber enhances molecular collisions at elevated temperature resulting improved reaction rate, short reaction time, early combustion with higher PCP all tested fuels. For 90% load, the maximum PCP for PBD0, PPBD20, PPBD40 and PPBD60 test fuels is attained at a preheating of 61°C as, 66.7, 67.7, 69.4 and 70.7 bar, respectively, whereas the PCP was seen minimum at 33 °C and the value being 59.2, 60.16, 61.3 and 62.1 bar. With intake air preheating, the average percentage increased in PCP for 61°C is about 15%, intake air at ambient conditions (33 °C).

7.3.2.4 Ignition Delay

The variation of ignition delay (ID) for various test fuels with intake air preheating engine is illustrated in **Figure 7.3(b)**. The result revealed that the ignition delay period for all tested fuel slightly decrease with increasing of intake air preheating temperature which is of similar trend as reported in the literature (Lapuerta et al., 2008). The overall combustion efficiency improved due to better mixing characteristics of high-temperature air with inducted vaporized fuel. It is seen in **Figure 7.3(b)** that, preheated biodiesel blended fuels have lower delay period than diesel fuel. The reason it may be due to oxygenated nature of biodiesel that helps to start early combustion and influence the delay period. Biodiesel has lower compressibility and higher viscosity compared to diesel fuel, which leads an early fuel injection, which is usually happening as soon as the injector needle lift up from its seat and affects adversely the combustion characteristics. The higher cetane number with relatively higher density and kinematic viscosity of biodiesel oil causes a smaller delay period than fossil diesel fuel. The result showed that, increasing biodiesel blend ratio to PPBD60 lowers the delay period significantly at varying intake air temperature. The reason may be due to an increase percentage of biodiesel in blend ratio that causes increased content of oxygen in the blended fuel, resulting in improved reaction rate and combustion, and shorten delay period. However, the ignition delay of neat diesel fuel (PBD0) was longer as compared to all preheated biodiesel blended fuels. This is because of a simple and slow pre-flame chemical reaction takes place at low temperatures. For 90% load, ignition delay for PBD0, PPBD20, PPBD40 and PPBD60 test fuels at a preheating of 61°C were found minimum with 14, 13, 11 and 10 °CA, whereas the ignition delay was seen maximum at 33°C and the values as 18, 17, 15 and 14 °CA, respectively. The overall percentage reduction of ignition delay for 61°C is in the range of 22-28%, when compared to without preheating intake air at 33 °C.

7.3.3 Effect of Fuel and Intake Air Preheating on the Engine Exhaust Emissions

In this section, emission analysis of exhaust gas has been performed. It describes the variation in emission of different species (CO, HC, CO₂ and NO_x) in exhaust gas. The experiments were carried out at intake air preheating temperatures of (41–61°C) at 90% loads without EGR and compared with ambient intake air temperature (33 °C) engine operating in conventional CI combustion mode.

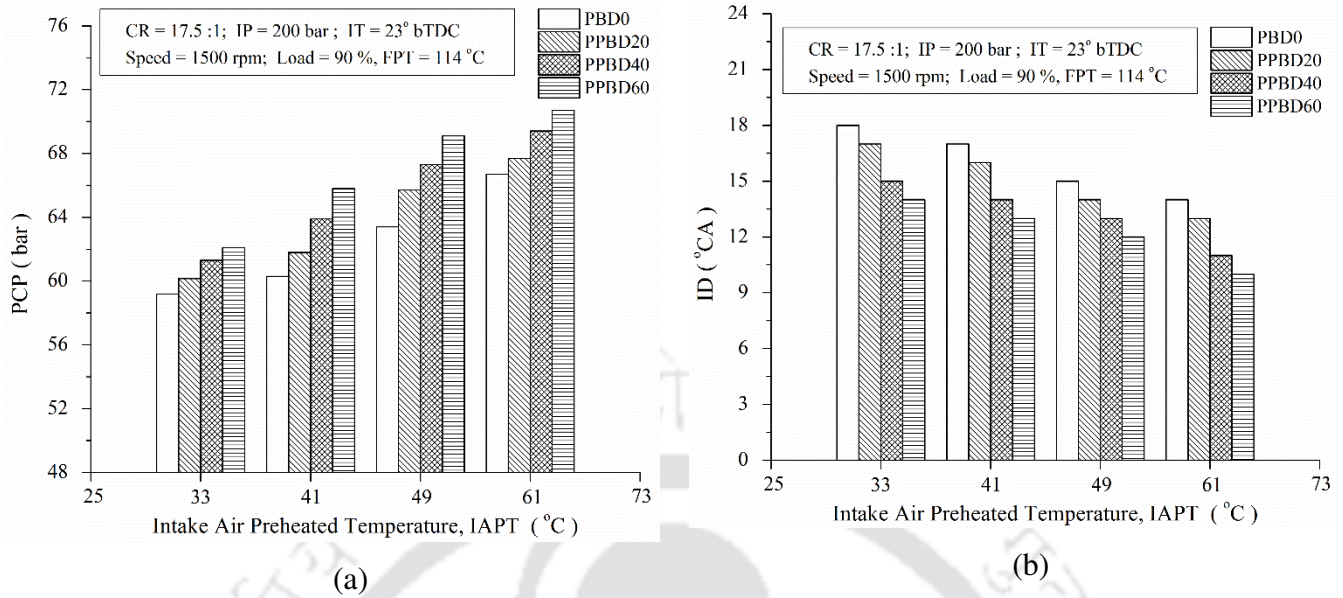


Figure 7.3: Variation of combustion parameters with intake air temperature: (a) PCP; (b) ID.

7.3.3.1 Carbon Monoxide (CO) Emission

The behavior of CO emission is consistent with the quality of the combustion process and poor oxidation of mixture. The variations of CO emissions at different intake air preheating temperatures for the inducted fuels at constant engine load 90%, are given **Figure 7.4(a)**. The intake air preheating temperature is varied from 33 °C to 61 °C as earlier. The results show the decrement in CO for higher intake air temperature which resembles similar nature as cited in reference (Yilmaz, 2012b). Intake air preheat helps biodiesel blends to reduce CO emissions by considerable amount as compared to ambient air. The cylinder temperature increases due to a betterment in combustion process at higher intake air temperatures. It leads to more amount carbon dioxide by successively dropping the level of carbon monoxide emissions. It was observed in **Figure 7.4(a)** that, at the maximum intake air temperature 61°C, CO emissions for PBD0, PPBD20, PPBD40 and PPBD60 are seen minimum as, 0.08643%, 0.06484%, 0.05096% and 0.04398%, respectively, and at 33 °C, it showed maximum i.e.. 0.15234%, 0.14023%, 0.13111% and 0.11567%, respectively. The higher inlet air temperature provided better air-fuel mixing and a higher rate of oxidation of CO. The average percentage reductions in CO emissions for various test fuels (PPBD0–PPBD60) at intake air preheating (61°C) is seen to be in the range of 43.26–61.98%, when compared to the conventional engine without preheating intake air at 33 °C.

7.3.3.2 Carbon Dioxide (CO₂) Emission

Referring to **Figure 7.4(b)**, the variation of CO₂ emission shows marginal increase in the amount of CO₂ with intake air preheating. The average percentage increase of CO₂ emission for test fuels PBD0, PPBD20, PPBD40 and PPBD60 at 61°C, are 24.4, 24, 28.3 and 32 %, respectively, as compared to 33 °C.

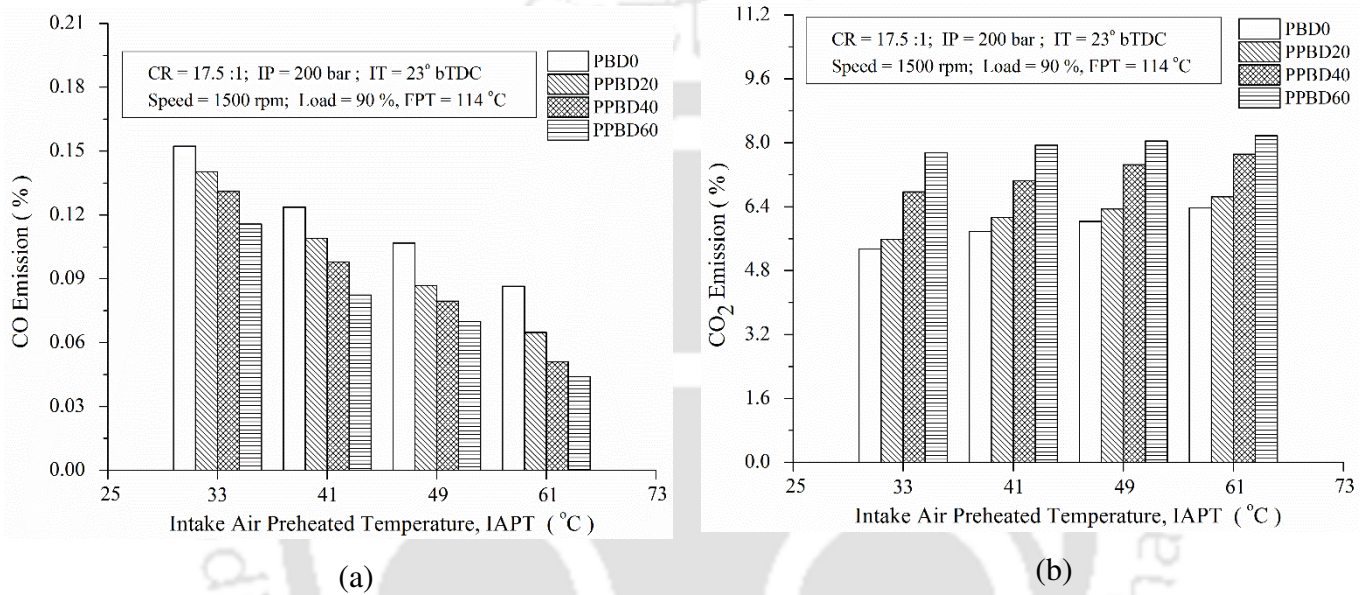


Figure 7.4: Variation engine exhaust emissions with intake air preheated temperatures: (a) CO, (b) CO₂.

7.3.3.3 Unburnt Hydrocarbon (HC) Emission

The intake air temperature had an important role in the HC emissions. The variation in HC emissions for different test fuels with intake air preheating temperatures for 90% constant engine load is shown in **Figure 7.5(a)**. The graph shows decreasing trends of HC emissions for higher intake air temperature for all test fuels as reported earlier (Yilmaz, 2012b). The increased combustion temperature results in lowering the levels of HC emission as shown in **Figure 7.5(a)**. The maximum of HC emission for PBD0, PPBD20, PPBD40 and PPBD60 fuels are found as, 43, 42, 40 and 36 ppm, and the minimum values as, 26, 19, 13 and 9 ppm, respectively at 33°C and 61°C intake air preheating temperature for 90% loading condition. The percentage reduction of HC emissions for (PBD0–PPBD60) fuel is seen to be in the range of 39.53–75%, respectively at 61°C intake air operation compared to stock preheating intake air (33°C).

7.3.3.4 Oxides of Nitrogen (NOx) Emission

It has been seen in **Figure 7.5(b)** that, the NOx emission of preheated biodiesel blend fuels was found higher at varying intake air preheating temperature compared to diesel fuel. The increase in the intake air preheating temperature improves the combustion characteristics which causes an immediate increase of engine gas temperature that leads increased NOx emission. After preheating the intake air, the NOx is being slightly increased due to high combustion temperature. The graph (**Figure 7.5-b**) shows that NOx for PBD0, PPBD20, PPBD40 and PPBD60 test fuels slightly increase for higher intake air preheating temperatures which compares well with reported data (Kumar et al., 2013; Yilmaz, 2012b). The minimum NOx emission for various experimental fuels (PBD0, PPBD20, PPBD40 and PPBD60) showed NOx levels as, 598, 670, 729 and 753 ppm at 33°C with maximum values as 774, 815, 868 and 942 ppm at 61°C intake air preheating temperature for 90% engine loading condition. The average percentage increased NOx emissions of (PBD0–PPBD60) is seen to be in ranges of 19.1–29.43%, compared to the operation of the engine without intake air preheating (33 °C).

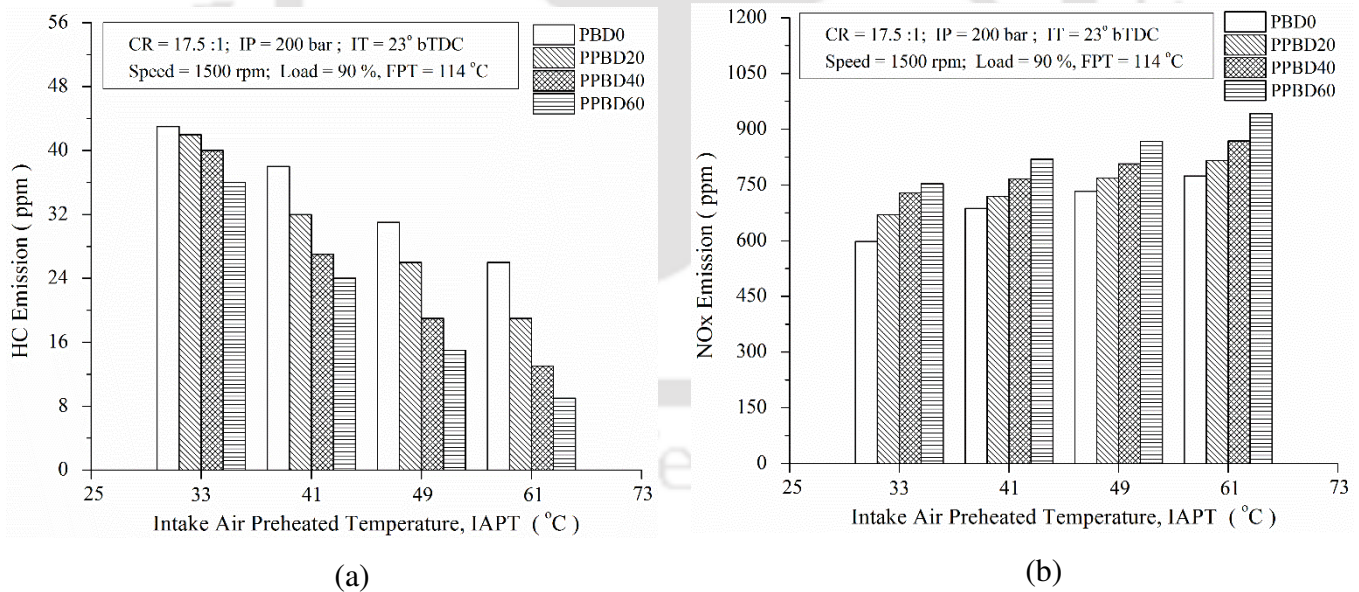


Figure 7.5: Variation engine exhaust emissions with intake air preheated temperatures: (a) HC, (b) NOx.

7.4 Summary

The aim of this study is to investigate the influences of fuel and intake air preheating in diesel engine performance, combustion and emission parameters running with blends of biodiesel (POME). From this investigation, it is observed that there is positive effect of inlet air preheating with preheated POME biodiesel/diesel blends in diesel engine on engine performance, combustion and emissions parameters. Preheating intake air is a major factor to affect the fuel combustion process and act as a concept for conservation of fuel, since it provides additional energy to preheated biodiesel/diesel blends. The following points can be summarized in this experimental study:

- From the test it is clear that the fuel consumption reduces and brake thermal efficiency increases with increase in intake air temperature.
- Preheating intake air reduces the ignition delay their by improving combustion process (complete combustion) of the engine, also shorter ignition delay leads to higher in-cylinder pressure.
- Preheating of intake air subsequently reduces the exhaust emissions. The CO and HC emission content will decrease with increase in intake air temperature, whereas NOx content in the exhaust gas slightly increases with increase in intake air temperature.

With reference to above consequences, the combined effects of fuel and intake air preheating play an important role for improvement of engine performance, combustion and emission parameters of diesel engine running with different levels biodiesel/diesel blends. Comparatively more content of exhaust emissions of CO and HC can be controlled when only inlet air preheating system attached with the engine than others condition of attachment. Overall, the test results indicated that the advantages gained are more with the combined effects of intake air preheating and preheating biodiesel/diesel blends, and it may be proved a valuable option palm oil methyl ester (POME) for substitution of diesel fuel in a diesel engine without any major modifications.

CHAPTER-8

Results and Discussion: Modified Operating Parameters on a Diesel Engine using Preheated Intake Air and Blend Ratios

Overview

The prime objective of the thesis is to understand the behavior of the preheated POME biodiesel in blends and intake air preheated at elevated temperature, run diesel engine under variable CR-IP-IT condition. So, modification in the engine is need of time to balance 'the law of supply and demand'. The modification is an amendment or changes made to establish something which results towards betterment. Many researchers had worked on a modification of engine like a change in combustion chamber geometry, cooling system, compression ratio (CR), injection timing (IT), injection pressure (IP). The effect of these parameters, on performance, combustion and emission characteristics of the engine. The engine characteristics are also influenced by various parameters like properties of fuel (density, calorific value, etc.). The aim of this study is to investigate the modification done in the diesel engine engines in the form of CR, IP and IT and study its effect on the engine characteristics of CI engine. However, to realize this fact, it is necessary to know the effects of original diesel fuel in similar combinations of CR, IP and IT, which may be considered as the baseline knowhow from the perspective of preheated palm biodiesel in blends. Alongside, being a fuel of different origin of biodiesel, the standard design limits of a diesel engine is not suitable for preheated POME biodiesel blends. Therefore, in this study the experimental observation performed in a VCR diesel engine with preheated fuel blends and intake air is analyzed to find out its optimum performance. The overall analyses are segregated into performance (BSFC, BTHE and EGT), combustion (cylinder pressure, net heat release rate, ignition delay, PCP, and MHRR) and exhaust gas emission (CO, CO₂, HC and NO_x) studies of the preheated fuel and intake air run engine. In each of these sections, explanations are provided based on CR, IP and IT variations. The outcome of the experimental study is compared with respect to the diesel run engine performance at standard CR-IP-IT setting for quantitative and qualitative assertion. The purpose of this study is to understand the behavior of preheated intake air and POME biodiesel blends in a CI engine under various combinations of CR, IP, and IT at rated engine load and EGR rate.

8.1 Preface

Due to some advantageous properties of biodiesel, it shows promising results in terms of emission (reduce exhaust emissions) but negative impact on engine performance (BSFC and BTHE) as compared to diesel in a diesel engine. So, it is imperative to increase the performance of biodiesel with a reduction of the exhaust emissions in a diesel engine by modifying the parameters (CR, IP and IT). It has been proven that these parameters have a significant effect on the BSFC, BTHE, and EGT of diesel engines (Raheman and Ghadge, 2008). Improve the performance of the engine with higher compression ratio (CR), injection timing (IT) and injection pressure (IP), with lower emissions which are still lower than that with diesel fuel of different loads and biodiesels (Jindal et al., 2010). With the increase in IP, BSFC and BTHE undergo positive impact. Because at a high injection pressure, the droplet size decreases and its surface area increases. So, mixing of air-fuel become better during ignition delay which results in an increase in BTHE and decrease in BSFC. But after a particular limit, it affects the penetration of fuel inside the combustion chamber. So, a mixture of air-fuel may not be homogeneous and performance of the engine falls down. According to Debnath et al. (2013b), CR and IT are important parameters that influence the brake thermal efficiency of diesel engines. Increase in the temperature and pressure of the air-fuel mixture inside combustion chamber along with turbulence effect is the direct consequence of increasing CR in a diesel engine. So, the fuel inside the chamber evaporates in less time and homogeneous mixture can be achieved. Hence, higher the CR resulted in improved performance of CI engine. Performance of engine can also be improved by change in injection timing (IT) depending upon the fuel type. At advanced IT, ignition delay (ID) period will be longer by which more fuel accumulates in the combustion chamber and achieve better combustion with increase premixed combustion phase. IT advancement increases PCP, which in the end rises peak temperature (Heywood, 1988). This is the reason of increase in NO_x concentration through IT advancement. And for retarded IT, accumulation of fuel is less which results in smaller peak heat release rate but due to the shorter ID, combustion rate is faster which increase the mean effective pressure to do the work. Therefore, more work output for retarded IT so the performance will increase. Lots of researchers had worked on a modification of engine like a change in combustion chamber geometry, cooling system, coating on engine parts (piston etc.), compression ratio (CR), injection timing (IT), injection pressure(IP) etc. (Bapu et al., 2017; Bari and Saad, 2013; Benajes et al., 2015; Jaichandar and Annamalai, 2012; Singh et al., 2017; Yaliwal et al., 2016). The effect of

these parameters on performance, emission and combustion characteristics of the engine. The engine characteristics are also influenced by various parameters like engine design, properties of fuel (density, calorific value, flash point, etc.).

The studies on engine operating parameters with alternative fuels are somehow vague in the available literatures. The main objective of this study is to investigate the modification done in the diesel engines in the form of CR, IP and IT and its effect on the performance, combustion characteristics and emissions, and to find out the best combination of the optimum CR, fuel IP and IT of the computerized VCR DI single cylinder four stroke diesel engine using experimentation analysis of preheated POME biodiesel-diesel blends (PPBD20, PPBD40 and PPBD60) and diesel fuel (PBD0) at rated 90% of load and EGR30% of EGR rate. Various parameters defining the performance of VCR diesel engine are calculated and they are used as means for obtaining optimum CR, fuel IP and IT. By plotting performance, combustion and emission graphs of different CRs, different fuel IPs and different fuel ITs from that optimum CR, fuel IP and fuel IT obtained.

8.2 Experimental Setup and Procedures

Experiment was conducted in a 3.5 kW single cylinder, four stroke, water cooled, VCR DI diesel engine to determine the performance, combustion and emission parameters. The specification of the tested engine is shown in **Table 3.1**. The properties of the optimized preheated biodiesel-diesel blends tested fuels (PBD0, PPBD20, PPBD40, PPBD60) are shown in **Table 6.1**. The performance parameters like BSFC, BTHE, EGT were determined. The combustion parameters like, cylinder pressure, maximum cylinder pressure, heat release rate and maximum heat release were determined. The emissions (CO, HC CO₂ and NO_x concentrations) are recorded by using A five gas analyzer (AVL DI GAS 444- India make). To ensure the accuracy of the measured values, the gas analyzer is calibrated before each measurement using reference gases. For all settings, the emission values and the other values are recorded thrice and a mean of these is taken for comparison. The engine was run at varying compression ratio (15:1, 16:1, 17.5:1, and 18:1), injection opening pressure (188 bar, 200 bar, 212 bar and 224 bar), and injection timing (19°, 23° and 27° bTDC) at 90% load corresponding to load at maximum brake thermal efficiency using neat diesel fuel (PBD0) and preheated POME biodiesel-diesel blends (PPBD20, PPBD40 and

PPBD60) to determine how each fuel would perform under identical engine, and rated load and EGR30% EGR rate (test matrix **Table 8.1**).

The compression ratio (CR) can be changed without stopping the engine and without altering the combustion chamber and specially designed tilting cylinder block arrangement is used for varying the CRs. It can be varied from 15 to 18 in diesel mode. The engine was run with different injection pressure (IP) 188 bar, 200 bar, 212 bar and 224 bar; fuel IP was set by adjusting the spring pressure and was calibrated in the lab. Fuel injection timing was changed by loosening the locking nut and rotating the adjusting screw clockwise /anticlockwise for advancing or retarding fuel injection timing. The combustion and performance parameters are measured by computerized systems. The initial arrangements of the system are properly checked and test fuel is filled in the fuel tank. The engine is operated at rate 90% of load under constant 30% EGR rate. For every test, air flow, fuel flow, temperatures at water inlet and exhaust, rotameters values are noted. The engine exhaust gases (CO, HC, CO₂ and NO_x) are noted by using gas analyzer. The above values are noted for different CRs, fuel IPs and fuel Its, and the same procedure is repeated for every different blends of fuel. For every blend the performance and emission values are tabulated and compared with each other.

Table 8.1: The modified engine operating parameters test matrix.

Experiment	Operating and Design Condition			Test Fuels	Preheating Temperature(°C)	
	IP (bar)	IT (bTDC)	CR		Fuel	Intake air
<i>Investigation of engine performance and emission parameters of diesel engine running with diesel and preheated POME biodiesel-diesel blends at varying CRs at constant engine load of 90% and 30% EGR rate.</i>						
1.1 Baseline test	Default 200	Default 23	17.5	PBD0	Without preheating	Optimal IAPT 61°C
1.2 Preheating POME– diesel blend	Default 200	Default 23	15, 16, 17.5, 18	PPBD20 PPBD40 PPBD60	Optimal FPT 114 °C	Optimal IAPT 61°C
<i>Investigations engine performance, combustion and emission parameters of diesel engine running with preheated POME biodiesel-diesel blends at varying fuel IPs at constant engine load of 90% and 30% EGR rate.</i>						
1.3 Preheating POME– diesel blend	188, 200, 212, 224 bar	Default 23	Optimal CR 18	PPBD20 PPBD40 PPBD60	Optimal FPT 114 °C	Optimal IAPT 61°C
<i>Investigations engine performance and emission parameters of diesel engine running with preheated POME biodiesel-diesel blends at varying fuel ITs at constant engine load of 90% and 30% EGR rate.</i>						
1.4 Preheating POME– diesel blend	Optimal IP 212	19, 23, 27°	Optimal CR 18	PPBD20 PPBD40 PPBD60	Optimal FPT 114 °C	Optimal IAPT 61°C

8.3 Result and Dissuasion

Experiments are performed on the diesel engine by varying CRs, fuel IPs and fuel ITs to arrive at optimum configuration. The results are discussed below.

8.3.1 Effect of Compression Ratio on Diesel Engine Parameters

Variable compression ratio is technology to adjust the compression ratio of an internal combustion engine while the engine is in operation. This is done to increase fuel efficiency while under varying loads. Higher loads require lower ratios to be more efficient and vice versa. Variable compression engines allow for the volume above the piston at 'Top dead center' to be changed. The variable compression ratio multi fuel engine performances, combustion characteristics and emissions are discussed with respect to different CRs (15, 16, 17.5 and 18) of the three blends of preheated POME biodiesel with diesel blends (PPBD20, PPBD40 and PPBD60), and base diesel (PBD0). Here, the focus of research is to optimize the engine performance. The variation of CR varies from 15 to 16, 17.5 and 18, the test matrix of sequence of experiments are highlighted in **Table 8.1**. The fuel and intake air is preheated at elevated temperature of 114 °C and 61°C, respectively.

8.3.1.1 Effect of Compression Ratio on Engine Performance Parameters

The prime objective of this research is to investigate the engine performance parameters (BSFC, BTHE and EGT) affected by the use of fuel and intake preheating at varying CRs operating condition. The variations of engine performance parameters at different values of CRs at 15, 16, 17.5 and 18 with a constant rated load of 90% and 30% EGR rate at IP of 200 bar is presented in **Figure 8.1** to **Figure 8.2**. Error bars indicate the standard deviation in each run.

8.3.1.1.1 Brake Specific Fuel Consumption (BSFC)

Figure 8.1(a) shows the variation of BSFC with respect to compression ratio for preheated POME and diesel blends (PBD0, PPBD20, PPBD40 and PPBD60) for 90% load. From the graph it is observed that, as compression ratio increases BSFC decreases. This is due to the reason that at higher CR the power generated is high with respect to fuel consumption (Bawane et al., 2014). BSFC for CR 18:1 is comparatively lower than other compression ratios of 15:1, 16:1, 17.5:1. As CR increases from 15 to 18, BSFC decreases for all preheated POME biodiesel with diesel blends. It is seen that the consumption of PPBD20, PPBD40, PPBD60 is about 0.25652 kg/kW.hr, 0.27048 kg/kW.hr, 0.28631 kg/kW.hr, and base diesel (PBD0) of 0.24884 kg/kW.hr at CR of 17.5 while

that at CR of 18 is reduce to 0.25894 kg/kW.hr, 0.27156 kg/kW.hr and 0.28089 kg/kW.hr. Conversely, the BSFC of PPBD20, PPBD40 and PPBD60 blends at CR of 15 and 16, respectively higher. For the blend of PPBD20 it is lower than other blend. At higher percentage of blends, the BSFC increases. This may be due to fuel density, viscosity and heating value of the fuels. PPBD40 has higher energy content than PPBD20. At higher percentage of blends (PPBD60), the specific fuel consumption increases. This is due to the decrease in calorific value at higher blends (Muralidharan and Vasudevan, 2011). Here among the four CRs, BSFC is found to be the least at CR 18 because at higher CR's complete combustion of fuel takes place due to high temperature of compressed air. Also as percentage of preheated POME biodiesel in the blend increases BSFC also increases. And also as the percentage of biodiesel in the blend increases, BSFC increases. It is due to the lower calorific value of biodiesel and its blends compared to diesel, more amount of fuel is required to produce the same power output (D'Souza et al., 2015).

8.3.1.1.2 Brake Thermal Efficiency (BTHE)

It can be noted from **Figure 8.1(b)** that, the variation of BTHE with respect to CR for preheated POME biodiesel diesel blends at a load of 90% under EGR30% rate. It observed that as compression ratio increases BTHE also increases for preheated biodiesel diesel blends and reaches maximum value at the CR of 18. This is due to the increase in compression ratio ensures better air-fuel mixing and faster evaporation and leads to complete combustion (Silambarasan et al., 2015). BTHE is maximum at compression ratio 18. With the increase in CR from 15 to 18, the BTHE increased approximately up to 17.72%, 14.4% and 15.03% for PPBD20, PPBD40 and PPBD60. At CR 18 the BTHE of the engine operated with PPBD20 was slightly higher than diesel PBD0 at (CR17.5), whereas for PPBD40 and PPBD60 the BTHE was 5.32% and 9.92%, respectively less than diesel PBD0 at base CR of 17.5. But as biodiesel in blend increases, BTHE decreases. This is due to combined effect of its lower heating value and increase in fuel consumption (Peter et al., 2002).

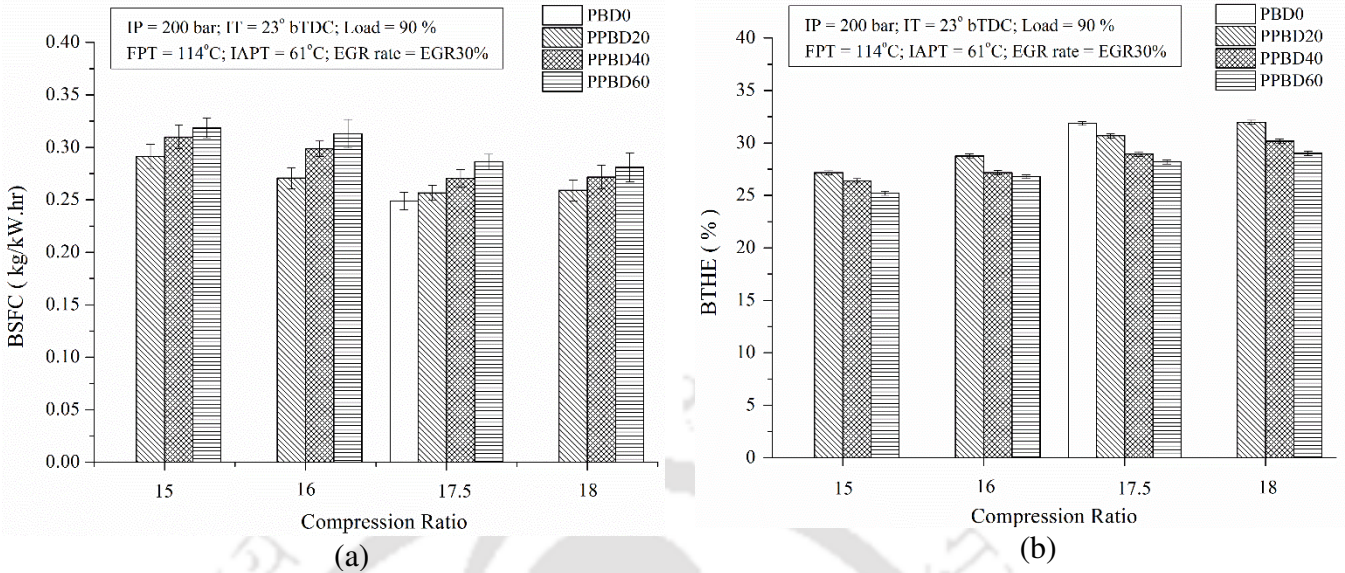


Figure 8.1: Variation of engine parameters with CRs for fuel samples: (a) BSFC, (b) BTHE.

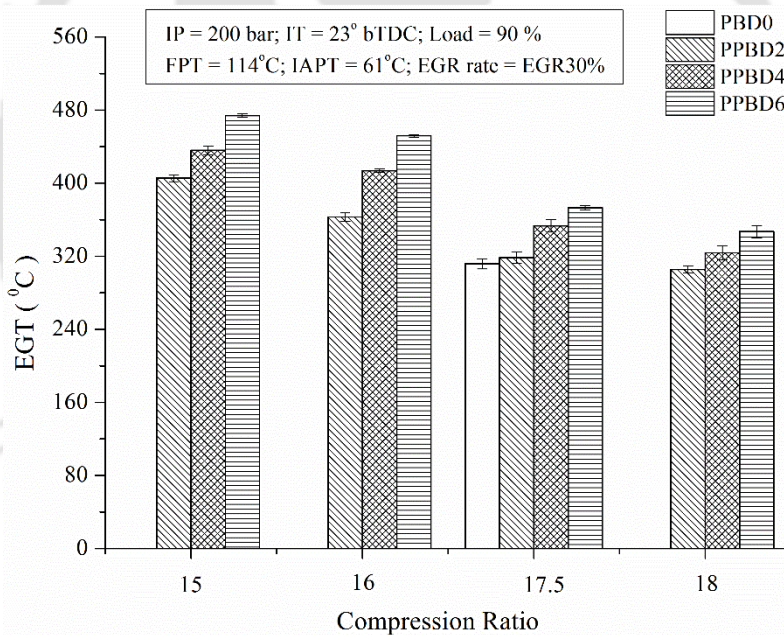


Figure 8.2: Variation of EGT with CRs for fuel samples.

8.3.1.1.3 Exhaust Gas Temperature (EGT)

It can be seen from the **Figure 8.2** that, EGT decreases with the increase in CR for all the fuels. It is observed that EGT reduces with the increase in preheated POME biodiesel percentage in blend. The rate of EGT drop is almost the same at different CRs for all fuels. The EGT reduces up to 24–26% for all fuels (PPBD20, PPBD40 and PPBD60) as the CR increases from 15 to 18. At CR

of 18, the values of EGT for preheated POME biodiesel diesel blends (PPBD20, PPBD40, and PPBD60) are found to be 305.456 °C, 323.687 °C and 346.876 °C, and diesel (PBD0) at base CR of 17.5 is 311.624 °C respectively indicating that EGT at higher CRs for blends are closer to diesel (CR17.5). As the compression ratio increases, the exhaust gas temperature of the various blends is decreasing than diesel (CR 17.5). The reason for this is preheated POME biodiesel blends have a low calorific value than diesel. So that it leads to lower temperature at the end of compression (Muralidharan and Vasudevan, 2011).

8.3.1.2 Effect of Compression Ratio on Combustion Characteristics

This section deals with the analysis of different combustion parameters which are cylinder pressure variation, maximum cylinder pressure, net heat release, and maximum heat release rate with different blending ratios at varying CRs at 15, 16, 17.5 and 18 with a constant rated load of 90% and EGR30% rate at IP of 200 bar is presented in **Figure 8.3** to **Figure 8.6**.

8.3.1.2.1 Cylinder Pressure

Figure 8.3(a–d) show the variation of pressure inside the cylinder with crank angle with preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) and diesel (PBD0) at varying CR of 18:1, 17.5:1, 16:1, 15:1. Zero degrees indicates the crank angle when the piston is at top dead center just at the end of the compression stroke of the four stroke engine. **Figure 8.4** shows the variation of maximum cylinder pressure in the combustion chamber for all blending ratios at varying compression ratios. It is clear from the results that as compression ratio increases, maximum cylinder pressure increases for all blending ratios. There is also marginal variation of maximum cylinder pressure with blending ratios. For the same compression ratio, the value of maximum cylinder pressure is higher for higher blending ratio (PPBD60). The value of maximum cylinder pressure is obtained for 18 compression ratio with 60% blending of biodiesel. Maximum cylinder pressure is 71.84 bar at 373 °CA with PPBD60 at CR18. The result shows that the minimum cylinder pressure (59.32 bar) is obtained for PBD0 at compression ratio 15. The result concluded that as the blending increases the cylinder pressure increases and as the compression ratio increases the maximum cylinder pressure increases. The possible supporting reason is short ignition delay and advanced injection timing for biodiesel (because of a higher bulk modulus, higher density and higher cetane number of biodiesel). The possible reason for the trends in the peak cylinder pressure is, because of the longer ignition delay for diesel than for biodiesel,

combustion starts later for diesel fuel. As a result, the peak cylinder pressure attains a lower value as it is further away from the top dead center (TDC) in the expansion stroke. It is concluded from this discussion that as biodiesel percent increases then the peak cylinder pressure will increase. (Mueller et al., 2009). It is clear that the combustion characteristics of biodiesel blends are better at higher CRs.

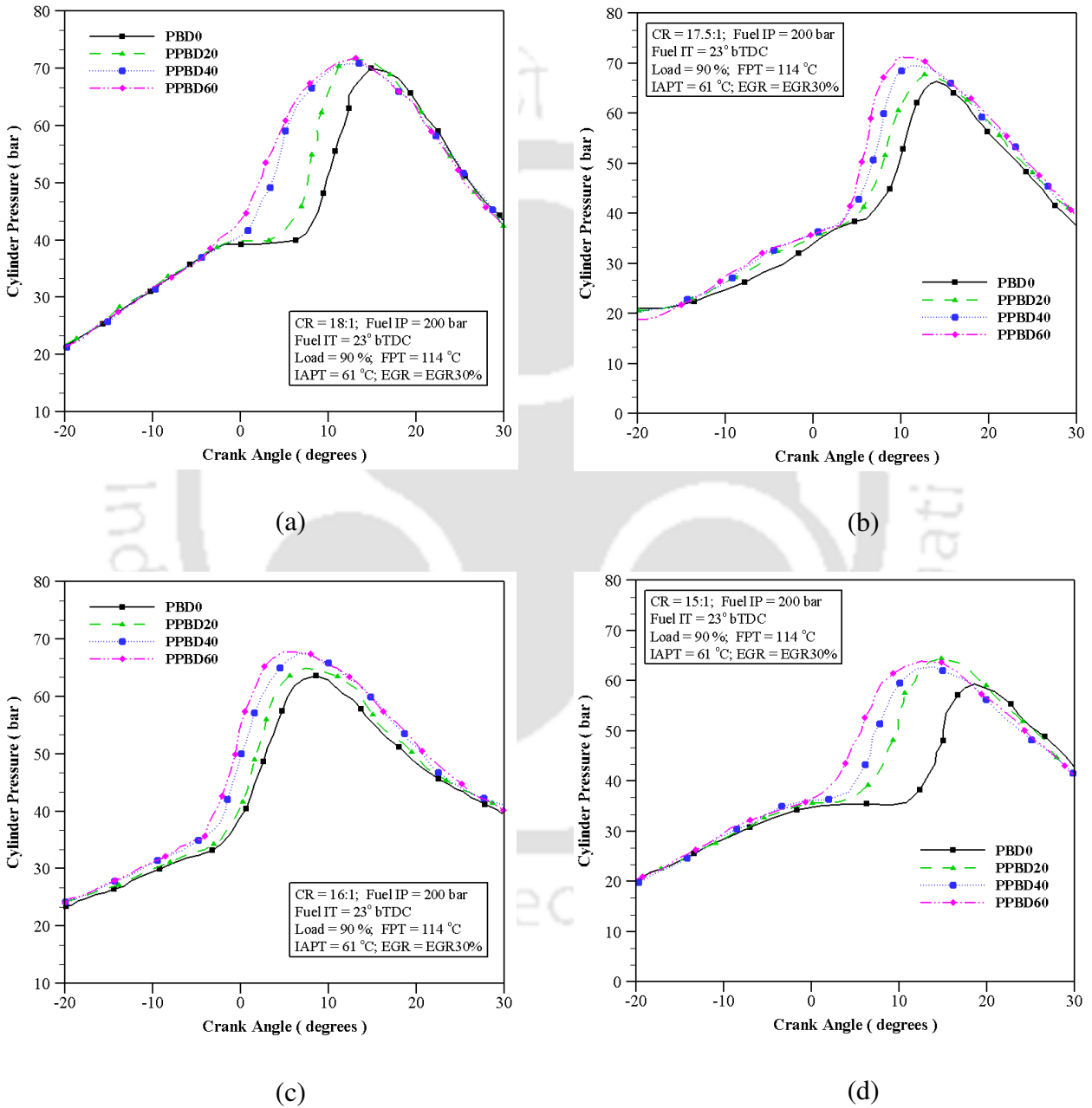


Figure 8.3: Variation of in-cylinder pressure with crank angle at various CRs: (a) CR 18; (b) CR 17.5; (c) CR 16; (d) CR 15.

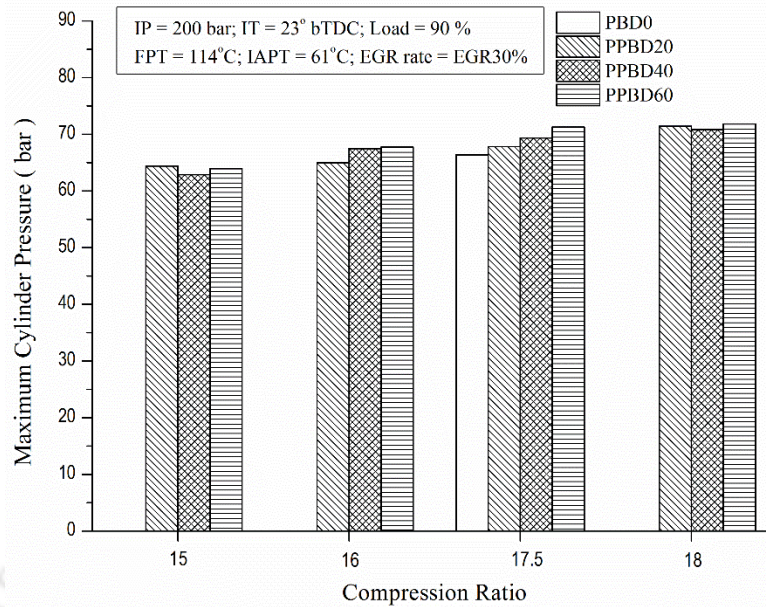


Figure 8.4: Variation of maximum cylinder pressure with varying CRs for different blends.

8.3.1.2.2 Net Heat Release

Figure 8.5(a–d) show the variation of net heat release ($J/^\circ CA$). Tests were performed at different preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) and diesel (PBD0) at varying compression ratios. The results show the variation of net heat release with crank angle 30 degree before top dead center in compression stroke and 30 degree after top dead center in power stroke. The maximum heat release rate for fuel blends PPBD20, PPBD40 and PPBD60 has been measured at different CRs and its value at CR 15 is 57.64, 51.36 and 41.17 J/deg respectively, and the maximum value is obtained for pure diesel, PBD0 (64.37 J/deg) at 366.32 degree crank angle. This may be attributed to the higher calorific value of diesel and more fuel accumulating owing to longer ignition delay (ID) (due to lower bulk modulus of diesel as compared to biodiesel) would increase the amount of fuel burned during the premixed combustion phase, causing a higher net heat release. Szybist et al. (2005), similarly at compression ratio 18 its value is 46.19, 42.26, 41.02 and diesel (PPBD0) 47.61 J/deg. Thus, net heat release is maximum for PPBD20 at compression ratio 18 at 362.92 degree crank angle. It has been seen that at higher compression ratio maximum (CR18) net heat released is obtained earlier as compared to lower compression ratio (CR15) because at higher CR cylinder temperature will be more and hence ignition delay will be less and ignition of fuel will start at an earlier stage. It is clear from **Figure 8.6** that as the

compression ratio increases the value of maximum net heat release decreases. This might be because at higher CR, combustion starts earlier and net heat release per degree crank angle will be less.

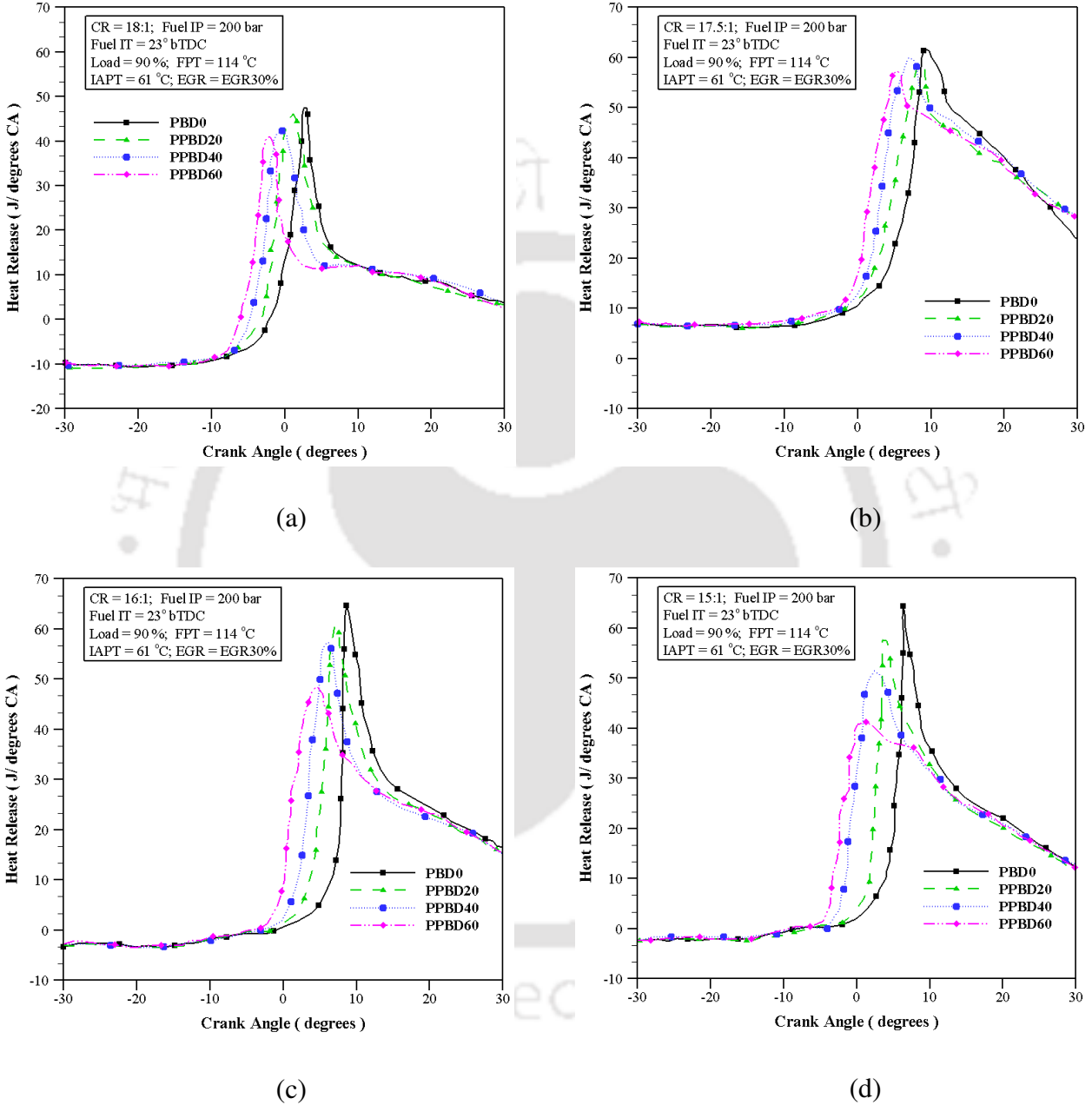


Figure 8.5: Variation of net heat release with crank angle at various CRs: (a) CR 18; (b) CR 17.5; (c) CR 16; (d) CR 15.

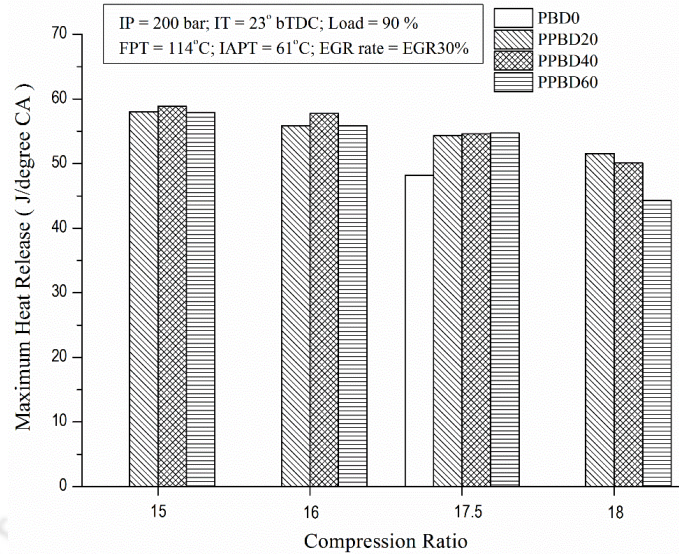


Figure 8.6: Variation of net heat release with varying CRs for different blends.

8.3.1.3 Effect of Compression Ratio on Emission Characteristics

The variation of emission constituents with CR considered are carbon monoxide (CO), unburned hydrocarbon (HC), carbon dioxide (CO₂), oxides of nitrogen (NO_x). The variations of exhaust emission constituents at different values of CRs at 15, 16, 17.5 and 18 with a constant rated load of 90% and EGR30% rate at IP of 200 bar is presented in **Figure 8.7** to **Figure 8.8**. Error bars indicate the standard deviation in each run.

8.3.1.3.1 Carbon Monoxide (CO) Emission

The variation of CO with respect to CR for diesel (PBD0) and preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) at a load of 90% under EGR30% rate is shown in **Figure 8.7(a)**. It is observed that as CR increased, the CO emission is reducing for all the fuels. This is due to better combustion of fuel at higher CR better combustion occurs due to high air temperature in the cylinder (Bawane et al., 2014). It can also be observed that CO emissions are higher at low CRs, and lesser at higher CRs. At lower CR, insufficient heat of compression delays ignition and so CO emissions increase. The possible reason for this trend could be that the increased CR actually increases the air temperature inside the cylinder therefore reducing the ignition lag which causes better and more complete burning of the fuel (Ramadhas et al., 2006). Also as percentage of biodiesel in the blend increased, emission of CO is decreased, and lesser for blends compared

to diesel (PPBD0). At CR of 18, CO emissions are 25.02%, 38.25% and 57.44% less for preheated POME biodiesel blends (PPBD20, PPBD40 and PPBD60) as compared to diesel (PBD0) at CR of 17.5. At a CR of 18 the values of CO emission are a minimum among the three CRs.

8.3.1.3.2 Carbon Dioxide (CO₂) Emission

More amount of CO₂ is an indication of complete combustion of fuel in the combustion chamber. **Figure 8.7(b)** shows the variation of CO₂ with respect to CR for preheated POME biodiesel diesel blends (PBD0, PPBD20, PPBD40 and PPBD60) at a load of 90%. It is observed that as CR increases, emission of CO₂ increases. This is due to the better combustion (Bawane et al., 2014). Also as percentage of preheated POME biodiesel in blend increases, the CO₂ emission decreases. This may be due to time consumption high to burning of fuel leading to incomplete oxidation of CO (Mani et al., 2011). At a CR of 18 for PPBD20, PPBD40 and PPBD60 fuel the CO₂ emission decreases by 8.64%, 13.9% and 21.1% compared to the emission of diesel PBD0 at a CR of 17.5.

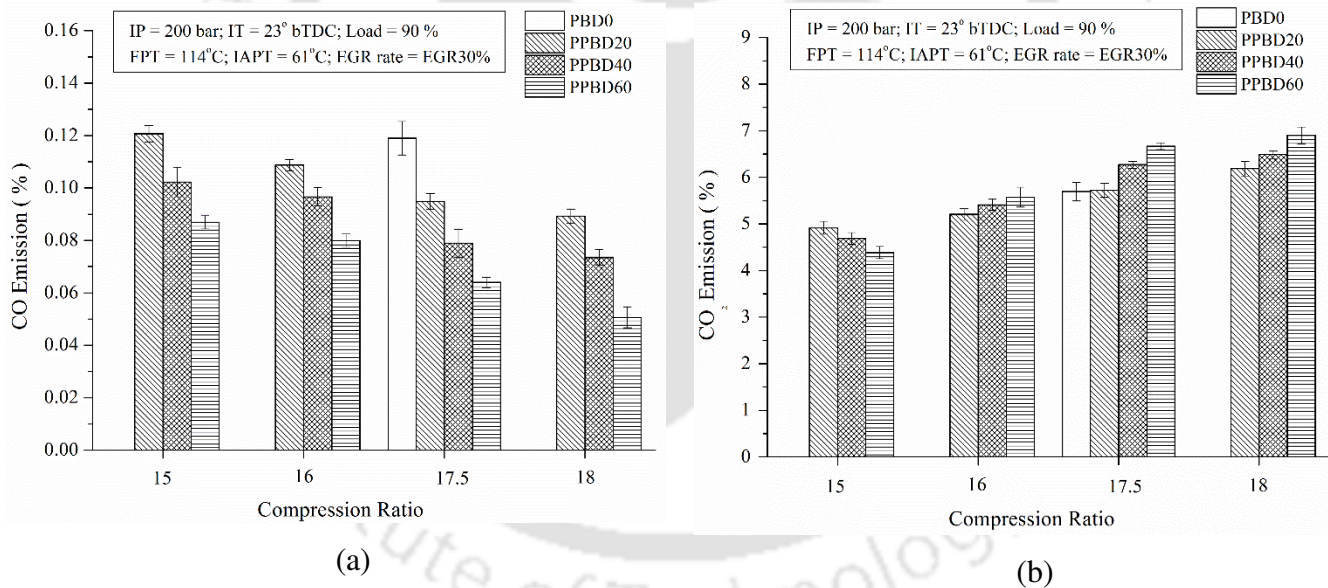


Figure 8.7: Variation of exhaust emissions with CRs for fuel samples: (a) CO, (b) CO₂.

8.3.1.3.3 UnburntHydrocarbon (HC) Emission

Emission product that is produced by the diesel engine is Unburned Hydrocarbon emission (HC). It consists of fuel that is only partially (or) completely unburned. The amount of HC depends on the engine operating fuel properties and condition. **Figure 8.8(a)** shows variation of HC with

respect to compression ratio for preheated POME biodiesel diesel blends at a load of 90%. It is observed that as compression ratio increases, emission of hydrocarbon decreases which is due to complete combustion of fuel at higher CR. This is due to the increase in air temperature at the end of compression stroke, enhancement in combustion temperature and reduction in charge dilution leads to complete combustion and reduction in hydrocarbon emissions (Ramadhas et al., 2006). **Figure 8.8(a)** shows HC emission for preheated POME biodiesel blends are lower than the biodiesel blend, as percentage of biodiesel in blend increases, emission of HC also decreases. This may be due to the inbuilt oxygen content in the molecular structure of POME biodiesel this may be responsible for complete combustion and this reducing the HC levels. The HC emission is least at compression ratio 18 among the four CRs. At a CR of 18, the mean percentage decrease in HC emission with preheated POMR biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) are decreased by 25.64%, 33.33% and 69.23% as compared to diesel (PBD0) base CR 17.5.

8.3.1.3.4 Oxides of Nitrogen (NOx) Emission

The NOx emission of diesel engine is formed during combustion because of high temperature inside the cylinder. **Figure 8.8(b)** shows the variation of NOx with respect to CRs (18, 17.5, 16 and 15) for diesel (PBD0) and preheated POME diesel blends (PPBD20, PPBD40 and PPBD60) at a load of 90%. It is observed that as CR increases, emission of NOx also increases. The minimum value of NOx emission was found at the CR of 15:1 and it increases as the compression ratio increases. Hence highest NOx for preheated POME biodiesel blends is observed at a compression ratio 18. At higher CRs of 15 to 18 NOx emission is found to be higher for the preheated biodiesel blends (PPBD20, PPBD40 and PPBD60) are higher by 26.58%, 39.86% and 49.81%, respectively, they are higher by 39.52%, 66.93% and 84.68% as compared to diesel (PBD0) at CRs of 17.5. Also as percentage of preheated biodiesel in blend is increased, emission of NOx also increased. This is due to the availability of excess oxygen in the biodiesel. This contributes towards NOx production (Pradeep and Sharma, 2007).

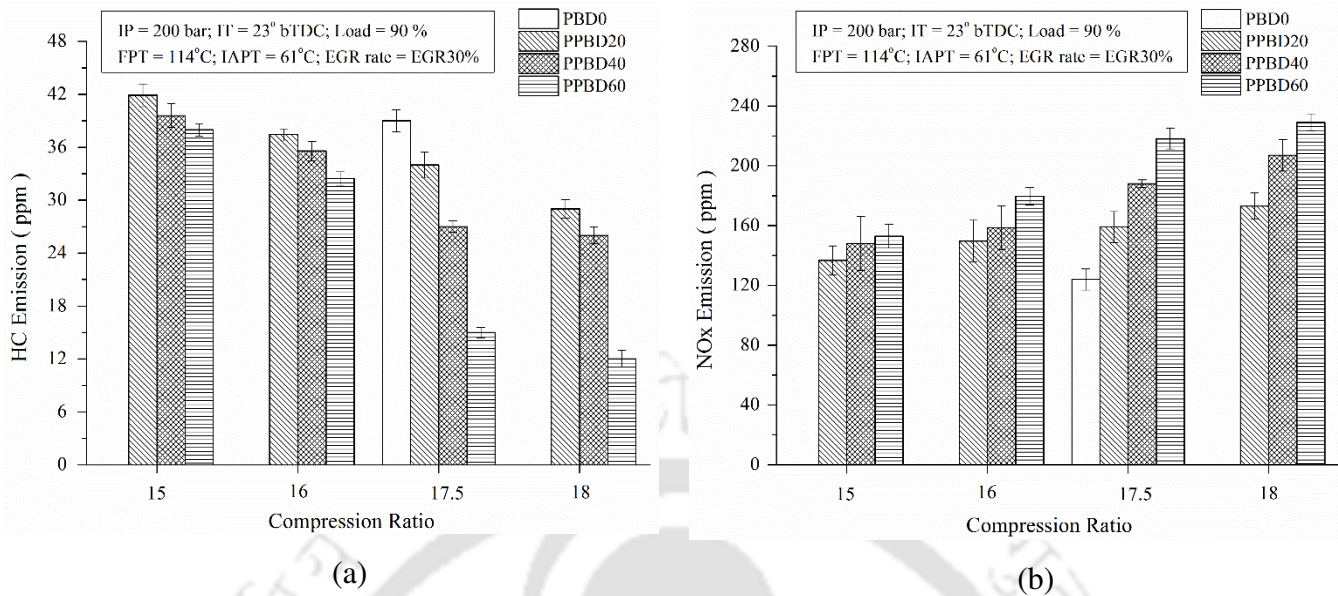


Figure 8.8: Variation of exhaust emissions with CRs for fuel samples: (a) HC, (b) NOx.

In general, the performance of preheated biodiesel approaches that of diesel fuel at higher CRs. Therefore, it can be concluded that higher CR should be the mode of operation when engine is fuelled with preheated POME biodiesel blends (PPBD20, PPBD40 and PPBD60). At higher compression ratios (15 to 18), combustion of fuel is efficient due to high temperature of compressed air. Due to which, the exhaust emissions are found to reduce at higher CRs. However, the NOx emissions are found to increase at higher CRs with PPBD20, PPBD40 and PPBD60 as compared to diesel (PBD0). So the selection of CR can be based on the combined effect on thermal performance and emission characteristics. It is preferable to operate the engine at CR of 18 for optimum thermal performance. If NOx is considered, then it is better to operate at CR 15. But it causes decrease in BTHE of (PPBD20, PPBD40 and PPBD60) about 11.51%, 8.86% and 10.57% and increase of BSFC of about 13.5%, 14.5% and 11.21% which are not recommendable just to reduce NOx emissions. It is found that CO₂ emissions are more for preheated biodiesel blends than that of diesel. Higher CO₂ emissions reduce harmful CO emissions. At CR of 18, the percentage reduction in CO and HC emissions for preheated POME biodiesel blends is significantly reduced as compared to that of diesel (CR17.5). Due to higher NOx emissions with preheated POME biodiesel blends, suitable blends can become a striking balance between NOx emissions on one end and all other emissions along with performance on the other hand. The emission characteristics show that the preheated POME biodiesel blends give minimum harmful emissions at CR of 15 as

compared to diesel (CR17.5). Further, at a higher CR of 18 the fairly reduced exhaust emissions are observed irrespective of the fuel. Therefore, in operating the diesel engine with preheated POME biodiesel blends at a CR of 18 results in minimum emissions but more NO_x emissions. It has been shown that the combustion characteristic results are better at higher compression ratio (CR18) for all preheated POME biodiesel diesel blending ratios. Ignition delay of biodiesel is smaller as compared to neat diesel. As the CR increases, ignition delay decreases for all blending ratios. Maximum cylinder pressure is obtained for PPBD60 at compression ratio 18. Maximum heat release is obtained for PBD0 at CR18. Net heat release decreases as CR increases.

8.3.2 Effect of Fuel Injection Pressure on Diesel Engine Parameters

Fuel injection pressures (IPs) are some the important parameters which influence the performance and emission characteristics of diesel engine. The combustion and formation in a diesel engine is governed mainly by fuel spray formation and mixing rate. The increase in fuel IP leads to better atomization of fuel, improved spray characteristics and reduced physical delay period; which enhances premixed combustion and rapid combustion rate with improvements in engine performance parameters and reduction CO and HC emission levels. However too high fuel IP, will lead to delayed injection, which may responsible to higher velocity of droplet which will pass away without mixing air properly and lower engine performance parameters and increased emission levels due to improper combustion. The performance and emission parameters considered in the present study are BSFC, BTHE and EGT, and CO, CO₂, HC and NO_x emissions. The effect of IP on the above parameters for blends of diesel and preheated biodiesels (PPBD20, PPBD40 and PPBD60) at a CR of 18 and diesel, PBD0 at 17.5, and constant load of 90% and EGR30% rate operating conditions. The fuel IP varies from 188 to 224 bar with 12 bar increments, the test matrix of sequence of experiments are highlighted in **Table 8.1**. The fuel and intake air is preheated at elevated temperature of 114 °C and 61°C, respectively.

8.3.2.1 Effect of Fuel Injection Pressure on Performance Parameters

The prime objective of this research is to investigate the engine performance parameters (brake specific fuel consumption, brake thermal efficiency and exhaust gas temperature) affected by the use of fuel and intake preheating at varying fuel injection pressures operating condition. The variations of engine performance parameters at different values of fuel IPs at 188, 200, 212 and 224 bar with a constant rated load of 90% and EGR30% rate at CR of 18 for preheated POME

biodiesels diesel blends (PPBD20, PPBD40 and PPBD60), and CR of 17.5 for diesel (PBD0) is presented in **Figure 8.9** to **Figure 8.10**. Error bars indicate the standard deviation in each run.

8.3.2.1.1 Brake Specific Fuel Consumption (BSFC)

Figure 8.9(a) presents effect of IP on BSFC for preheated POME biodiesel and its blends with diesel. It can be observed that BSFC reduces with the increase in fuel IP for all the fuels tested. This is due to better atomization at higher IP which exposes larger surface area of fuel droplet to the high temperature air leading to complete combustion of fuel. BSFC is observed to be higher for higher blend proportion at a constant IP. The trend is mainly because of relatively higher viscosity and lower calorific value of POME biodiesel. It can be understood that, BSFC is reduced with the rise in fuel IP (212 bar) from a standard 200 bar. The higher the fuel IP, the better the atomization of the fuel, this leads to a better air-fuel mixture formation and combustion characteristics and improved performance. The main reason is at this fuel IP, fuel atomization and spray characteristics of fuel injection process, and combustion efficiency significantly improved. However, increased fuel IP from 200 to 224 bar, the BSFC slightly increased, due a more homogeneousness of the atomized mixture (Wategave et al., 2014). At 212 bar fuel IP, BSFC for preheated biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) is lower by (0.23765, 0.24945, and 0.25487 kg/kW.hr), respectively as compared to that of diesel (PBD0) of 0.24884 kg/kW.hr at IP of 200 bar.

8.3.2.1.2 Brake Thermal Efficiency (BTHE)

The brake thermal efficiency (BTHE) of the engine is an important performance parameter of interest for any fuel used as far as CI engine performance is concerned. It is an indication of efficiency of fuels used i.e. the ratio of work output of the engine to the energy input in general. The BTHE of the engine is influenced by various parameters (such as mass of fuel consumed, mass of air required, calorific value of fuel, density of fuel and air, viscosity of fuel, heat carried away by auxiliaries, compression ratio, and the service conditions of test engine etc.), while using different fuels having different properties. For the above mentioned reasons, the percentage values of BTHE of all test fuels shows lower from the expected efficiency values (**Figure 8.9-b**). This leads the amount of utilize the fuel energy supplied converted into useful power developed in the engine cylinder at the end of combustion process is caused to lower as compared to the normal operating conditions. The effect of IP on BTHE for preheated biodiesel blends is presented in

Figure 8.9(b). It is found that the BTHE increases with the increase in fuel IP for all the fuels tested. This is mainly due to higher degree of atomization at higher IPs ensuring complete combustion though the amount of fuel being injected is the same across different injection pressures. It is also observed that BTHE increases linearly for preheated biodiesel, PPBD20, PPBD40, PPBD60 and Diesel fuel (PBD0). The trends for BTHE can be observed to be opposite to the BSFC for all test fuels. It is seen that, the BTHE of preheated POME biodiesel diesel blends is higher at 212 bar. The BTHE was maximum for diesel (PBD0) at original fuel IP 200 bar, which is 31.87%, this is because of its higher calorific values of diesel. However, for PPBD20, PPBD40 and PPBD60, maximum BTHE was found 32.38%, 31.68% and 30.2% at 212 bar fuel IP. At 212 bar fuel IP, average increased BTHE of PPBD20, PPBD40 and PPBD60 was by 1.28%, 4.97% and 4.14% compared to 200 bar original fuel IP. While about 224 bar fuel IP, the BTHE of POME (114 °C) dropped as compared to 212 bar fuel IP. The overall results revealed that, about 212 bar fuel IP was the best suitable combination for preheated POME biodiesel diesel blends operated in a diesel engine. The fuel IP beyond 212 bar, may lead to shorter ignition delay period and thus affect the possibilities of homogenous mixture, fall down combustion efficiency and drop in performance parameters.

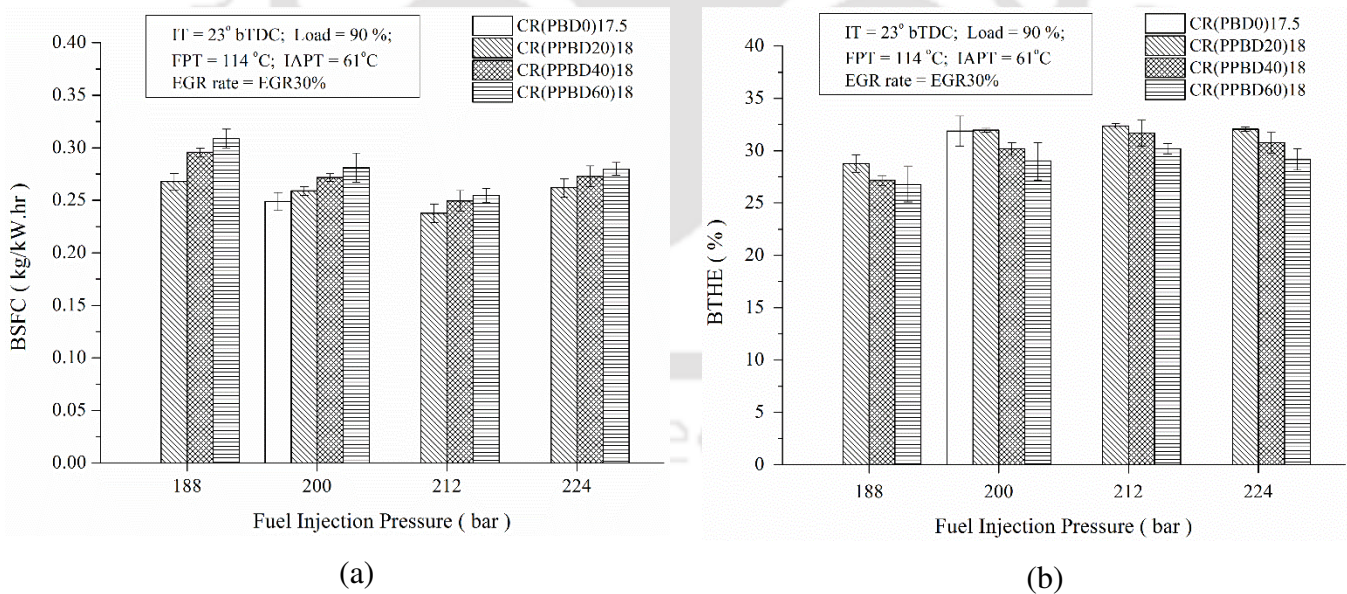


Figure 8.9: Variation of engine parameters with fuel IPs for fuel samples: (a) BSFC, (b) BTHE.

8.3.2.1.3 Exhaust Gas Temperature (EGT)

The effect of IP on EGT for preheated POME biodiesel with diesel blends is shown in **Figure 8.10**. It is seen that EGT increases linearly with the increase in IP for all the fuels tested. The trend is due to complete combustion of fuel at higher IP due to which more heat is generated in the exhaust. It is also observed that EGT is higher for blends compared to diesel (PBD0). The trend may be due to lower calorific value of POME biodiesel. At IP of 212 bar, the values of EGT for preheated POME biodiesel blends, PPBD20, PPBD40, PPBD60 and Diesel (PBD0) are found to be 346.567 °C, 379.269 °C, 405.341 °C and 311.624 °C respectively. The increase in EGT by 41.11 °C, 55.58 °C and 58.46 °C, respectively. It is around 13.46%, 17.17% and 16.85% higher as the IP is increases from 200 to 212 bar.

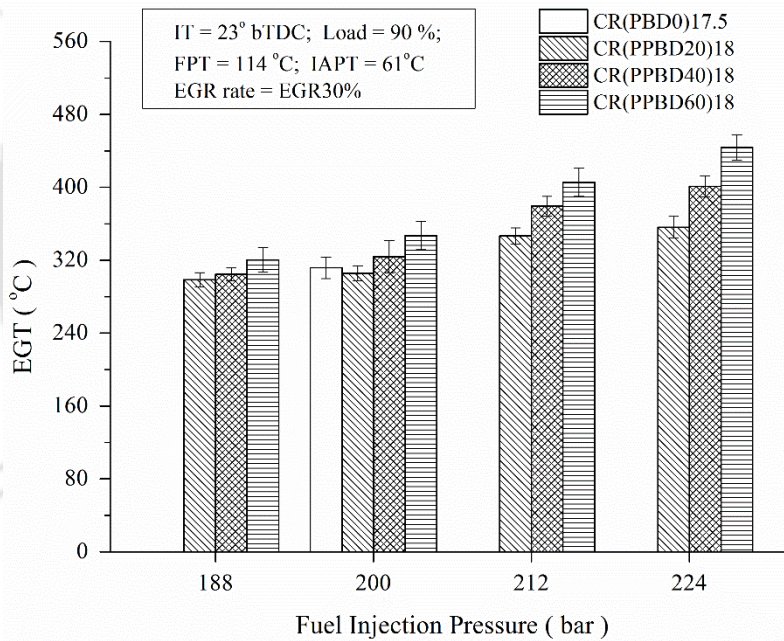


Figure 8.10: Variation of EGT with fuel IPs for fuel samples.

8.3.2.2 Effect of Fuel Injection Pressure on Exhaust Emissions

The emission constituents considered are CO, CO₂, HC and NO_x. The variations of emission constituents at different values of fuel IPs at 188, 200, 212 and 224 bar with a constant rated load of 90% and EGR30% rate at CR of 18 for preheated POME biodiesels blends (PPBD20, PPBD40

and PPBD60), and CR of 17.5 for diesel (PBD0) is presented in Figures **Figure 8.11** to **Figure 8.12**. Error bars indicate the standard deviation in each run.

8.3.2.2.1 Carbon Monoxide (CO) Emission

Figure 8.11(a) displays the influence of fuel IP on CO emission for diesel (PBD0) and preheated POME biodiesel blends (PPBD20, PPBD40 and PPBD60) at 90% load under EGR30% rate. The CO emission reduces with increase in fuel IP. This is due to the reason that the fuel is atomized into very fine droplets and more surface area is available for combustion which results in formation of a good quality fuel mixture which causes complete combustion. The CO emission of preheated biodiesel blends (PPBD20, PPBD40 and PPBD60) was found lower at 212 bar fuel IP compared to the CO emission at base fuel IP of 200 bar. Increasing the fuel IP to 212 bar causes more atomization of fuel making small droplets. Now as the droplets are small, so surface to volume ratio is large, which helps in easy heat transfer among the drops causing easy combustion. However, a further increase of fuel IP to 224 bar, the CO was slightly decreased due to the non-uniform in composition of the mixture throughout because of a lack of the air entrainment required for the achievement of a stoichiometric mixture (Karra and Kong, 2010). Conversely, decreasing the fuel IP to 188 bar for all preheated POME biodiesel diesel blends, the level of CO emission increased. It is also observed that CO emission reduces as the blend proportion is increased. This is because as the concentration of POME biodiesel in the blend increases the percentage of oxygen in the blend also increases. When the fuel is burnt, more of carbon combines with oxygen forming CO₂ resulting in lesser CO emissions. When the IP is increased from 200 bar to 212 bar, the CO emissions reduce up to 11.5%, 6.44%, and 3.42% for preheated POME biodiesel diesel blends, PPBD20, PPBD40 and PPBD60 respectively. However, the level CO emission was decreased by (6.84%, 2.12%, and 5.92%) at 224 bar as compared to base fuel IP of 200 bar. At fuel IP of 212 bar, CO emissions for preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) are lesser by about 33.65%, 42.22% and 58.89% as compared to diesel PBD0 (200 bar). Hence, amongst all fuel IPs, the lower CO emission occurred at 212 bar fuel IP for all fuel samples.

8.3.2.2.2 Carbon Dioxide (CO₂) Emission

Figure 8.11(b) gives the effect of IP on CO₂ emissions for preheated POME biodiesel blends at constant rated engine load of 90% and EGR30% rate. The CO₂ emission increases with the increase in IP. The CO₂ emissions increase by 26% and 22% for preheated POME biodiesel blends

(PPBD20, PPBD40 and PPBD60) as the IP increases from 200 to 212 bar. At an IP 212 bar, CO₂ emission for PPBD20, PPBD40 and PPBD60 are found to increase by about 0.7434%, 3.098% and 4.16%, respectively as compared to base fuel IP of 200 bar. While with a further increased to 224 bar, the percentage of CO emission slightly reduced by 0.32%, 1.17%, and 1.93% related to 200 bar fuel IP at standard conditions, due to a lack of air for proper combustion. The CO₂ emissions also increase with blend proportion. This is because of high oxygen content in the biodiesel due to which more of the carbon gets oxygenated during combustion which results in higher CO₂ emission.

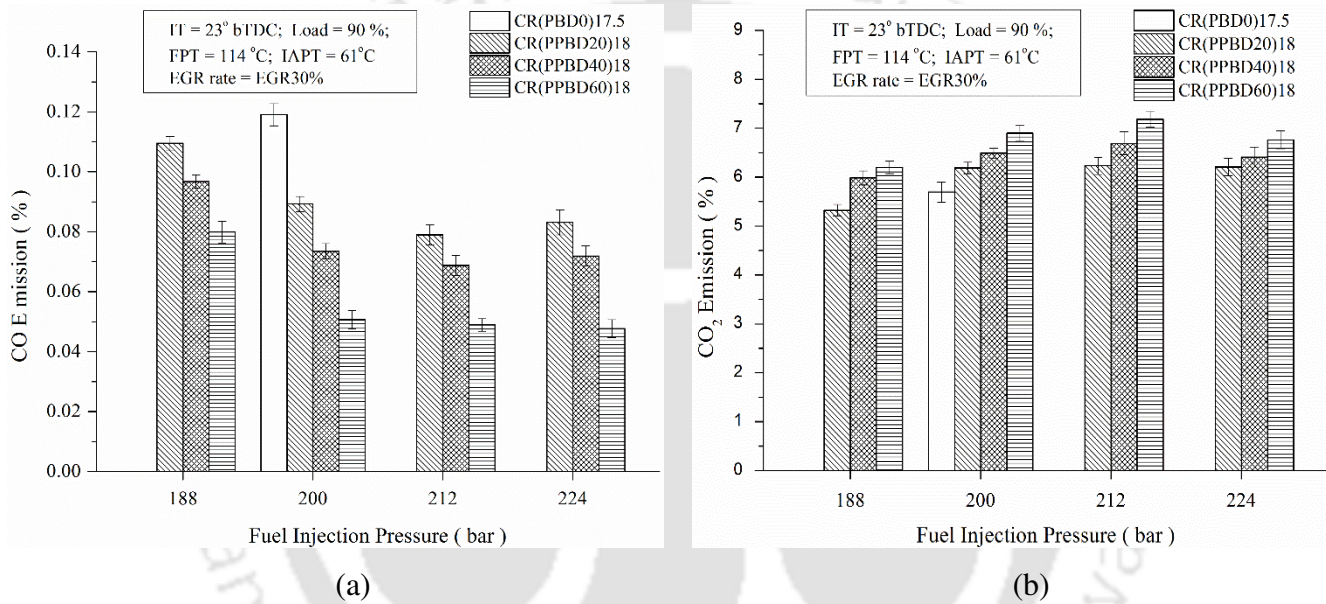


Figure 8.11: Variation of exhaust emissions with fuel IPs for fuel samples: (a) CO, (b) CO₂.

8.3.2.2.3 UnburntHydrocarbon (HC) Emission

The HC emission is the result of incomplete combustion of fuel. The effect of IP on HC emission for preheated POME biodiesel blends at 90 % loading and EGR30% rate is shown in **Figure 8.12(a)**. It is seen that HC reduces significantly with increase in IP. This is due to better combustion of fuel at higher IP. It is realized that; HC emission for PPBD20, PPBD40 and PPBD60 decreases considerably with rise in fuel IP 212 bar from base fuel 200 bar. This is due to better vaporization and mixing rate of fuel with increasing fuel IP. When fuel IP drops to 188 bar, HC emission increased, due to a decrease width of fuel fine droplets. With increase of fuel IP from 224 bar from 200 bar, HC emission slightly increased compared to engine operated a standard 200 bar fuel IP.

This is because of the occurrence of additional fuel in cylinder combustion chamber and the unburned hydrocarbons are high (Karra and Kong, 2010). The HC emissions decrease by 10.34%, 11.54% and 33.33% for preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60), respectively when the IP increases from 200 to 212 bar. When it was operated at 224 bar was slightly decreased by 3.84–6.89%, respectively compared to a base fuel injection pressure of 200 bar fuel IP. The decrease in HC emission with increase in blend proportion may be due to complete combustion as a result of higher oxygen content.

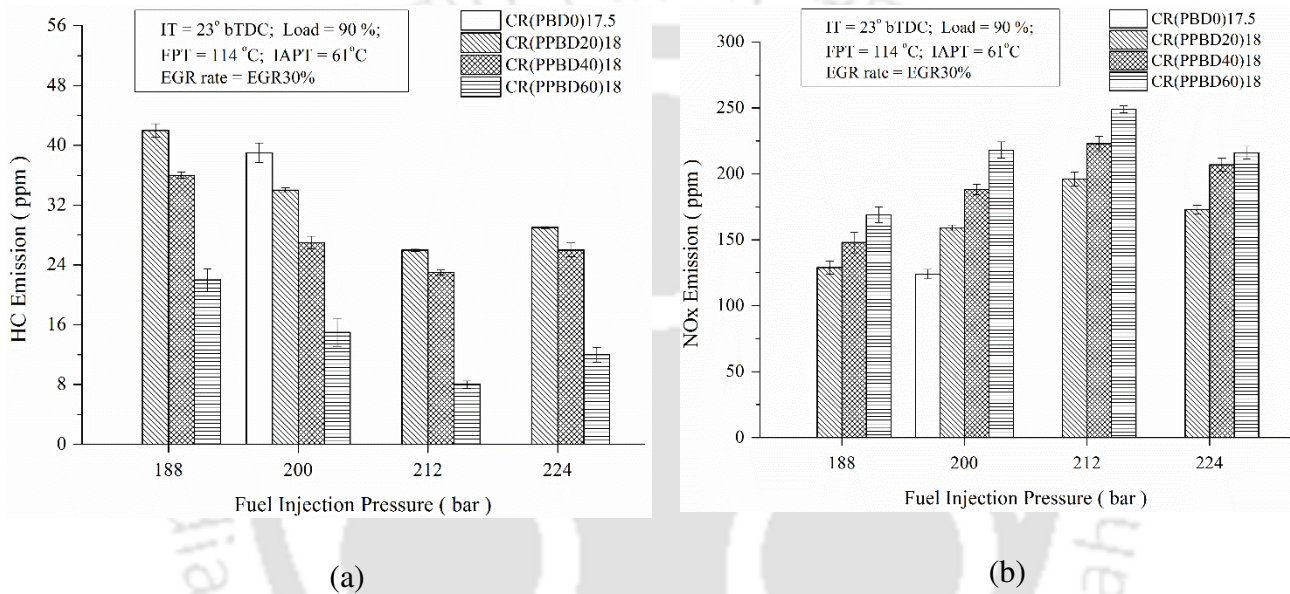


Figure 8.12: Variation of exhaust emissions with fuel IPs for fuel samples: (a) HC, (b) NO_x.

8.3.2.2.4 Oxide of Nitrogen (NO_x) Emission

Figure 8.12(b) illustrates the consequence of fuel IP on NO_x emission for preheated POME biodiesel blends at 90% of load and EGR30% rate. The level NO_x emission for preheated POME biodiesel with diesel blends (PPBD20, PPBD40 and PPBD60) was shown higher related with diesel (PBD0), because of the existence of extra oxygen in the biodiesel, which results improved ignition. It offered maximum heat release rate in premixed combustion phase and produced greater cylinder temperature. This causes increased NO_x formation in exhaust gas which strongly depends on temperature. The NO_x emissions increase with increase in IP. This is due to burning of more fuel at higher IP. At an IP of 212 bar, the NO_x emission for preheated POME biodiesel blends

(PPBD20, PPBD40 and PPBD60) are 196 ppm, 223 ppm and 249 ppm higher than that for diesel PBD0 (200 bar) of 124 ppm. As the IP increases from 200 to 212 bar, the NO_x emissions increase by about 13.29%, 7.73% and 8.73% for PBD0, PPBD20, PPBD40 and PPBD60, respectively. However, when fuel IP decreases from the standard 200 bar fuel IP to 188 bar, NO_x emission of PPBD20, PPBD40 and PPBD60 was decreased by 34.11%, 39.86% and 35.5%. The NO_x emissions increase as the blend proportion is increased. This is mainly due to higher oxygen content in biodiesel and higher temperature in the engine cylinder. Thus, the optimal fuel IP of 188 bar is suitable for reduction of NO_x emission.

In general, based on the experimental studies of effect of fuel IPs, the following are the observations and summaries: A single cylinder four stroke VCR CI engine may be operated on preheated POME biodiesel diesel blends without any system hardware modifications. At higher injection pressure of 212 bar, the thermal performance of preheated POME biodiesel-diesel blends (PPBD20, PPBD40 and PPBD60) are closest to that of diesel (PBD0) at 90% load under EGR30% rate. Preheated POME biodiesel diesel blend (PPBD60) gives minimum harmful emissions as compared to all other blends. Further at a higher compression ratio of 18 and injection pressure of 212 bar, fairly reduced emissions are observed irrespective of the fuel used. The NO_x emissions increase with increase in IP, and it increases as the blend proportion is increased. The PPBD20 and PPBD40 blends gives better thermal performance compared to other blend of biodiesel, but it is not recommended because it causes higher levels of exhaust emissions.

8.3.3 Effect of Fuel Injection Timing on Diesel Engine Parameters

Fuel injection timing essentially controls the crank angle at which combustion starts. While the state of the air into which the fuel is injected changes as injection timing is varied and thus ignition delay would vary, these effects are predictable. If injection starts earlier (advanced IT), initial air temperature and pressure are lower and so the ignition delay would increase. If injection starts later (retarded IT), air temperature and pressure are slightly higher initially, and a decrease in ignition delay would occur. This means that fuel ITs variation has a strong effect on ignition delay, and consequently on engine combustion and emissions because of the changing maximum pressure and temperature in engine cylinder (Rostami et al., 2014). The performance and emission parameters considered in the present study are BSFC, BTHE and EGT, and CO, CO₂, HC and NO_x emissions. The effect of IT on the above parameters for blends of diesel (PBD0) and preheated biodiesels (PPBD20, PPBD40 and PPBD60) at a constant load of 90% and EGR30%

rate operating conditions. The fuel and intake air is preheated at elevated temperature of 114 °C and 61°C, respectively. Three different fuel ITs (19°, 23° and 27° bTDC) are considered, which varies at 19° bTDC, retarded by 4° (from 23° bTDC); at 27° bTDC, advanced by 4° (from 23° bTDC); and at default 23° bTDC, the test matrix of sequence of experiments are highlighted in **Table 8.1**, and discussed in this section.

8.3.3.1 Effect of Fuel Injection Timing on Performance Parameters

The prime objective of this research is to investigate the engine performance parameters (BSFC, BTHE and EGT) affected by the use of fuel and intake preheating at varying fuel ITs operating condition. The variations of engine performance parameters at different values of fuel ITs at (19°–27° bTDC) at CR of 18 and 212 bar fuel IP under a constant 90% load and 30% EGR rate for preheated POME biodiesels diesel blends, and CR of 17.5 and 200 bar fuel IP for diesel (PBD0) operating conditions are considered and is presented in **Figure 8.13** to **Figure 8.14**. Error bars indicate the standard deviation in each run.

8.3.3.1.1 Brake Specific Fuel Consumption (BSFC)

Figure 8.13(a) illustrates the variations of BSFC with fuel ITs for preheated POME biodiesel blends (PPBD20, PPBD40 and PPBD60) when compared to the neat diesel fuel (PBD0). It can be seen from figure that, overall, BSFC values reduced with variable fuel ITs at 90% load condition. Advancing fuel IT, at 27° bTDC, the BSFC of preheated POME biodiesel diesel blends was slightly lower than other fuel ITs. The BSFC of the PPBD20, PPBD40 and PPBD60 blends at the fuel IT of 27° bTDC are 0.22377 kg/kW h , 0.23189 kg/kW h and 0.24878 kg/kW.hr, respectively whereas for diesel (23° bTDC) it is 0.24884 kg/kW.hr. This may be due to higher viscosity and low volatility which causes better utilization of oxygen which influences engine combustion and emissions. It may be due to improved combustion in the fuel-rich zone due to the presence of oxygenate. Retarding the fuel IT by 4° from the standard 23° to 19° bTDC, the BSFC of PPBD20, PPBD40 and PPBD60 blends was increased by 17.17%, 20.69% and 25.03%, respectively. This increase in BSFC of all blends may be due to inadequate time availability for fuel vaporization and mixing with air in premixed combustion increased. As BSFC measures how efficiently an engine uses the supplied fuel to produce work. It is inversely proportional to the brake thermal efficiency of the engine. Therefore, lower value of BSFC indicates higher overall efficiency of an

engine (Rakopoulos et al., 2012). Hence, advancing the fuel IT helped for improvement of BSFC of the engine fueled with preheated POME biodiesel diesel blends.

8.3.3.1.2 Brake Thermal Efficiency (BTHE)

Figure 8.13(b) shows the variations of BTHE with respect to the fuel ITs for preheated POME biodiesel blends when compared to the neat diesel fuel. It can be seen from the figure that, overall, BTHE values improved by advancing IT at 90% load condition. At 27° bTDC gives highest BTE than all other fuel ITs when compared to that of neat diesel fuel (23° bTDC). The BTHE of the PPBD20, PPBD40 and PPBD60 blends at the IT of 27° bTDC is 33.64%, 32.67% and 31.74%, and almost equal to or higher than that of neat diesel fuel, PBD0 (23° bTDC) 31.87%. This may be due to combination of low volatility and mass flow rate which indicates inputs to the engine, which in case of preheated POME biodiesel diesel blends, are more compared to neat diesel. Retarded IT to 19° bTDC, diminished BTHE values for all the blends compared to base fuel IT due to incomplete combustion. Fuel injection timing (IT) plays crucial role in the start and quality of combustion. The optimal fuel IT based on BTHE for all preheated POME biodiesel diesel blends (PPBD20, PPBD40 and PPBD60) was 27 °CA BTDC and at 90% load condition, the base IT shows the best result for diesel (PBD0).

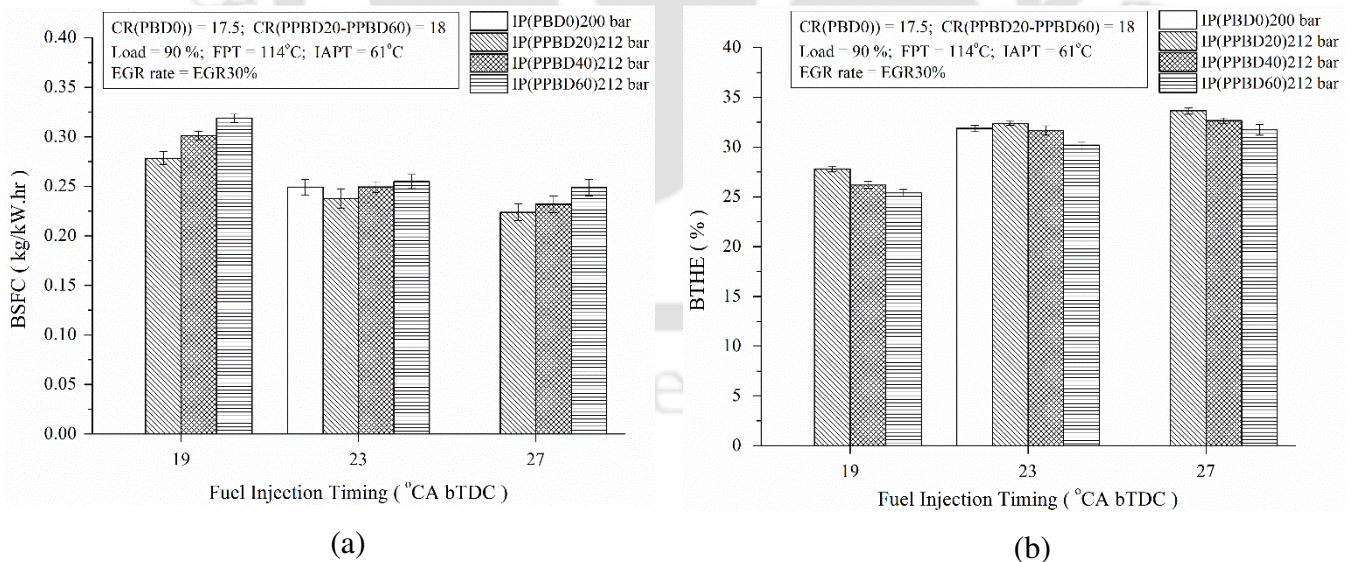


Figure 8.13: Variation of engine parameters with fuel ITs for fuel samples: (a) BSFC, (b) BTHE.

8.3.3.1.3 Exhaust Gas Temperature (EGT)

Figure 8.14 shows the variations of EGT with fuel injection timing for preheated POME biodiesel blends when compared to the neat diesel fuel at constant 90% of load and EGR30% rate. A slight increase in the exhaust temperature is noted with the advancement of IT by 4 °CA. This may be due to the fact that advancement of IT increases the ignition delay period resulting in higher in-cylinder pressure and temperature. As compared to diesel, EGT of palm oil methyl ester is always higher due to its inherent oxygen content leading to a more complete combustion. Hence, the maximum EGT is observed with PPBD20, PPBD40 and PPBD60 at 27° bTDC.

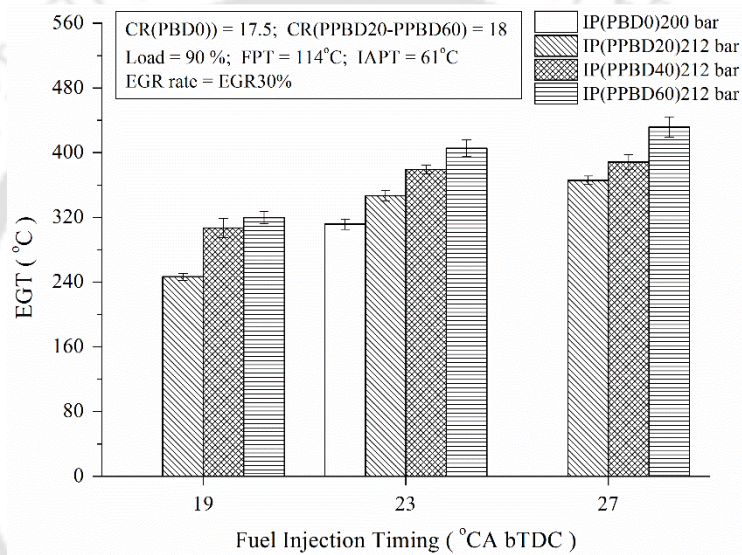


Figure 8.14: Variation of EGT with fuel ITs for fuel samples.

8.3.3.2 Effect of Fuel Injection Timing on Exhaust Emissions

Emission characteristics of the test engine fueled with preheated POME biodiesel with diesel blends and diesel fuel measured from emission analyzer have been discussed in terms of CO, CO₂, HC and NO_x, emissions. The variations of emission constituents at different values of fuel ITs at 19°, 23° and 27° bTDC at CR of 18 and 212 bar fuel IP with a constant rated load of 90% and EGR30% rate for preheated POME biodiesels diesel blends, and CR of 17.5 and 200 bar fuel IP for diesel (PBD0) operating conditions are considered and is presented in **Figure 8.15** to **Figure 8.16**. Error bars indicate the standard deviation in each run.

8.3.3.2.1 Carbon Monoxide (CO) Emission

The emission of CO from a direct injection diesel engine mainly depends on the physicochemical properties of the fuel. **Figure 8.15(a)** shows the variations of CO emission with respect to fuel ITs for PPBD20, PPBD40 and PPBD60 blends when compared to the neat diesel fuel (PBD0). Retarded IT by 4° (from 23° to 19° bTDC), shows a noticeable increase in CO emission, while advanced IT to 27° bTDC from standard 23° bTDC shows a reduction in CO emission. At 27° bTDC gives low CO emission than all other fuel ITs when compared to that of neat diesel fuel (23° bTDC). The CO emission of the PPBD20, PPBD40 and PPBD60 blends at the IT of 27° bTDC are 0.066523%, 0.05791% and 0.04312% whereas for diesel, PBD0 (23° bTDC) it is 0.119%. This may be due to oxygen concentration and cetane number. Since POME biodiesel contains oxygen it acts as a lesser combustion promoter inside the cylinder.

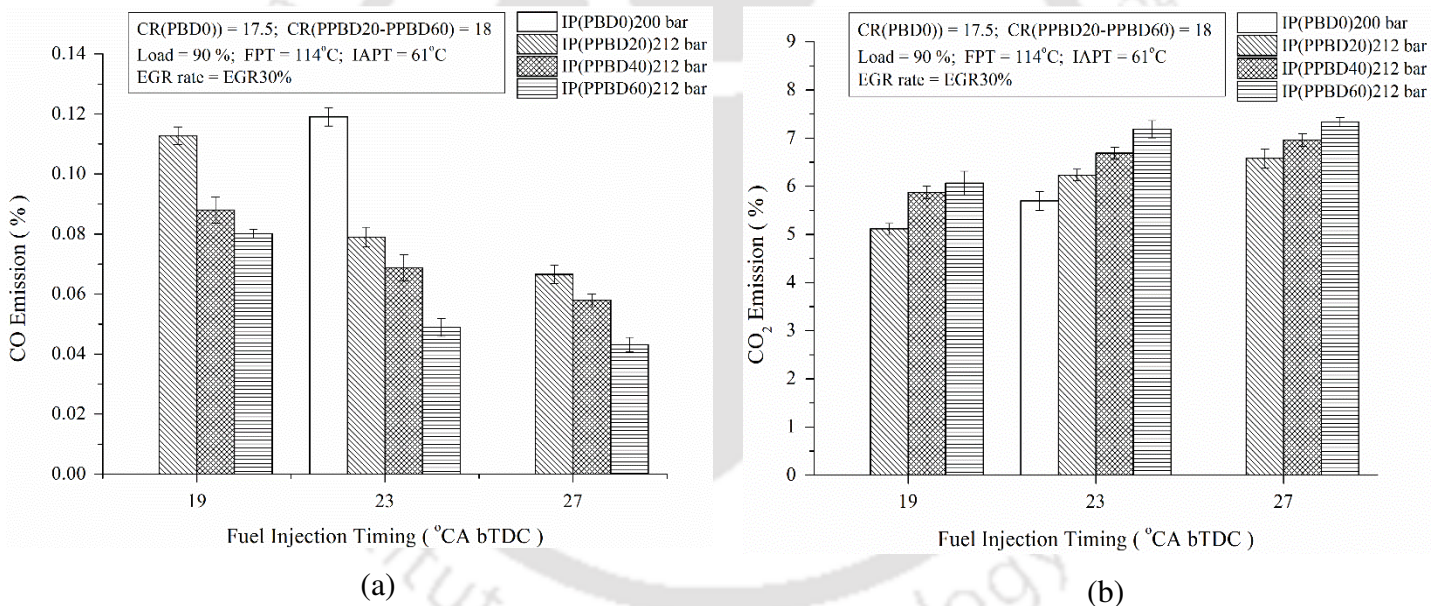


Figure 8.15: Variation of exhaust emission with fuel ITs for fuel samples: (a) CO, (b) CO₂.

8.3.3.2.2 Carbon Dioxide (CO₂) Emission

The effect of fuel ITs on CO₂ emission of preheated POME biodiesel diesel blends at 90% of load under 30% EGR rate is exposed in **Figure 8.15(b)**. It has opposite trends to that of CO emission. Advancing the fuel IT from the standard increased the level of CO₂ emission of preheated POME biodiesel blends in the exhaust emission. Advancing fuel IT by 4° (from standard 23° to 27°

bTDC), slightly increased the CO₂ emission for PPBD20, PPBD40 and PPBD60 by 5.58%, 4.08% and 2.07%, respectively as compared diesel of PBD0 at base fuel IT (23° bTDC). Whereas retarding fuel IT by 4° (from the standard 23° to 19° BTDC), reduced CO₂ emission by 17.96%, 12.29% and 15.54%, respectively. This is because, advancing IT, caused to have early combustion in the premixed combustion phase, which enhanced combustion process, in-cylinder gas pressure, and temperature thereby, augmenting the complete combustion with significantly drop in by-product CO emission level. The reverse is true for retarding fuel IT operating condition for all blends.

8.3.3.2.3 *Unburnt Hydrocarbon (HC) Emission*

The variation in unburned hydrocarbon (HC) emission versus variable fuel IT of preheated POME biodiesel blends when compared to that of neat diesel fuel at 90% rated engine load under EGR30% rate are displayed in **Figure 8.16(a)**. As illustrated in the figure, advanced IT for optimum blend fuel shows reduced HC emission. Overall, at 90% load condition in DI diesel engine, HC emission was low and varied modestly with fuel IT. The general trend of HC emission reduction was observed in the results with advanced IT. Higher amount of HC emission was evident between the base injection timing with retarded IT. At 27° bTDC gives low HC emission than all other fuel ITs when compared to that of neat diesel fuel. The HC emission of the PPBD20, PPBD40 and PPBD60 blends at the IT of 27° bTDC are 21 ppm, 16 ppm and 11 ppm whereas for diesel of PBD0 at base fuel IT of 23° bTDC it is 39 ppm. This may be due to viscosity and surface tension that affect penetration rate and droplet size of fuel, which in turn affect mixing of fuel and air. Cetane number of fuel also plays a vital role in ignition process. Preheating intake air caused to significantly the HC emission in exhaust gases. According to these figures, it was revealed that 27° bTDC was the best IT based on HC emission.

8.3.3.2.4 *Oxides of Nitrogen (NOx) Emission*

In direct injection diesel engine, variable fuel ITs has a strong effect on NO_x emissions (Sathiyamoorthi and Sankaranarayanan, 2015). The various factors like availability of oxygen, combustion temperature, nitrogen content of the fuel itself and the reaction time determine NO_x emission during the combustion process. The production of NO_x is proportional to combustion efficiency. Better combustion efficiency has higher value of exhaust temperature, which increases

the level of NO_x production (Palash et al., 2014). Most of the techniques to reduce NO_x emission depend on the reduction in peak flame temperature. The peak flame temperature can be reduced by various methods like retarded IT, exhaust gas recirculation (EGR) into the engine's intake manifold and use of biodiesel alternative fuels (Anand and Mahalakshmi, 2007; Subramanian, 2011). The variation in NO_x emissions versus fuel ITs of preheated POME biodiesel diesel blend when compared to the neat diesel fuel at 90% of load and EGR30% of EGR rate is displayed in **Figure 8.16(b)**. As illustrated in the figure, the advancement of IT for optimum blend fuel shows increased NO_x emission. In advanced IT, the fuel is injected into compressed air having lower temperature. It extends the ignition delay period. Longer ignition delay period causes more fuel to be injected during that period, and better mixing of fuel and air. This means that, the earlier fuel is injected, the longer ignition delay and the greater the premixed portion of the fuel prior to ignition leads to higher NO_x formation (Heywood, 1988; Sathiyamoorthi and Sankaranarayanan, 2015). Retarded IT shows the opposite effect of a sharp decrease in NO_x emission as expected due to reduced peak pressure and temperature. At 27° bTDC gives low NO_x emission than all other ITs and compared to that of neat diesel fuel of PBD0 (23° bTDC). The NO_x emission of the PPBD20, PPBD40 and PPBD60 blends at the injection timing of 27° bTDC are 151 ppm, 167 ppm and 183 ppm whereas for diesel (PBD0) at base fuel IT, it is 124 ppm. This may be due to the presence in biodiesel of oxygen, which leads to complete combustion of biodiesel than diesel. As a result, maximum temperature inside cylinder is more in case of biodiesel than diesel. Preheating intake air caused to higher NO_x emission of all bends of fuel, however implementations of EGR in a diesel engine, and recirculating exhaust gas EGR30% plays vital role for reductions of NO_x emission at 90 % loading engine operating conditions. Overall, the optimum IT based on NO_x emission was 19° bTDC for PPBD20, PPBD40 and PPBD60 optimum blends.

In general, based on the experimental studies of effect of fuel ITs, the following are the observations and summaries: At 90% load condition, the BSFC of base IT was slightly higher than other ITs, it reduced compared to other ITs and gave the best result for PPBD20, PPBD40 and PPBD60 blends for advanced fuel IT to 27° bTDC. BTHE values improved by advancing IT. Retarded IT diminished the BTHE values for all the blends of fuel compared to the base IT (23° bTDC). Advanced IT reduced CO, HC, while retarded IT increased. Higher amount of CO and HC emission with retarded IT than the base IT was evident. The optimum based on CO, HC at 90% load condition was 27° bTDC. CO₂ emissions increased compared to the base injection timing

over all ITs. The advancement of IT increased NO_x emission, while the retarded IT showed the opposite effect with a sharp decrease in NO_x emissions. In summary, on the basis of the above observations, it can be summarized that advanced IT at 27° bTDC reduced CO and HC emissions, and the retarded IT at 19° bTDC reduced BTHE and NO_x emissions.

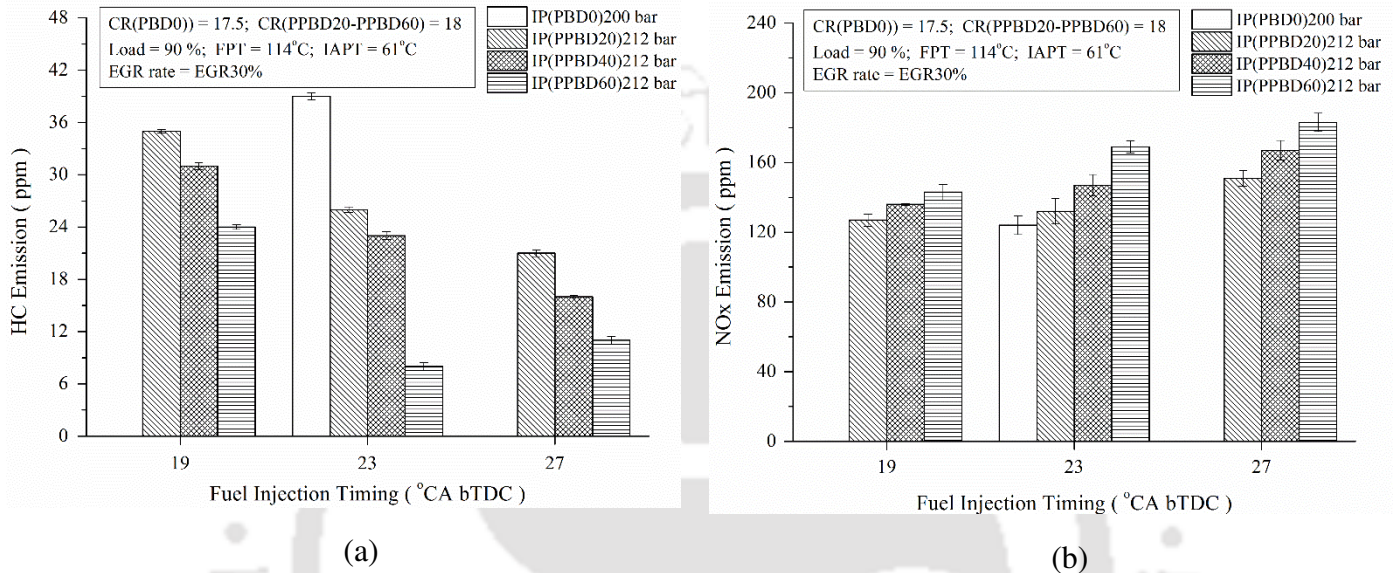


Figure 8.16: Variation of exhaust emissions with fuel ITs for fuel samples: (a) HC, (b) NO_x.

8.4 Uncertainty Analysis

In experiments, uncertainty analysis has to be performed due to error caused by instrument's inaccuracy, in-apt calibration, human inadequacy etc. The sequential perturbation technique (Kline and McClintock, 1953) is used to compute the uncertainties of the parameters (Appendix - D). Some of these are air flow rate (1.1%), liquid fuel flow rate (0.1%), engine load (0.1%), engine speed (1.3%), LHV of liquid fuel (1.0%) etc. Based on these the calculated accuracy of the performance and combustion study (at CR 18, 212 bar fuel IP with base fuel IT of 23° bTDC) for both PPBD60 run engine are found to be within $\pm 4.34\%$. However, the accuracy of the emission study is found as $\pm 2.3\%$.

8.5 Summary

The optimum preheated POME biodiesel-diesel blend PPBD20, PPBD40 and PPBD60 were tested as fuels in an unmodified CI engine for varying CRs, fuel IPs and fuel ITs to evaluate its effect

and to determine the optimum CR, fuel IP and fuel IT. Engine performance, combustion and emissions characteristics were influenced by varied CR, high fuel IP, and the advancement or retardation of IT for both the fuels. The engine characteristic shows improved result at higher CR, high IP and advanced IT without dependency on types of engine. BTHE and BSFC can be improved along with a reduction in emissions except for NO_x emission at higher CR (CR18). This is due to the better combustion and air-fuel mixing. The maximum reduction for CO and HC emissions were found in this condition due to higher oxygen content and better combustion. Similarly, positive results were found at high IP. There is a reduction in CO and HC emission along with an increase in BTHE at high IP (212 bar). This is due to a reduction in drop size of fuel at high IP and better mixing with air to make a homogeneous mixture. Mixed results came out at varying injection timing. The engine characteristics can be improved by advancing, retarding or at the same IT depending on the condition of engine working at 90 % of engine load. For advance IT (27° bTDC), engine performance (BSF, BTHE and EGT) and emission characteristics (CO, CO₂ and HC) of engine shows enhanced results because of the longer ID at advanced IT; inject the more fuel which promotes premixed combustion. Also, reduction in NO_x emission is found at retarded IT (19° bTDC) due to decrease in the in-cylinder temperature inside the engine cylinder combustion chamber because of late combustion.

- The percentage of maximum BTHE improved at higher CR (18) is 31.97%, 30.18% and 28.99% for PPBD20, PPBD40 and PPBD60 blends as compared to diesel (31.97%) at CR 17.5. Also, BSFC reduces by 4.06%, 9.13% and 12.88% at the same condition. This is due to the better combustion and air-fuel mixing. The maximum reduction in CO was found by 25.02%, 38.25% and 57.44% while HC emissions was decreased by 25.64%, 33.33% and 69.23%, respectively for PPBD20, PPBD40 and PPBD60 whereas, increased in NO_x emission was seen by 39.52%, 66.94% and 84.68%, respectively due to higher oxygen content and better combustion.
- The increment by 1.58% in BTHE for PPBD20, whereas slightly decreased by 0.62% and 5.26%, respectively for PPBD40 and PPBD60 at IP 212 and CR18. And, 4.49 % reduction in BSFC for PPBD20, and slightly increased by 0.25% and 2.42% for PPBD40 and PPBD60 as compared to diesel (CR17.5 with 200 bar IP). The CO and HC emission reduces in maximum percentage by (33.65%, 42.23% and 58.89%) and (33.33%, 41.03% and 79.49%), respectively. The NO_x was increased by 58.06%, 79.84% and 100.81%, respectively for PPBD20, PPBD40

and PPBD60 at IP 212 and CR18 as compared to diesel at base (IP 200 and CR17.5). This is due to the reduction in drop size of fuel at high IP and better mixing with air to make a homogeneous mixture.

- For the advancement in IT, BTHE was found to be increased by 5.55% and 2.49% for PPBD20 and PPBD40, while slightly decreased by 0.4% for PPBD60 at 27° bTDC as compared to diesel at 23° bTDC. Meaningfully decreased in BSFC by (10.1%, 8.81% and 0.02%) for PPBD20, PPBD40 and PPBD60 blends at same IT. Whereas, EGT was found to be decreased by (20.84% and 1.55%) for PPBD20 and PPBD40, but for PPBD60 slightly increased by 2.59% at same IT. The CO emission was found to be reduced by 44.1%, 51.34% and 63.76%, respectively while HC is decreased by (46.15%, 38.97% and 71.79%), respectively for PPBD20, PPBD40 and PPBD60 by as compared to diesel (IT=23° bTDC). The CO₂ emission (at 27° bTDC) is increased by 15.55%, 22.23% and 28.76%, and the NO_x emission is higher by (83.06%, 91.13% and 110.48%), respectively for the PPBD20, PPBD40 and PPBD60 blends. This is due to the longer ID at advanced IT; inject the more fuel which promotes premixed combustion. Marginally reduced in NO_x emission (retarded IT at 19° bTDC) was found by 10.48%, 14.52% and 41.13%, respectively for PPBD20, PPBD40 and PPBD60 as compared to advanced IT 27° bTDC. This may be due to decrease the temperature in the cylinder.

From the above observation, the best results of performance, combustion and emission characteristics diesel engine can be found to be optimum when using the preheated biodiesel blends up to 60% (PPBD20–PPBD60) as a fuel. The corresponding optimum parameters are as follows: high injection pressure (212 bar), high compressor ratio (CR18) and varying IT (advancing 27° bTDC/retarding 19° bTDC) at 90% engine load and EGR 30% rate operating condition.

CHAPTER-9

Thermodynamic and Thermo-economic Analysis of Preheated Blend Ratios

Overview

The theoretical route towards the effective distribution of energy at various components of thermal energy system is done by coupling the first and the second laws of thermodynamics. The thermodynamic analysis (first and second law) as a technique, is getting attracted day by day to understand and control the thermodynamic energy distribution and maximum possible performance of the thermal energy system especially IC engines from a small scale range to robust entity. Meanwhile, this type of analysis locates and estimates the energy and exergy distribution, destruction and directs people to find strategies for better available energy management. A complete thermodynamics analysis of engine helps to acquire a most convincing picture of engine behavior fuelled at different test fuels. These analyses help economic situation, to evaluate the performance of different test fuels for direct application in a compression ignition engine. However, the thermodynamic analysis of a diesel engine run on preheated blended biodiesel is rare in open literature. In order to establish the preheated blended biodiesel as an alternative to diesel, it is essential to reveal its effects on thermomechanical and thermo-economic exergy distribution, while running in a diesel engine. It is observed that, a preheated blended biodiesel has close fuel properties with diesel. Hence, the energy and availability distribution of different blend ratios of preheated biodiesel run engine will not be identical to that of a diesel engine run at standard settings. To enlighten this ambiguous fact of literature, this chapter provides the first and second law analysis of a compression ignition engine running in different engine load condition with biodiesel, preheated biodiesel and preheated blended biodiesel. The energy analysis is performed applying the first law of thermodynamics. It includes the effect on energy distribution (shaft, cooling water, exhaust gas, and uncounted), destruction, cost rates of product exergy and cost unit exergy unit for a set of standard operating conditions of CR, IP and IT. The exergy analysis covers the results studied in view of second law. The effect of variation of engine load on the fuel exergy distribution (shaft, cooling water, and exhaust gas), destruction, second law efficiency and entropy generation are discussed, sequentially.

9.1 Preface

In the present study, two important thermodynamics laws are applied to examine the energy potential (both quality and quantity) in a compression ignition engine using mineral diesel and five preheated POME biodiesel/diesel blends of fuel. It may be emphasized that complete conversion of input energy from fuel to shaft power of the engine is not possible because it would violate the second law of Thermodynamics (Boles and Cengel, 2014; Tat, 2011). For this purpose, the laws of thermodynamics (first and second) are generally used for analyzing the engine. The first law of thermodynamics provides information about fuel inlet energy converted into different energy rates carried by cooling water, exhaust gases and useful brake power. In order to access a most efficient performance of fuels in terms of engine performance, second law analysis (exergy) has to be employed to know the quality of energy (Sorathia and Yadav, 2012). It is used to realize actual efficiencies of engine from exergy balance and locating major losses in the components where they occur (da Costa et al., 2012; López et al., 2014). A complete thermodynamics analysis of engine helps to acquire a most convincing picture of engine behavior fuelled at different test fuels. These analyses help economic situation, to evaluate the performance of different test fuels for direct application in a compression ignition engine and there are limited reported literatures. In this regards, a complete analysis has been carried out to study the thermodynamic and economic analysis for preheated blends of biodiesel. **Table 9.1** presents the fuel properties of different fuel used. The thermodynamics analysis is carried out based on 1st and 2nd laws of thermodynamics by considering control volume analysis as shown in **Figure 9.1**. The exergy and economic balance the control volume of CI engine is shown in **Figure 9.2**. The relevant data for details of thermos-economic analysis for different blended fuels have been given in **Table 9.2**.

The study is aimed at investigation of the thermodynamic and thermos-economic potentials of different blends of fuel used in a CI engine at varying loading conditions. To this aim, the experiments were carried out using diesel (PBD0), preheated POME biodiesel/diesel blends (PPBD20–PPBD100) with 20% increments with variation of engine loads (low load– 10%, medium load– 50% and high load– 90%) at constant speed (1500 rpm) with default CR 17.5, 200 bar and 23° bTDC of fuel IP and IT, and are shown in test matrix **Table 3.3**. The experiments are conducted in the same experimental setup as used in Chapter-6.

For the current investigation, neat palm oil methyl ester “POME” (PBD100) has been preheated at elevated temperature (114 °C) prior to blending with diesel for performance in a CI engine partly or fully. The engine power, fuel consumption, exhaust gas temperature, outlet water temperature and exhaust emissions were measured and the equation of the combustion was determined using experimental data. Finally, the thermo-economic analysis was performed using energy, exergy and economic balances.

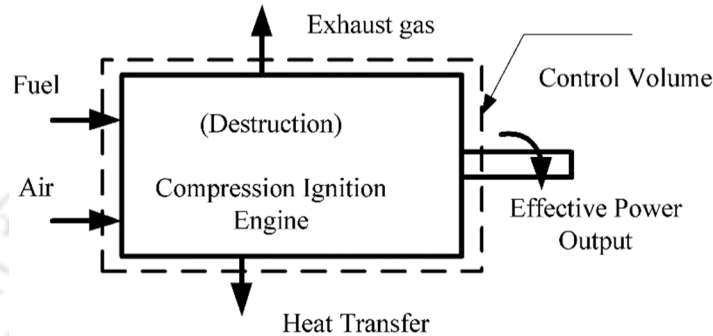


Figure 9.1: Control volume of the CI engine.

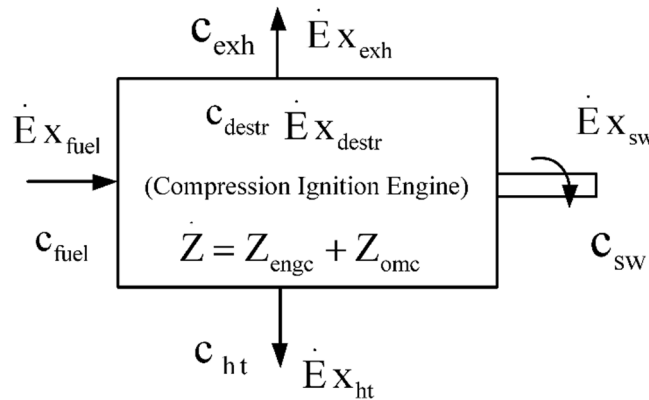


Figure 9.2: The exergy and cost rate stream of the CI engine.

9.2 Analysis Methodology

The sequence of events happening in the engine operation can be identified as fuel and air entrainment, combustion, conversion of chemical energy into mechanical work, heat loss through cooling water, friction, radiation, surroundings and exhaust gas (Al-Najem and Diab, 1992). Combustion analysis begins with the measurement of exhaust gas concentrations, cooling water temperature and gas temperature. The molecular formulas of pure diesel fuel and neat POME

biodiesel are approximated as $C_{12}H_{26}S_{0.024}$ and $C_{18.07}H_{34.93}O_2$, respectively (Debnath et al., 2013b). Further it is assumed that the combustion air and exhaust gases are ideal gas mixtures and their potential and kinetic energy changes are minor (Sayin et al., 2007). Dry air which is assumed to contain 21% oxygen and 79% nitrogen enters the engine at reference atmospheric conditions are considered as 1 atm and 27 °C of pressure (P_{amb}) and temperature (T_{amb}). The detailed calculations used for this analysis are given in Appendix B.

Table 9.1: Important properties of various blend fuels calculated based on ASTM D6751 standards.

Fuel	Diesel (PBD0)	POME (PBD100)	Blends of fuel (Biodiesel preheated at 114 °C)				
			PPBD20	PPBD40	PPBD60	PPBD80	PPBD100
Kinematic viscosity [§] , (mm ² /s)	3.23 (40 °C)	8.71 (40 °C)	3.38	3.76	4.08	4.86	5.37
Density [§] , (kg/m ³)	842 (27 °C)	946 (27 °C)	851	859	863	869	876
Pour point [§] , (°C)	-8	5	-1	2	4	6	8
Flash point [§] , (°C)	63	265	103	141	181	226	289
Lower heating value [§] , (MJ/kg)	44.69	38.29	43.12	42.32	41.67	40.878	39.82
Molecular formula [§]	*C ₁₂ H ₂₆ S _{0.0024}	*C _{18.07} H _{34.93} O ₂	C _{14.32} H _{26.96} O _{0.35}	C _{15.46} H _{28.3} O _{0.85}	C _{16.13} H _{30.43} O _{1.32}	C _{17.28} H _{32.94} O _{1.99}	C _{18.07} H _{34.93} O ₂
\mathcal{E}_{fuel} # (eqn. B14)	1.07129	–	1.06758	1.06952	1.07137	1.07351	1.07691

[§]Measured value; #Calculated value; * reference (Debnath et al., 2013b).

Table 9.2: Comparative test fuels and investment cost estimations, and exergy rates (at 90% of load) for thermos-economic analysis.

Fuel	Cost per liter of fuel (\$/Liter)	Investment cost		Exergy rates (kW)		
		Z _{engc} (\$/hr)	Z _{omc} (\$/hr)	$\dot{E}x_{fuel}$ (MJ/hr)	$\dot{E}x_{sh}$ (MJ/hr)	$\dot{E}x_{destr}$ (MJ/hr)
PBD0	1.029757	40.652	11.108	27.469	0.03126	0.02726
PPBD20	1.528583	42.510	11.144	29.202	0.03126	0.02726
PPBD40	2.027408	43.494	11.031	30.327	0.03126	0.02726
PPBD60	2.526233	44.561	11.012	31.388	0.03126	0.02726
PPBD80	3.025059	45.201	10.946	32.052	0.03126	0.02726
PPBD100	3.523884	45.929	10.887	32.815	0.03126	0.02726

9.3 Results and Discussion

The results of this study comprised three main parts: energy, exergy and economic results. Initially the effect of preheated POME biodiesel/diesel blends on heating values and brake specific fuel consumption were investigated. Then the energy analysis results consist of various energy rates (input energy rate, useful work energy rate, energy rate of cooling water, energy rate of exhaust gas

and uncounted energy rate) and first law efficiency (energetic efficiency), while the results of exergy (second law) analysis are various exergy rates, exergetic efficiency, the rate of exergy destruction and entropy generation, sustainability index and, finally, cost rates of fuel, power, losses and exergy destruction for different blends of biodiesel are reported as the economic results.

9.3.1 Effect of Preheating and Blend Ratios on LHVs and BSFCs

The heating value of a fuel is the amount heat released by burning a definite amount of fuel in an engine after the latent heat of vaporization. It is the energy input in the engine and are measured using bomb calorimeter. The biodiesel has lower calorific value compared to neat diesel fuel and preheating marginally improved the lower heating values (LHVs) of neat POME (PBD100) from 38.29 to 39.82 MJ/kg (**Table 6.1**). In addition to preheating PBD100, blending with diesel fuel benefit to boost the LHVs of different blends of fuel. As seen in **Table 9.1**, the highest LHV is attained for PBD0, while PPBD100 has least LHV. The result implies that increase of the volume of fraction of biodiesel in the blend ratio decrease the LHVs of blends of fuel.

The variations in brake specific fuel consumption (BSFC) for the all blends of fuel with varying engine loads is presented in **Figure 9.3**. The BSFC of PPBD100 fuel was found maximum for varied engine loads. This is because of the relatively high viscosity and poor mixture formation of PPBD100 even after preheated. The BSFCs of the blends of fuel are in order from minimum to maximum (PBD0 < PPBD20 < PPBD40 < PPBD60 < PPBD80 < PPBD100) for every engine loads. The BSFC is inversely proportional to the percentage fractions of preheated biodiesel. The significant reduction on BSFC values are seen up to medium (50%) engine load. Beyond this point, there is a slight variation among all test fuels. At low load (10%), the amount fuel injected into engine combustion chamber is higher whereas the amount of intake air in engine cylinder is lower. This leads to a richer mixture ratio and incomplete combustion. The trends of BSFC is seen to decrease with increase of load. This is mostly attributable to the improvement of combustion efficiency of fuel with increase of suction of more intake air into the engine cylinder, which it can produce the same brake power compared PBD0.

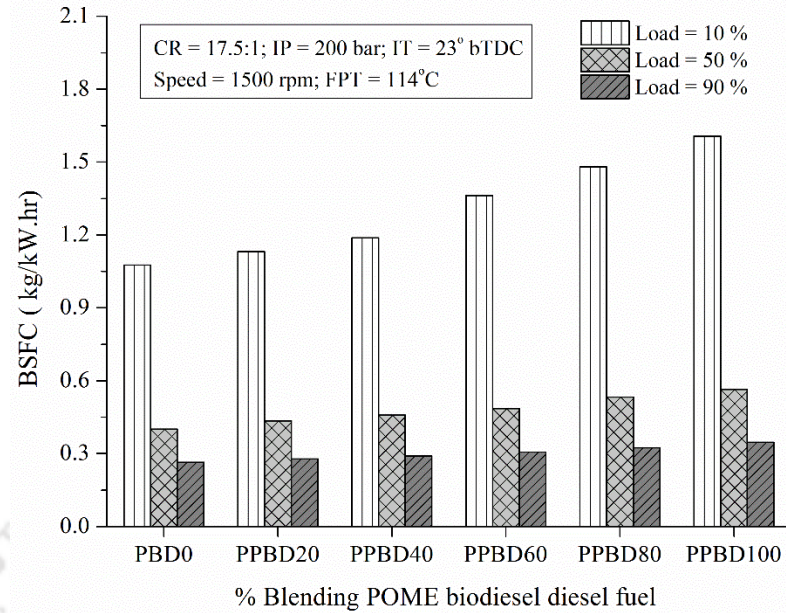


Figure 9.3: The variations of BSFC with blends of fuel.

9.3.2 Energy Analysis

The energy distributions for a CI engine operated with different blends of fuel at varied engine loads are tabulated in **Table 9.3**. It may be noticed that the standard deviation among all fuel energy input, $\dot{H}_{e_{in}}$ lies between 0.024–0.053 and drop in average value is very negligible (<0.038). At 90% of load, the mean input energy per unit time for all blends of fuel of the studied are 11.34 kW. A portion $\dot{H}_{e_{in}}$ (only fuel energy is considered, $\dot{H}_{e_{fuel}}$) is changed into useful brake power for different blends fuel was 3.06 kW. It is seen in **Table 9.3** that, all energy rates except the useful brake power, are directly proportional to the blends of fuels. At 90% of load, the useful brake power slightly decreases (as much as 2%) when using PPBD100 as fuel compared to PBD0. This is because of the lower heating value and relatively high kinematic viscosity of biodiesel. It is seen in **Figure 9.4(a)** that, the amount of input fuel energy rates increased with increase of the volume fraction of biodiesel in blend ratio. It mostly depends on the mass flow rate fuel for a given time interval. When percentage of biodiesel in the blends is high, engine consumed more fuel because of increase in the volume fractions of biodiesel caused the quality of the blend fuel degrades resulting engine to consume high volume of fuel to produce similar power output as compared with a low content of biodiesel in the blended mixture for fixed running time. Thus, the input fuel energy rate varies from maximum to minimum as, PPBD100 > PPBD80 > PPBD60 > PPBD40 >

PPBD20 > PBD0. Since, the BSFC is directly proportional to the blends of fuel (**Figure 9.3**), the input fuel energy rate is maximum at higher blends ratios.

Increasing the percentage fraction of biodiesel in the blend ratios causes a drop in generating useful shaft work converted from the input energy rate. Because of the inferior quality, fuel atomization and mixing rate, results more in energy loss, due to high viscosity and density of higher blend ratio. However, exhaust gases and uncounted energy losses were found higher for PPBD100. This means more exhaust gases energy or energy loss in the engine for a high level blends of fuel. Hence, low blends of fuel only consider a better option. The energy rates taken away by cooling water heat transfer for all blends of fuel almost identical and the variations of the energy rates are negligible even with loads (**Table 9.3**).

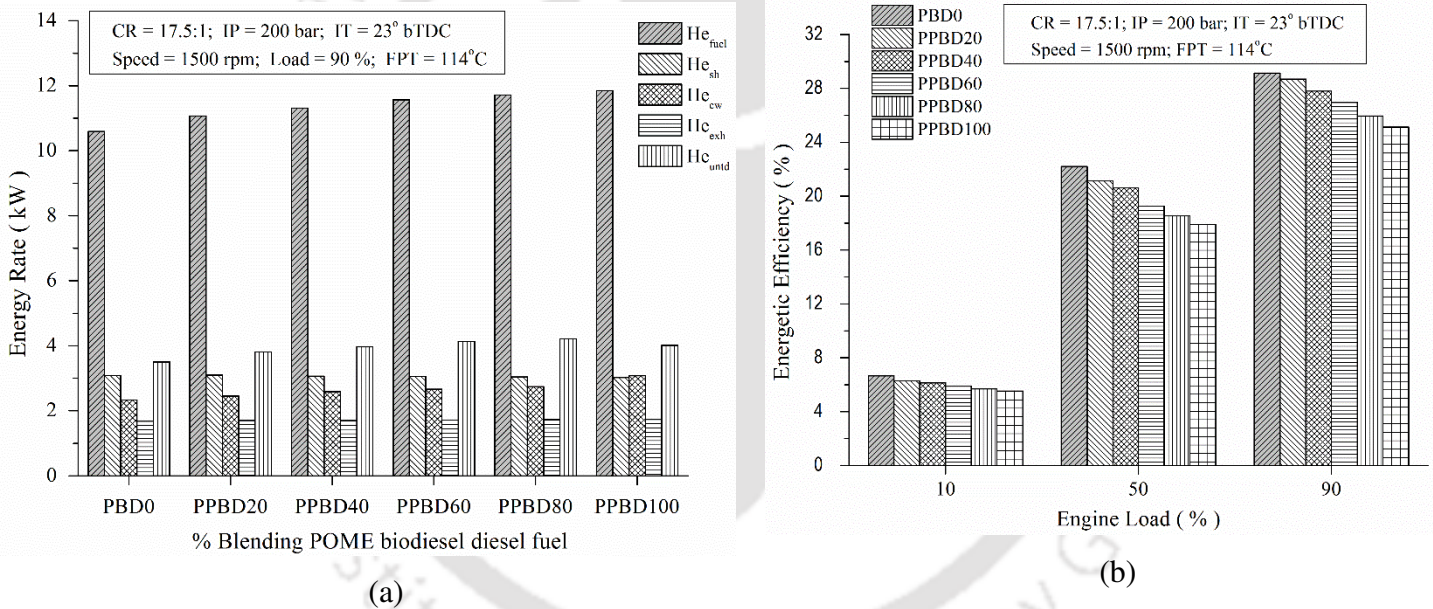


Figure 9.4: The energy rates and energetic efficiency of CI engine for different blends of fuels: (a) energy rates; (b) energetic efficiency.

Furthermore, the CI engine has a better energetic efficiency when operated at low blend ratios for every loads. “Energetic efficiency” is the means of effective conversion of fuel energy input to useful brake power in the engine. As can be seen from the **Figure 9.4(b)**, more efficient conversion occurs in engine with neat diesel (PBD0) usage and gave a relatively higher energetic efficiency whereas neat preheated biodiesel (PPBD100) fuel was found least for every engine loads. The lower calorific value of the POME is the main reason for this fact. However, the

combined effects preheating and blending POME with diesel fuel has helped to increase the LHVs of POME. The relative values of each test fuels vary with one another and depends of the percentages fraction of biodiesel. At 90% of load, the energetic efficiencies of all blends of fuel change from 25.5% to 29 %. The energetic efficiency of the engine for fuels from lower to higher are: PPBD100 < PPBD80 < PPBD60 < PPBD40 < PPBD20 < PBD0.

9.3.3 Exergy Analysis

The findings of exergy rates for CI engine operated with blends of fuel are presented in **Table 9.4**. It may be noticed that the standard deviation among all exergy input rate, $\dot{E}x_{in}$ lies between 0.026–0.057 and drop in average value is very negligible (<0.04). At 90 % of load, the mean $\dot{E}x_{in}$ for all blends of fuel studied are 12.16 kW. This can also be termed as total input exergy. Since input exergy through air for combustion is neglected, a portion of this exergy rate is changed into useful brake power for different blends fuel was 3.06 kW. Some amount of exergy is flown through engine cooling water and exhaust gas. The rest amount of exergy has been lost due to friction radiation, heat transfer to surroundings. At 90 % engine load, the fuel exergy rate has a same fashion with fuel energy rate (**Figure 9.4-a**), and the fuel input exergy rates for all blends of fuel are 7.12 to 7.69 % times higher than the fuel input energy rates. The chemical exergy factor, also influences this rate, which makes the fuel exergy rate higher as compared with fuel energy rate. At same load, the input fuel exergy of neat diesel fuel (PBD0) is 12.44% lower than that of preheated POME (PPBD100) due to a less fuel consumption per unit time. The exergy rates of $\dot{E}x_{in}$, $\dot{E}x_{sh}$, $\dot{E}x_{exh}$ and $\dot{E}x_{dstr}$ are increased at higher engine loads for all blends of fuel. It is also seen in **Figure 9.5(a)** that, all exergy rates (except useful shaft work) are directly proportional to the blends of fuels. The input fuel exergy rate from higher to lower are as follows: PPBD100 > PPBD80 > PPBD60 > PPBD40 > PPBD20 > PBD0. The chemical exergy factors and fuel consumption for each fuel play significant role on input fuel exergy. The useful brake exergy rate remains almost the same with the useful energy rates for all fuels. The exergy rate of useful work for all blends of fuel drops with increase in percentage fractions of biodiesel at every loads (**Table 9.4**). This is because of the lower LHVs and higher viscosity of biodiesel with increasing the content of biodiesel in the blends. For all blends of fuel, the calculation indicates the conversion about 23.7–27.2% of the chemical exergy input into useful shaft work in the engine operated at 90 % of load (**Figure 9.5-a**). Therefore, approximately 30% of input exergy is found which can be called as available energy

from the thermodynamics viewpoint. As a result, 70% of input exergy is destroyed from the system.

As seen in **Table 9.4**, the exergy rates of exhaust gas marginally increased with increase percentage fractions of biodiesel at every engine load because of increased input exergy entering the engine cylinder. The change of exergy rate of exhaust gas loss for blends of fuel are initiated from variances of the exhaust temperatures. The exergy rate of exhaust gas loss for all blends of fuel are higher than diesel fuel (**Figure 9.5-a**). This is because of the fact that biodiesels offer maximum exhaust gas temperatures. It is seen in **Table 9.4** that; the exergy rate of cooling water marginally rises for greater volume fraction of biodiesel. At engine load of 90%, the calculation indicates the loss of 13.35% to 13.9% of the chemical exergy input in the cooling water and exhaust gases respectively for different blends of fuel.

On the other hand, the rate of exergy destruction has shown a lowering trend (**Figure 9.6-a**). The exergy destruction rate for all tested fuel decrease with increase of loads. This is can be due to excess air at higher loads and sufficient time for heat transfer within the combustion chamber. The increase trends of exergy rate of useful work coupled with diminishing of fuel exergy rate may be also the probable reason of the reduction of exergy destruction at the circumstance of slight changed cooling water and exhaust exergy rate for load variation. At 90% of load operation, for all blends of fuel, the calculation indicates the destruction of 59.46–62.4% of the chemical exergy input due to the irreversibility from the engine.

The exergetic efficiency for different blends of fuel with engine loads, shown in **Figure 9.5(b)** indicates the same trend as the energetic efficiency. At 90 % of load, the exergetic efficiency is nearly 6.65%, 6.33%, 6.50%, 6.66%, 6.85% and 7.14% lower than energetic efficiency of test fuels, PBD0, PPBD20, PPBD40, PPBD60, PPBD80, and PPBD100, respectively. It should be remembered that the inlet fuel energy ($\dot{H}_{e_{in}}$) is 6.75 to 7.69% lower than $\dot{E}_{x_{in}}$. It causes the difference between two efficiencies for the same test fuels and operating conditions. At 90 % of engine load, the exergetic efficiencies for all blends of fuel lies between 23.7 % and 27.2%. The exergetic efficiency of the engine considering all blends of fuel from higher to lower are as follows: PBD0 > PPBD20 > PPBD40 > PPBD60 > PPBD80 > PPBD100. The increase percentage fractions biodiesel in blend ratio caused to gradually increase kinematic viscosity, which prevents a formation of better air–fuel mixture. This tends to deteriorate the combustion efficiency and decrease the exergetic efficiency (η_{exerg}).

Table 9.3: Energy rate analysis of test fuels.

Fuel	Load (%)	Energy rates (kW)				
		$\dot{H}_{e_{fuel}}$	$\dot{H}_{e_{sh}}$	$\dot{H}_{e_{cw}}$	$\dot{H}_{e_{exh}}$	$\dot{H}_{e_{untid}}$
PBD0	10	5.42±(0.024)	0.37±(0.00)	0.39±(0.008)	0.67±(0.02)	3.99±(0.10)
	50	7.83±(0.035)	1.74±(0.01)	0.93±(0.019)	1.09±(0.033)	4.07±(0.102)
	90	10.59±(0.04)	3.08±(0.01)	2.34±(0.047)	1.68±(0.05)	3.49±(0.087)
PPBD20	10	5.70±(0.026)	0.37±(0.00)	0.47±(0.009)	0.69±(0.021)	4.17±(0.104)
	50	8.31±(0.037)	1.76±(0.01)	1.28±(0.026)	1.13±(0.034)	4.14±(0.104)
	90	11.05±(0.05)	3.09±(0.01)	2.45±(0.049)	1.70±(0.051)	3.81±(0.095)
PPBD40	10	5.79±(0.026)	0.36±(0.00)	0.34±(0.007)	0.73±(0.022)	4.36±(0.109)
	50	8.46±(0.038)	1.75±(0.01)	0.91±(0.018)	1.10±(0.033)	4.70±(0.118)
	90	11.30±(0.051)	3.07±(0.01)	2.58±(0.052)	1.69±(0.051)	3.96±(0.099)
PPBD60	10	6.08±(0.027)	0.36±(0.00)	0.36±(0.007)	0.75±(0.023)	4.61±(0.115)
	50	8.88±(0.04)	1.71±(0.01)	1.12±(0.022)	1.13±(0.034)	4.92±(0.123)
	90	11.56±(0.052)	3.06±(0.01)	2.66±(0.053)	1.71±(0.051)	4.13±(0.103)
PPBD80	10	6.22±(0.028)	0.35±(0.00)	0.38±(0.008)	0.75±(0.023)	4.74±(0.119)
	50	9.19±(0.041)	1.70±(0.01)	1.30±(0.026)	1.13±(0.034)	5.06±(0.127)
	90	11.71±(0.053)	3.04±(0.01)	2.75±(0.055)	1.72±(0.052)	4.20±(0.105)
PPBD100	10	6.34±(0.029)	0.35±(0.00)	0.37±(0.007)	0.77±(0.023)	4.86±(0.122)
	50	9.48±(0.043)	1.69±(0.01)	1.53±(0.031)	1.17±(0.035)	5.09±(0.127)
	90	11.85±(0.053)	3.02±(0.01)	3.09±(0.062)	1.73±(0.052)	4.01±(0.100)

There is an opposite trend between exergy destruction and exergetic efficiency for blends of fuel (**Figure 9.6-a**). It is seen that blends of has higher exergy destruction at varying loads compared to diesel fuel(PBD0), When the amount of exergy destruction increases the exergetic efficiency of blends fuel decreases as expected. The exergy destruction of various blends of fuel was obtained maximum at lower load (10%). So, they are inversely proportional with increasing engine load.

Entropy generation rates defines the performance of the engines and is linked to thermodynamic irreversibility. It is due to heat transfer during the thermal engine cycle. The entropy generation, S_{gen} increases with the engine loads increasing for all blends of fuel as shown in **Figure 9.6(b)**. It is directly related to the loads. The values of S_{gen} taking into consideration for all the blends of fuel from higher to lower are mentioned as: PPBD100 > PPBD80 > PPBD60 > PPBD40 > PPBD20 > PBD0. At 90% of engine load, the entropy generation for all blends of fuel lies between 0.0225 kW/K and 0.0265 kW/K.

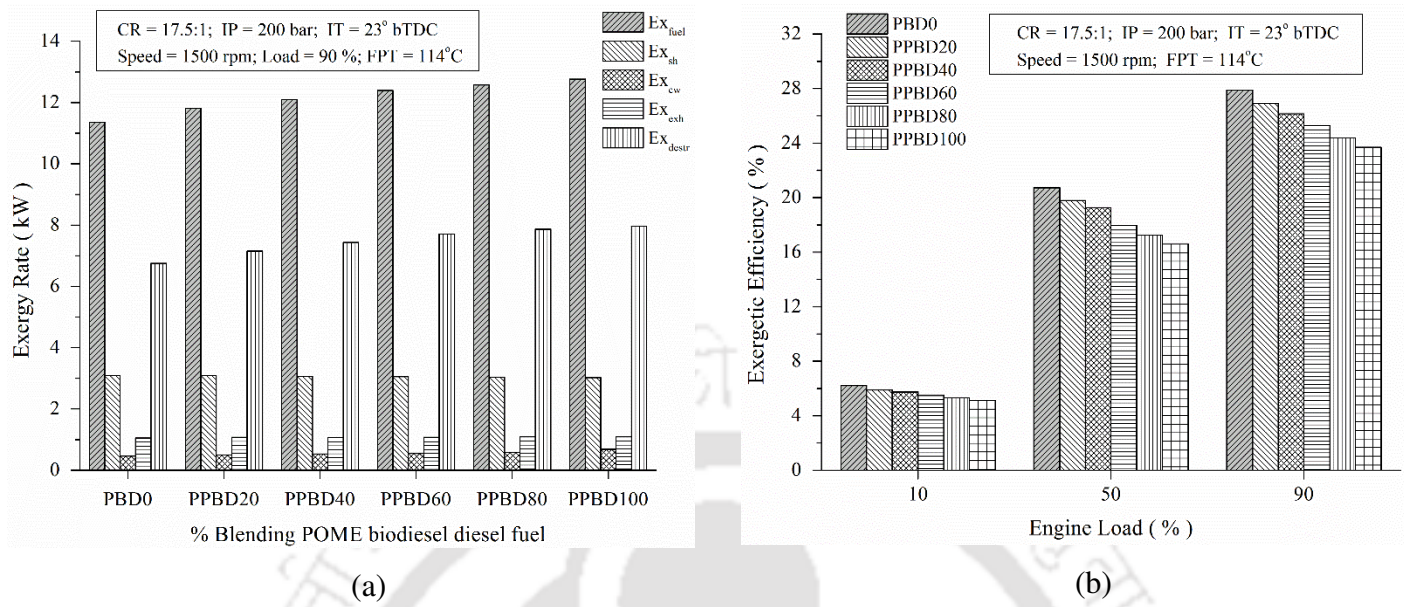


Figure 9.5: The exergy parameters for different blends of fuels: (a) exergy rates; (b) exergetic efficiency.

“Sustainability index, SI” is the main parameter to evaluate the viable option for the optimization of the CI engine and related to η_{exerg} . The SI results are directly proportional to the loads. The variation of SI of blends of fuel with engine loads is displayed in **Figure 9.7**. It is maximum at 90% and minimum at 10% of load for varied the blends of fuel. The POME (biodiesel) fuel are less sustainable than the diesel (PBD0), because of the relatively high consumptions of biodiesel in the engine for every engine loads, leading a lower “exergetic efficiency and sustainability index”. Increasing the percentage fractions of biodiesel in blend ratios caused to drop the values SI as, PBD0 > PPBD20 > PPBD40 > PPBD60 > PPBD80 > PPBD100 at every engine loads. Thus, the engine is better sustainable if the low blends of fuel are used to run in the engine compared to high levels. At engine load of 90%, the sustainable index for PBD0, PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100 of fuels are 1.37, 1.35, 1.34, 1.32, 1.31 and 1.31, respectively.

Table 9.4: Exergy rate analysis of test fuels.

Fuel	Load (%)	Exergy rates (kW)				
		$\dot{E}x_{fuel}$	$\dot{E}x_{sh}$	$\dot{E}x_{cw}$	$\dot{E}x_{exh}$	$\dot{E}x_{destr}$
PBD0	10	5.81±(0.026)	0.36±(0.00)	0.05±(0.001)	0.11±(0.003)	5.28±(0.042)
	50	8.39±(0.038)	1.74±(0.007)	0.14±(0.004)	0.51±(0.015)	6.01±(0.048)
	90	11.35±(0.051)	3.08±(0.012)	0.46±(0.012)	1.05±(0.032)	6.75±(0.054)
PPBD20	10	6.08±(0.027)	0.36±(0.00)	0.05±(0.001)	0.13±(0.004)	5.54±(0.044)
	50	8.87±(0.04)	1.75±(0.01)	0.21±(0.005)	0.53±(0.016)	6.38±(0.051)
	90	11.81±(0.053)	3.09±(0.01)	0.49±(0.012)	1.07±(0.032)	7.14±(0.057)
PPBD40	10	6.19±(0.028)	0.35±(0.00)	0.04±(0.00)	0.17±(0.005)	5.62±(0.045)
	50	9.05±(0.041)	1.74±(0.01)	0.13±(0.003)	0.51±(0.015)	6.65±(0.053)
	90	12.09±(0.054)	3.06±(0.012)	0.53±(0.013)	1.06±(0.032)	7.43±(0.059)
PPBD60	10	6.52±(0.029)	0.36±(0.00)	0.04±(0.00)	0.19±(0.006)	5.92±(0.047)
	50	9.52±(0.043)	1.71±(0.01)	0.17±(0.004)	0.53±(0.016)	7.1±(0.057)
	90	12.39±(0.056)	3.06±(0.01)	0.55±(0.014)	1.07±(0.032)	7.71±(0.062)
PPBD80	10	6.67±(0.03)	0.35±(0.00)	0.05±(0.001)	0.18±(0.005)	6.09±(0.049)
	50	9.87±(0.044)	1.7±(0.01)	0.21±(0.005)	0.54±(0.016)	7.41±(0.059)
	90	12.57±(0.057)	3.04±(0.01)	0.58±(0.015)	1.09±(0.033)	7.86±(0.063)
PPBD100	10	6.83±(0.031)	0.35±(0.00)	0.04±(0.00)	0.2±(0.006)	6.23±(0.05)
	50	10.21±(0.046)	1.69±(0.01)	0.26±(0.006)	0.56±(0.017)	7.69±(0.062)
	90	12.76±(0.057)	3.02±(0.01)	0.68±(0.017)	1.09±(0.033)	7.96±(0.064)

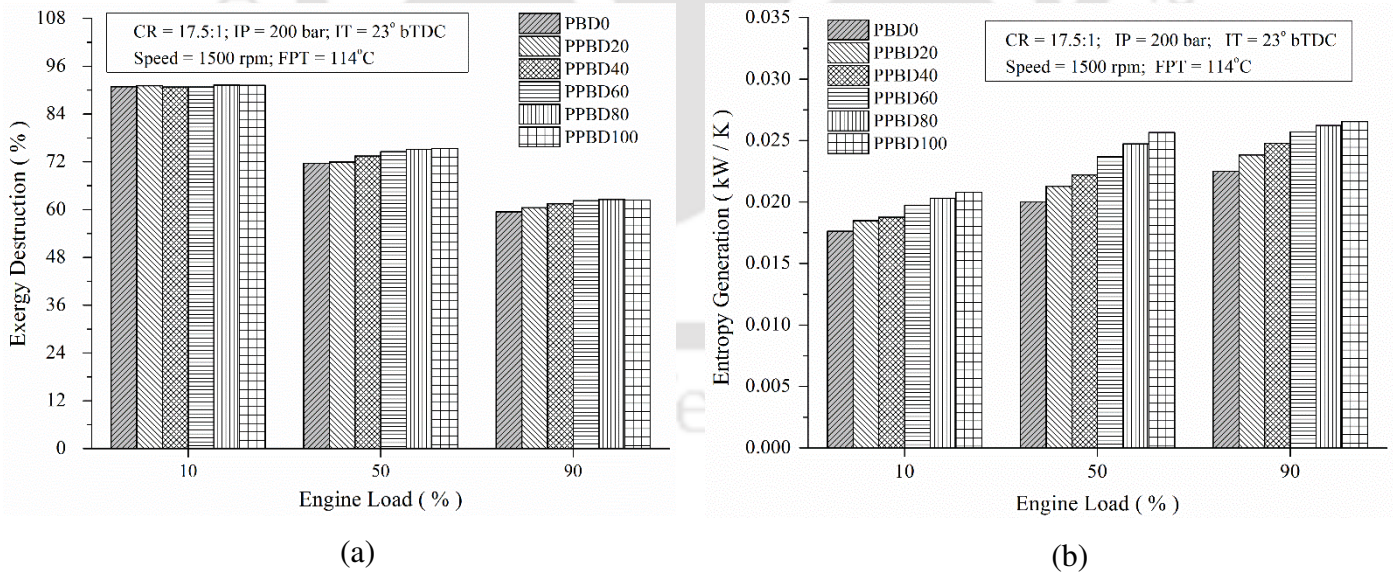


Figure 9.6: The exergy parameters with a varying engine loads:: (a) exergy destruction; (b) entropy generation.

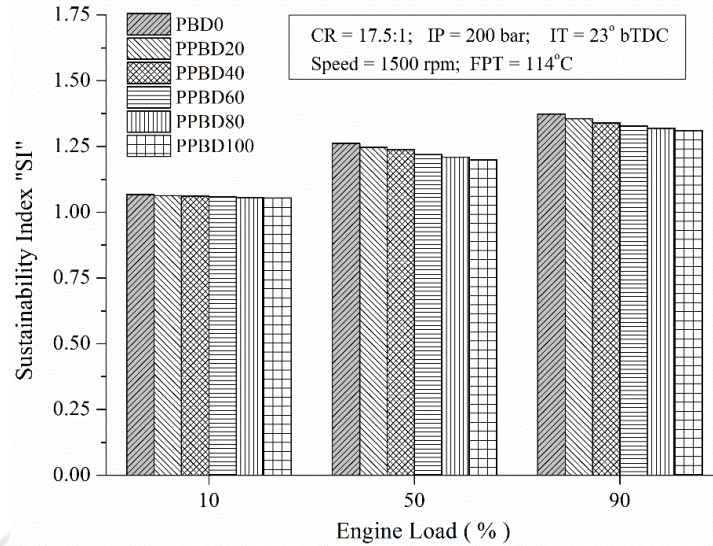


Figure 9.7: Sustainability index for blends of fuels used in CI engine with a varying engine loads.

9.3.4 Economic (Exergy Cost) Analysis

In the present study, diesel (PBD0) and five different preheated blend of biodiesel (PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100) used in a CI engine (1500 rpm), were evaluated in terms of exergy cost at engine load of 90%. Then, different blends of fuel are used in the engine at standard operating condition of CR 17.5, fuel IP 200 bar and IT 23° bTDC. When the engine runs by blends of fuel with available exergy rate, a product stream generated as, effective power and destruction of exergy due to irreversibility. The cost per unit exergy of the engine involves the cost per input fuel unit exergy (C_{fuel}), the cost per unit exergy of generated power output (C_{sw}) and the cost per unit exergy destruction (C_{destr}) for blends of fuel, all of them were calculated by using Eqn. (B25) Appendix-B and shown in **Table 9.5**. While calculating the cost per exergy units of C_{sh} and C_{destr} of different blends of fuel, the input exergy rate, \dot{E}_{fuel} (MJ/hr) and their corresponding product exergy rates of useful exergy, \dot{E}_{sh} (MJ/hr) and exergy destruction, \dot{E}_{destr} (MJ/hr) registered in **Table 9.1** were used. In the present study, the same engine was for all test fuels, so that the engine capital cost was 0.03126 \$/hr with operating and maintenance cost of the system (engine) as, 0.02726 \$/hr (as shown in **Table 9.2**).

Figure 9.8(a), highlights the cost rate associated with energy from sources of different blends of fuel. The cost rate of 4.43 \$/hr, linked with 100PBD was the maximum among all energy

sources due to a recent high biodiesel price. The cost rate linked with diesel (PBD0) was 1.04 \$/hr., roughly four times smaller than that connected with PPBD100. However, the cost rate associated with PPBD40 was two times higher than diesel (PBD0) and two times smaller than neat preheated biodiesel (PPBD100). The cost rate increases even more dramatically for increasing percentage fractions of biodiesel in blend ratios. At this moment, among the five types of blends of fuel, up to PPBD40 blends of fuel was found the best fuels to provide the lowest cost rate compared to high levels of blended fuel. But, in case of mass production biodiesel can at least compete with higher diesel fuel price. Using PPBD100 blend of fuel would become economic due to sufficiently enough supply of biodiesel to the world market to run in a diesel engine as a diesel fuel which leads significant reduction of price of biodiesel oil. The capital investment on the various test fuels were the same (**Table 9.2**). It was obvious that the cost rate on PPBD100 was the highest among the other test fuels. This is due to a high production cost per liter of biodiesel compared with diesel. Here, it should be emphasized that diesel is produced across the world in a large scale for which the cost of production is low at this moment. With renewed interests on biodiesel and increased costs associated with diesel, it can be predicted that larger scale production of biodiesel can supersede diesel in future demands. Hence in today market, the cost of one liter neat biodiesel (POME) was obtained 3.52 \$ which is about three times higher than the price of one liter mineral diesel (PBD0) as mentioned in **Table 9.5**. Moreover, the biodiesel oil offered extra benefits for developing countries economy, creating new jobs, reduced toxic exhaust emission and meet emission norms. In view of this, today many developed countries are engaged in a mass scale generation of biodiesel from vegetable oil and animal fats to substitute diesel fuel partly or full and satisfied a high-energy requirements of fast industrialization. This provides a great opportunity in a long term operation and the price of biodiesel may reduce significantly even below the current price of diesel fuel. Thus, at this circumstance the cost rate per useful exergy unit of a high levels blend (up to PPBD100) run a diesel engine may be lowest as related to diesel (PBD0).

Figure 9.8(b) displays the results of cost per unit exergy available in the form of useful brake power and exergy destruction. Among the different blends of fuel, the cost per useful power exergy of PPBD100 was maximum, with the price of 0.75759 \$/MJ (**Figure 9.8-a**). This was owing to the very high biodiesel oil price in the current market. While, the neat diesel fuel (PBD0) was lowest and found to be 0.1708 \$/MJ. Increasing the percentage fractions of biodiesel in blend mixture ratios, the cost per generated useful power exergy was shown increasing, due to the price

increments of blends of fuel with increasing fractions of biodiesel in mixture ratios. Similarly, the cost per destruction exergy of PPBD100 was higher, with price of 0.39869 \$/MJ, whereas for the cost per destruction exergy due irreversibility of PBD0 was found lowest with 0.11117 \$/MJ (Figure 9.8-b). From the results of full economic analysis (fuel cost, and useful power exergy and exergy destruction), it was found that the most economical blends were (PPBD20–PPBD60) and the costly blends were (PPBD80–PPBD100) respectively compared to diesel fuel (PBD0). The best thermodynamic feasibility of the engine operation occurred when the engine runs with pure diesel fuel and the most economical condition is seen biodiesels with the PPBD20 and PPBD40 blends.

Table 9.5: Fuel cost rates associated with blends of fuel and cost per product exergy unit during engine operation (90% load and 1500 rpm).

Blends of Fuel	Fuel cost		Cost per unit Exergy	
	C_{fuel} (\$/MJ)	C_{sh} (\$/MJ)	C_{destr} (\$/MJ)	
PBD0	0.027	0.171	0.111	
PPBD20	0.042	0.272	0.166	
PPBD40	0.056	0.382	0.221	
PPBD60	0.072	0.499	0.278	
PPBD80	0.088	0.625	0.339	
PPBD100	0.104	0.754	0.397	

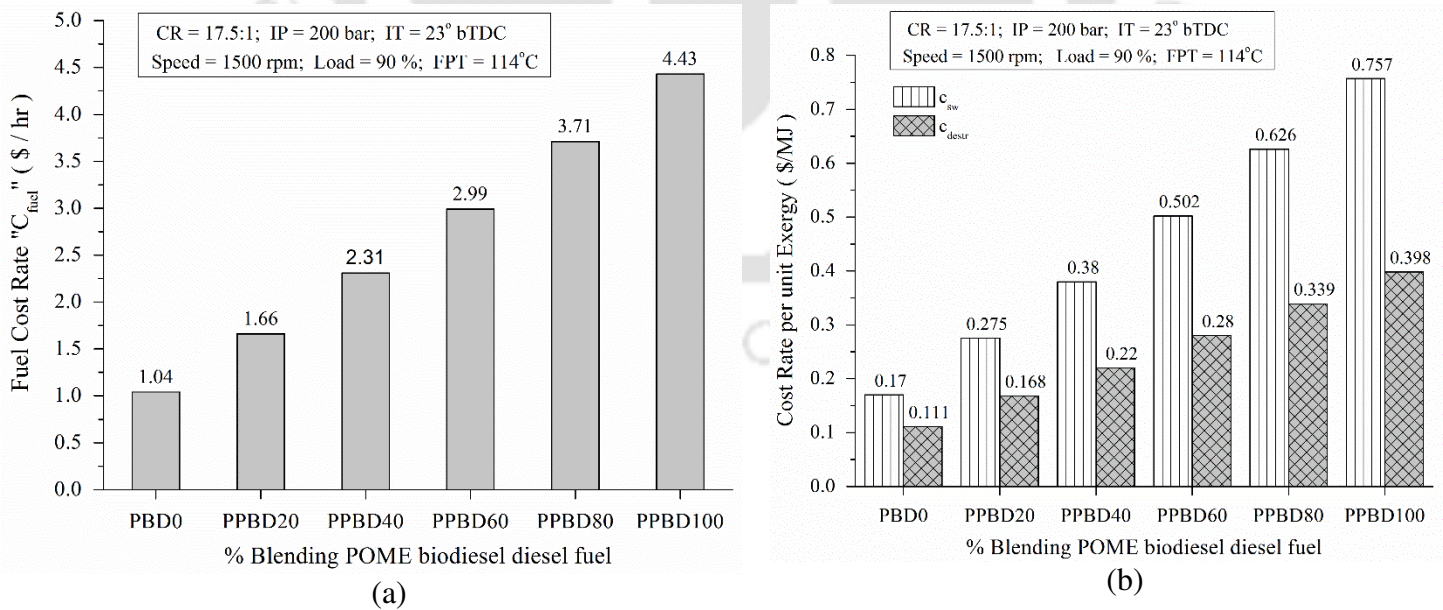


Figure 9.8: Cost analysis for different blends fuel in the CI engine: (a) Fuel cost rate associated with different blends of fuel, (b) Cost per unit exergy products (i.e. useful and destruction).

9.4 Summary

In the present investigation, thermodynamic and thermo-economic analysis of a diesel engine was performed. To this aim, palm oil methyl ester biodiesel is preheated (114 °C) blends with diesel from 0% to 100% by volume (PBD0, PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100) were used to test a single cylinder VCR DI compression ignition engine. Then, the combustion reactions were determined using experimental data. Thermo-economic analysis was performed applying energy, exergy and exergy cost (economic balance) to steady state engine control volume. The results of thermodynamic analysis revealed that at 90% of load, the maximum values energetic and exergetic efficiency were 28% and 26.2% when using blend fuel (PBD60), closer to diesel (PBD0) to 29.12% and 27.18%, respectively. It means that, the percentage fractions of biodiesel are inversely proportional to useful exergy. The minimum values of brake specific fuel consumption for PPBD60 was 0.30612 kJ/kW.hr whereas it was 0.26345 kJ/kW.hr for diesel (PBD0) and destroyed exergy (7.71 kW and 66.75 kW) were also obtained in the same conditions, respectively.

Input fuel exergy rate is greater than fuel energy rate for all blends of fuel, due to chemical exergy factors of each fuel. The optimum preheated biodiesel/diesel blends in the energy analysis section (PPBD20–PPBD60) was also admissible in exergy analysis. The exergetic efficiency of this blend is very close to that of diesel fuel. All blends of fuel offered quite higher cost fuel with respect to diesel. The cost of fuel price per liter of POME is quite significant that diesel in the present study. The full economic analysis (cost rate per unit generated useful exergy and exergy destroyed) showed higher with increased percentage fractions of biodiesel, and only up to 60% biodiesel–diesel blends of fuel (PPBD20–PPBD60) were affordable.

Considering all results obtained from thermo-economic analysis, it is conducted that the best thermodynamic condition of the engine occurred when the engine was fueled with PPBD20–PPBD60, however engine operated with PPBD0 was the most economic condition compared to other preheated POME biodiesel/diesel blends

Diesel engines are robust, durable and having lowest fuel consumption than any other prime movers. The importance of CI engines for public and private transport sector application is growing day by day and their use will continue to grow in future. There is no doubt that the magnitude of the present fuel crisis is going to worsen in future because the fuel consumption rate had significantly increased over the last few decades. The steep increase of the prices of mineral fuels, the scare of its source and ever-growing environmental concerns, however, made the use of fossil diesel in CI diesel engine becomes a serious issue. Nevertheless, the vast scale of the applications of this diesel fuel has made it almost impossible for complete replacement, at least for the next few decades. As a result, the search of a suitable alternatives becomes essential to tackle most of these problems effectively and instantly. Around the globe, the most practiced alternative fuel for diesel engine is the methyl ester of vegetable oil or 'biodiesel'. However, present diesel engines, tailor made for fossil diesel, cannot replicate diesel like performance run by 'biodiesel', because of its diesel alien properties. There are two possible methods, by which, diesel like performance or better than that may be achieved; firstly, by modifying the fuel itself; secondly, by modifying the fundamental operating characteristics of the engine, namely, (CR, fuel IP, fuel IT). In this regard, the concept of the present work is set to combine both of these methods in diesel engine. The whole experiments are conducted on a single-cylinder, four stroke-stroke, constant-speed, and VCR DI CI engine. To begin with, the base engine is slightly modified so as to allow it to operate under fuel and intake air preheating mode operations. In this connection, a biodiesel and intake air preheating mechanisms to be added with the base diesel setup for preheated biodiesel blend operations is developed and the necessary fuel and intake air preheaters are fabricated. The diesel engine is then run with neat diesel as baseline testes for reference results. Then a series of experimental investigations have been conducted to explore the performance, combustion and emission characteristics with optimization of engine operation using biodiesel–diesel blends, preheating neat biodiesels, preheating biodiesel–diesel blends in a VCR CI engine with/without intake air preheating mode. The biodiesel selected for this study are the nonedible castor oil methyl ester (COME) and edible palm oil methyl ester (POME), because of its exceptional combustion characteristics. The results from the above studies are compared with neat diesel data, experimented at standard diesel specifications. This is followed by the systematic thermodynamic and thermo-economic studies that are performed at three loads where maximum efficiencies are obtained and mutually compared. The principal contributions of the present work and further research range have also been highlighted.

10.1 Contribution of the Present Work

In India non-edible and edible oil like castor oil biodiesel and palm oil biodiesel are available in abundance, which can be used as a fuel in a diesel engine. Exhaustive studies of these methyl ester biodiesels have been carried out in this research work. The main aim of the present experiment was to improve the fuel properties of biodiesel by reducing viscosity and density close to that of conventional diesel fuel and evaluate the performance of engine with that of modified oil. An extensive investigation on performance parameters, combustion and emission characteristics of compression ignition engine using biodiesel blends in varying volumetric ratios, preheated neat biodiesel, preheated biodiesel blends in varying volumetric ratios were evaluated and compared to those of diesel or its reference data. Thus their suitability as an alternative fuel is examined. The important properties of different biodiesel blends are determined/measured, and the results are also compared to the blended biodiesel standards specifications for validation. Thus better performing preheated biodiesel blends ratios with their suitable combinations of engine operating parameters are identified. The present effort has contributed mainly in the following aspects:

- A comprehensive survey of available literature has been done on CI engines fuelled with biodiesel, biodiesel blends, preheated biodiesel and preheated biodiesel/diesel blends with intake air preheating mode operation, to develop an understanding of performance, combustion and emission behavior of the engine. In addition to this an exhaustive literature review was also undertaken on modifications engine operating parameters, EGR techniques of exhaust NO_x emission reductions. Finally, the thermodynamics and thermo-economic cost estimation of biodiesel was also thoroughly reviewed.
- The design and development of test setup and experimental data collection and analysis are equally important for any experimental research. A suitable test rig including fuel preheater (helical coiled heat exchanger), intake air preheater (shell and tube heat exchanger), EGR system, the required alternative fuel lines and fuel tank, piping for inlet and outlet of exhaust gases and fresh air to the preheater are made. Also thermocouples are connected to pipe for recording required temperature. An emission measuring equipment exhaust gas analyzer is incorporated to the experimental setup. A detailed experimental investigation on a diesel engine characteristics have been carried out.
- A comprehensive experimental analysis and optimization of engine operating parameters was carried out using diesel fuel, biodiesel blends, preheated neat biodiesel, preheated

biodiesel–diesel their different blends in diesel without/with intake air preheating mode engine operation by varying engine loads and EGR rates in a VCR DI CI engine and large amount of useful experimental data was generated.

On the basis of thorough experimental investigations carried out, following salient inferences have been drawn:

10.1.1 Engine operation using Baseline Diesel Fuel

- Extensive experimental results suggest that the combination of CR of 17.5 with 200 bar fuel IP and 23° bTDC IT exhibited best engine performance with diesel fuel for the entire range of investigation. Hence, it is considered as optimum engine parameters for diesel fuel engine operation.
- Higher brake thermal efficiency (28.7%), higher volumetric efficiency (82.7%), lowest brake specific fuel consumption (0.27018 kg/kW.hr) and lowest exhaust gas temperature (267.3°C) were observed at optimum engine parameters of CR17.5, 200 bar fuel IP and 23° bTDC of IT for diesel fuel.
- Lowest carbon monoxide (0.16702%) and unburnt hydrocarbons (43 ppm) with slightly higher NO_x emissions (598 ppm) were observed at optimum parameters of CR17.5, 200 bar fuel IP and 23° bTDC IT for diesel fuel operation.
- Higher maximum peak cylinder pressure (72.5 bar) and maximum heat release rate of 45.70 J/deg.CA was achieved at full load for optimum engine operation with diesel fuel. The ignition delay observed for optimum operation for diesel fuel mode was 18.12 °CA.

10.1.2 Engine operation using Biodiesel Blends

This section represents the results of COME and POME biodiesel–diesel blends (by volume basis) run in a diesel engine with a default or standard operating conditions CR17.5, 200 bar of fuel IP and 23° bTDC of IT of engine. The results are compared with baseline diesel fuel at standard operating combinations of CR, fuel IP and IT. Some of the important conclusions are as follows:

- The fuel properties of POME and its blends, and COME and its blends in relation to that of blended biodiesel standard specifications were compared. In relation to ASTM D7467, the viscosity of PBD100 and PBD10, PPBD20 were found to be about 112% was higher, and 14.4% and 2.9% was lower, respectively. Whereas, the viscosity of CBD100, CBD10 and CBD20 were obtained to be about 434.4% was higher, 2.2% was lower, and 26.3%

was higher, respectively. In relation to EN14214, the calorific value of was lower by about 13.7% and 9.7% for PBD100 and CBD100 biodiesel.

- Performance parameters of engine such as bp, BTHE dropped with increasing the content of biodiesel in the blends, BSFC is increased for all blends compared to those of diesel. This is because of high viscosity coupled with lower heating value of the fuels.
 - BTHE for PBD10, PBD20, PBD30 and PBD40 oils is reduced by 4.2%, 23.7%, 34.9% and 39.3% respectively, whereas for CBD10, CBD20, CBD30 and CBD40 oils is decreased by 10.6%, 32.2%, 38.1% and 43.9% respectively compared to diesel (BD0) at the rated load (12 kg). This is because of lower Calorific value and higher viscosity coupled with density of the fuel.
 - BSFC for BD10, BD20, BD30 and BD40 oils is increased by 4.7%, 31.6%, 44.7% and 58.8% respectively, whereas for CBD10, CBD20, CBD30 and CBD40 oils is raised by 10.4%, 38.8%, 55.9% and 70.4%, respectively compared to diesel at rated load (12 kg), and is result of delay in ignition process.
- Emission parameters of engine such as CO and HC are decreased for all blends compared to diesel. This is due to lower calorific value and high viscosity coupled with density of the fuels chosen. While NO_x emission for all blends are increased with increasing the percentage of biodiesel in the blend.
 - CO Emission for BD10, BD20, BD30 and BD40 oils is decreased by 14.5%, 30.6%, 44.2% and 67.6% respectively, whereas for CBD10, CBD20, CBD30 and CBD40 oils, it is reduced by 7.8%, 18%, 29% and 57.5%, respectively compared to diesel at the rated load (12 kg). This is due to the presence of more oxygen in the biodiesel, leads complete combustion of the fuel.
 - HC Emission for BD10, BD20, BD30 and BD40 oils dropped by 15.3%, 29.6%, 41.8% and 54.1% respectively, whereas for CBD10, CBD20, CBD30 and CBD40 oils, it is dropped by 4.1%, 16.3%, 31.6% and 46.9%, respectively compared to diesel at the rated load (12 kg). It has the same reasons as that of CO emission mention above.
 - NO_x Emission for BD10, BD20, BD30 and BD40 oils is increased by 11.3%, 16.1%, 22.9% and 30.9% respectively, whereas for CBD10, CBD20, CBD30 and CBD40 oils, it is higher by 1.5%, 4%, 7% and 11.3%, respectively compared to

diesel at the rated load (12 kg). The reason for this trend is the availability of excess oxygen in biodiesel, resulting complete combustion.

10.1.3 Engine operation using Preheated Neat Biodiesels

The engine performance and emission parameters are evaluated with neat POME (PBD100) and COME (CBD100) as fuel in a diesel engine were thoroughly studied at full engine load. Preheating of neat biodiesel was done from temperature of 54 °C –138 °C with increments of 12 °C. For all fuel inlet temperatures, the performance of engine was evaluated and compared. The results of this study is summarized as follows.

- Most of the fuel properties of POME (PBD100) and COME (CBD100) biodiesel in relation to that of mineral diesel fuel were comparable. However, in relation to diesel, the kinematic viscosity of PBD100 and CBD100 were found approximately to be 6.78 and 2.69 times that of diesel, density was nearly 12.5% and 12.9% higher, the flash point and cetane number are higher and met the lower limits fuel standards (ASTM D6751, EN14214 and IS 15607). The calorific value was lower by about 10.96% and 14.1% respectively.
- Preheating of the COME (PBD100) and POME (CBD100) caused a considerable decrease in its kinematic viscosity and density, thus causing them to approach the values of biodiesel standards close to diesel fuel. It contributes to better fuel atomization at the elevated temperature of the biodiesel. The density of COME and POME was reduced by 6.97% and 6.28%, respectively, whereas the viscosity of oils was reduced by 59.32% and 58.75% with an increase in fuel inlet temperature from 54 °C to 114 °C.
- From the properties of neat POME and COME biodiesel, and engine test results, it was observed that fuel inlet temperature of 114 °C is an optimal preheating temperature for neat biodiesel and can be substituted for diesel without any engine modification. Particularly, PBD100 (114 °C) and CBD100 (114 °C) yielded with a high improvement in the BTHE with significant reduction CO and HC exhaust emissions. However, for higher value of fuel inlet temperature above 114 °C, performance was observed to be marginally inferior, because of excessive fuel leakage (i.e. lower fuel viscosity). The upper limit of fuel inlet temperature tested was 114 °C.
- At full load condition, the engine performance (BSFC and BTHE) and emission levels (CO, HC, CO₂ and NO_x) of preheated POME and COME at elevated fuel inlet temperature 114 °C as compared with 54 °C; BSFC was reduced by 40.81% and 4.5%, respectively;

BTHE was increased by 48.7% and 53.19%, respectively. The percentage reduction of CO was 85.47% and 48.44%, respectively; the percentage reduction of HC was 5.38% and 59.09%, respectively; the percentage increased CO₂ emission of was by 35.64% and 39.6%, respectively; and the percentage increased NO_x emission was by 44.57% and 58.84%, respectively.

10.1.4 Engine operation using Preheated Biodiesel Blends

Studies on performance, combustion and emission characteristics of a VCR DI compression ignition engine fueled with preheated POME biodiesel blends (20–100% by volume with diesel) with increments of 20% under different operating conditions. Experiments were carried out at varying engine loads and EGR rates under standard operating conditions (CR17.5, fuel IP and IT of 200 bar and 23° bTDC). The following conclusions are drawn from the studies.

- Preheated biodiesel and blending with diesel caused improvement in some properties such as kinematic viscosity, density, pour point, calorific value of various blends. However, flash point decrease as the percentage of diesel increases.
 - The viscosity results of preheated blended fuel samples measurement vary in the range of 3.38 to 5.37 mm²/s for PPBD20 and PPBD100 respectively, whereas the density was varying from 851 to 876 kg/m³. Accordingly, the viscosity and density of the preheated blend increases as the palm oil methyl ester amount increase in the fuel mixture.
 - The kinematic viscosity of PPBD60 blends falls from 5.37 to 4.08 mm²/s at 114 °C, which is 24.02% less than PPBD100 at the same temperature. While, the density of the blending PPBD40 is 1.94% lower than that of PPBD100 due to the combined effect of preheating and blending biodiesel with diesel. Thus, the blended fuel viscosity and density still meets the blended biodiesel standards requirements (ASTM D7467 and BIS) for up to 60% preheated blended ratio (PPBD60).
 - The blending decreases the flash point of the neat biodiesel POME (PBD100). However, the effect of preheating biodiesel and blending with diesel caused to increased slightly and higher at PPBD100. All preheated biodiesel blends meet ASTM D7467 standards (minimum 52) and are extremely safe to handle.
 - The calorific value is not specified in the blended biodiesel standard ASTM D7467 but is presented in BIS and EN14214 with a minimum value of 35 MJ/kg. The

result shows that the calorific value of the preheated biodiesel/diesel blend decreased with increasing volumetric percentage of biodiesel. The minimum heating value measured was 39.82 MJ/kg (PPBD100), which satisfies the requirements of the EN14214 standard. It is approximately 10.89% lower than the heating value of diesel (44.69 MJ/kg).

- So an optimum blending ratio of up to 60% blending (up to PPBD60) has been recommended by compliance of the preheated biodiesel/diesel blend specification to the specified standards (ASTM D7467 and BIS), which can be used in diesel engines without modification and without sacrificing much power loss.
- For optimum preheated biodiesel blends of 60% POME biodiesel content in diesel, the combination of engine parameters of CR17.5, 200 bar of fuel IP with 23° bTDC of IT resulted in best engine performance with minimum exhaust emissions. At the optimum engine parameters, the highest BTHE, lowest BSFC and slightly higher EGT and volumetric efficiency was observed. Moreover, lowest CO and HC emissions were obtained with a slight increase in NO_x emissions at optimum preheated biodiesel blends running conditions. However, EGR30% rate significantly reduced the NO_x emission with a slight effects of engine performance parameters and CO and HC emission levels.
- A maximum peak cylinder pressure and lower ignition delay was achieved at full load for standard engine operation with optimum preheated biodiesel/diesel blends (PPBD20–PPBD60). The average values of PCP 70.7 bar and 15 °CA of ID were observed for optimized preheated biodiesel/diesel blend (PPBD60) at near optimal engine operation. The application EGR rates, caused a longer ID for all blends of fuels.
- In general, the inferences from experimental findings have been described as follows: With respect to utilization of test fuels in the diesel engine, the preheated biodiesel/diesel blends (PPBD20–PPBD60) with EGR rate of EGR30% offers best alternative for replacement of diesel. At full load operated conditions; the average percentage, increase of BSFC by 7.6 –10.5%; BTHE dropped by 3.1 – 4.9%, EGT decreased by 19.13 –22.54%; and volumetric efficiency decreased marginal by 1.54 –1.94%.The average percentage, PCP decreased by 6.95 –12.73%; and ID increased by 11.78 – 20%.The average percentage CO increased by 21.1 – 42.2%; HC increased by 14.3 –19.44%; and NO_x decreased by 64.1 – 70%.

- The effect of increasing EGR rates up to 40% was also investigated. EGR was intended to reduce the high NO_x emissions that are prevalent during high engine loads with preheated biodiesel added to diesel fuel.
 - BTHE decreases for all preheated biodiesel blends at increasing EGR rates with the loss being more for preheated blends. The maximum drop of 11.3% from baseline diesel performance was realized with PPBD60 blend at full engine load. BSFC also decreased for preheated biodiesel blends with increase in EGR rates.
 - NO_x emissions were reduced by 45.9% for PPBD60 blend at EGR30% rate for full engine load, as compared to diesel fuel operation.
- As per the experimental observation, it was found that combined effect of fuel preheating attachment and EGR system provided better result on engine performance and emission parameters run for biodiesel blends than individual effect for utilization of biodiesel oil in a diesel engine. The optimal EGR rate of EGR30% rate is an effective solution for a high level of NO_x reduction with a minor affects engine performance parameters, combustion and emission characteristics.

10.1.5 Engine operation using Intake Air Preheating Mode for Preheated Biodiesel Blends

This investigation focused on understanding the impact of intake air preheating and preheated biodiesel blends on understanding the fundamental properties of biodiesel blends along with engine performance. The inlet air temperature is an important parameter for creating the homogeneous mixture. An increasing inlet air temperature is used to improve fuel vaporization and create more homogeneity air/fuel charge. The major findings of this research are listed below.

- The inlet temperature increased the BTHE, decreases the specific fuel consumption, reduces emissions (CO and HC) and the ignition delay period. Due to preheating of inlet air uniform combustion takes place to reduce engine noise.
- With respect to 33 °C (ambient condition), at 61 °C intake air temperature operating in a diesel engine using various blends (PBD0, PPBD20, PPBD40 and PPBD60): The average percentage increase of BSFC by 5.5 – 7.83%, BTHE decreased by 5.86 –9.65%; the average percentage PCP increased by 12.53–13.85%, ID decreased by 22.2 – 28.6%; the average percentage CO decreased by 43.3 –62%, CO₂ increased by 5.6 – 19.3%, HC decreased by 39.5 –75.0%, and NO_x increased by 19.8 –29.4%.

10.1.6 Effects of Modified Operating Parameters (CR, IP, IT) in a Diesel Engine for Preheated Biodiesel under Intake Air Preheating Mode Operations

Further, investigations are carried to enhance the performance of preheated biodiesel blends (PPBD20, PPBD40 and PPBD60) in diesel engine characteristics. This chapter represents the results of preheated POME biodiesel blends run diesel engine for different combinations of preset CR, fuel IP and IT at 90% load and EGR30% rate. The results are evaluated and compared with diesel at standard CR, IP and IT combinations. Based on the results the following inferences are drawn.

- Optimization of VCR DI CI engine with optimized preheated biodiesel-diesel blends (PPBD20–PPBD60) were conducted. It was observed that the combination of CR 18 with 200 bar of fuel IP and 23° bTDC of IT, exhibited superior performance of the engine with minimum exhaust emissions. The BTHE and BSFC can be improved along with a reduction in emissions except for NO_x at higher CR (CR18). This is due to the better combustion and air-fuel mixing. The maximum reduction for CO and HC emissions were found in this condition due to higher oxygen content biodiesel and better combustion.
- At higher loads, BTHEs of preheated biodiesel blends are a little lower than diesel. For a constant, IP and IT, an increase in CR increases the BTHE and lowers the BSFC. However, increase in CR to 18 causes a maximum reduction of EGT of PPBD20, PPBD40 and PPBD60 and found to be 24.6–26.8%.
- Preheated biodiesel/diesel blends runs more smoothly than diesel in diesel engine at higher CR. Increase in CR to 18 causes a maximum of 10.8–12.6% increase in the PCP. The most significant finding of this study is the decrease of ID for preheated biodiesel blends with respect to diesel. At higher CR, the high pressure and temperature at the primary phases of combustion causes the preheated biodiesel blends to burn more proficiently. As the CR increases the value of maximum net heat release decreases. A heat release is maximum (46.2 J/deg) for PPBD20 at CR 18 at 362.92 deg crank angle.
- In addition, the engine operated with higher CR of 18, the emission constitutes of CO and HC were found the least. The increase in CR from 15 to 18 causes an average reduction of CO and HC emission for PPBD20, PPBD40 and PPBD0 blends were found by (26.1%, 28.1% and 41.7%), and (30.8%, 34.3% and 68.4%), respectively.

Whereas, the average increased of NO_x emission were obtained by (26.6%, 39.9% and 49.8%), respectively.

- Based on the experimental studies on performance and emission characteristics of optimum preheated biodiesel blends with various injection pressures (188, 200, 212 and 224 bar) at higher CR 18 and standard IT (23° bTDC) operating conditions, it can be concluded that with the increase in fuel IP, the performance of diesel engine operated using preheated POME biodiesel blends approach to that operated using diesel fuel. Positive results were found at high IP (212 bar) and CR of 18. There is a reduction in CO and HC emissions along with an increase in BTHE at high IP compared to that operated at other IPs of 188, 200 and 224 bars. The reduction in drop size of fuel at high IP of 212 bar, is preferable for biodiesel blends due to its higher viscosity and better mixing with air to make a homogeneous mixture. Thus, the best combination of input parameters in terms IP and CR will be 212 bar and 18 at base fuel IT of 23° bTDC, respectively when the engine is operated at 90% load. At this stage, the results specify the optimum preheated POME biodiesel blends (PPBD20–PPBD60) that will simultaneously maximize thermal performance and minimize emission constituents.
 - ❖ On the basis of comparison made with earlier investigations with similar input parameters of CR, fuel IP and load of 18, 212 bar and 90% load respectively, the thermal performance of the engine operated with preheated biodiesel/diesel blends is found to be superior in comparison with that operated (CR, fuel IP and load of 18, 200 bar and 90% load).
 - BTHE of the engine operated using (PPBD20, PPBD40 and PPBD60) are found to be more by 1.3%, 4.9% and 4.1% respectively. BSFC is found lesser by 8.1–9.3%, and, EGT is seen higher by 13.4–17.2%.
 - ❖ Preheated POME biodiesel blends (PPBD60) gives minimum harmful emissions as compared to all other blends. Further, at a higher CR of 18 and IP of 212 bar, the fairly reduced exhaust emissions are observed irrespective of the fuel blend used. Therefore, operating the diesel engine with preheated biodiesel blends at a CR of 18 and IP of 212 bar results in minimum emissions CO and HC but for slight increased NO_x emissions. A comparison with that operated (CR, fuel IP and load of 18, 200 bar and 90% load) investigations revealed that;

- The CO emission of PPBD20, PPBD40 and PPBD60 were lower by 11.5%, 6.4% and 3.4%. HC emissions were lower by 23.5%, 14.8% and 46.7% respectively. NO_x emission were found to be higher by 23.3%, 18.6% and 14.2%, respectively.
- Mixed results on performance and emission characteristics of PPBD20, PPBD40 and PPBD60 came out at varying ITs at higher CR 18 and optimum fuel IP of 212 bar operating conditions. The engine characteristics can be improved by advancing or retarding or at full load condition of engine. For advancing in fuel IT (27° bTDC), engine performance (BSFC and BTHE) and emission (CO and HC) characteristics of engine operated using preheated biodiesel blends approach to that operated using diesel fuel with marginal increases of NO_x emissions. This means that, more fuel is injected the longer ignition delay and the greater the premixed portion of the fuel prior to ignition which promotes premixed combustion and leads to higher NO_x formation. However, retarding fuel IT caused a significant reduction the levels of NO_x emission due to reduced peak pressure and temperature due to less accumulation of fuel. Retarded IT to 19° bTDC reduced the BTHE values for all the blends of fuel compared to the base IT. A comparison with that operated (CR, fuel IP, IT and load of 18, 212 bar, 23° bTDC and 90% load) investigations revealed that;
 - Advancing IT by 4° (from standard to 27° bTDC) caused a maximum reduction of BSFC by (5.8%, 7.0% and 2.4%) and EGT by (28.8%, 19.1% and 21.1%). BTHE was raised by (3.9%, 3.1% and 5.1%) respectively for PPBD20, PPBD40 and PPBD60 blends.
 - Advancing in IT to 27° bTDC also causes drop in CO and HC emissions. It was found to be by (15.8%, 15.7% and 11.9%) and (19.2%, 30.4% and 37.5%), respectively for PPBD20, PPBD40 and PPBD60 blends. While the NO_x emission was slightly increased by (15.8%, 6.3% and 4.8%), respectively.
 - Further, retardation of IT by 4° (from standard to 27° bTDC), causes both BSFC , CO and HC emission be slightly higher. While the retarded IT showed the opposite effect with a sharp decrease in NO_x emissions by (30.1%, 36.3% and 29.7%), respectively for PPBD20, PPBD40 and PPBD60 blends.

From the above results, the high value of CR (CR 18), fuel IP (IP = 212 bar) and advancement of fuel IT (IT = 27° bTDC) shows the improvement in the combustion, performance and emissions characteristics of CI engines using (PPBD20–PPBD60) as a fuel at 90% engine load and EGR30% rate operating condition. At modified conditions BSFC, CO and HC emission decreases and BTHE and NO_x emission increases with respect to modified ones. The result also shows that retarded IT (19° bTDC) as compared to original IT (23° bTDC), emission of NO_x decreases.

Finally, it can be concluded that best results can be found to be optimum combinations of a high CR (CR 18), high fuel IP (212 bar) and varying (advancing/retarding) IT (27° bTDC/19° bTDC).

10.1.7 Thermodynamics and Thermo-economics Study

This phase describes the thermodynamic and thermo-economic analysis of preheated POME biodiesel–diesel blends (PPBD20, PPBD40, PPBD60, PPBD80 and PPBD100) run VCR DI compression ignition under three varying load conditions (low–10%, medium–50% and high–90%) at standard CR-IP-IT operating conditions. Based on the experimental, following are the important observations are noted:

- Preheating and high levels of blend ratios provided maximum input fuel energy and exergy rate compared to diesel (PBD0) due to higher volume flow rate of fuel and lower heating value.
- Input fuel exergy rate is greater than fuel energy rate for all blends of fuel, due to chemical exergy factors of each fuel. The engine run by diesel can recover almost 29% of the fuel energy supplied in the form of shaft power. Rest is flown through the cooling water, exhaust gas and other uncounted losses. The analysis demonstrates that PPBD100 can recover around 25.5% of the energy supplied by the fuel. Rest of the energy is flown through the cooling water, exhaust gas and other
 - Low levels of blend of fuels have a better energetic and exergetic efficiency at every engine loads operation. At engine load of 90%, the energetic and exergetic efficiency of PPBD20 fuel was maximum with 28% and 26.2%, closer to diesel (29.12% and 27.18%), respectively. It means that, the percentage fractions of biodiesel are inversely proportional to useful exergy.
 - The destruction of exergy increases when using biodiesel. Among various

biodiesel blends, up to the 60% preheated biodiesel–diesel blend shows an optimum condition.

- The entropy generation of blends of fuel increased with higher percentage fractions of biodiesel and engine loads.
- All blends of fuel offered quite higher cost fuel with respect to diesel. The cost of fuel price per liter of POME is quite significant that diesel in the present study.
- The results of full economic analysis (fuel cost, fuel consumption and cost rate per unit generated useful exergy and exergy destroyed) showed that higher with increased percentage fractions of biodiesel, only up to the 60% preheated biodiesel blends were more affordable compared to diesel

10.1.8 Concluding Comments

In summary, the use of preheated biodiesel and blending with diesel in compression ignition engines has the advantage of reducing most regulated emissions, as well as improving combustion and engine thermal efficiency. This effect is noticeable even at high biodiesel contents. There is an opposite trend between increasing biodiesel content and any change in combustion and performance characteristics. The effect of biodiesel on these characteristics establishes itself for at low to high biodiesel levels. The PPBD20–PPBD60 has the most pronounced effect on performance parameters, combustion and emissions characteristics at all loads. This is particularly important for future variable compression ratio (VCR) compression ignition engine running on different biodiesel-diesel blends. According to the finding of this thesis, the current level of optimum preheated POME biodiesel blends showed substantial improvement in performance parameters and low level of regulated emissions at a best combination of CR of 18, fuel IP of 212 bar at 27° bTDC (refer **Table 10.1**). However, plans towards reducing dependence on fossil fuels push towards the use of alternative fuels of palm oil methyl ester biodiesels. Apart from India, the use of palm oil methyl ester (POME) biodiesel in the engine is still not used. The highly used biodiesel in India are non-edible vegetable origins of Jatropha and karanja oil limited to low proportion biodiesel-diesel blends (ranging from 6% to 20%). The changes in engine performance parameters and NO_x emission levels, when running on high percentage POME biodiesel/diesel blends (up to neat POME–100% POME biodiesel), should be taken into account in future flexi-fuel engine design.

Table 10.1: Engine performance and emission characteristics for blends of fuel at optimal operating (CR-IP-IT) as compare to diesel (CR 17.5, IP 200 bar, 23° bTDC of IT) conditions.

Optimal operating parameter (CR, IP, IT) [A]			Performance analysis (%) [B]	Emission analysis (%) [C]				Optimal condition with respect to diesel					
								B	C	Remarks			
CR	IP	IT	Blends	BSFC	BTHE	EGT	CO	HC	CO ₂	NO _x			
18	212	27	PPBD20	-10.1	+5.6	-20.8	-44.1	-46.2	+15.6	+21.7	↑	↑	Performance, and emission parameters were improved at optimal operating conditions except NO _x
			PPBD40	-6.8	+2.5	-1.6	-51.3	-58.9	+22.2	+34.7			
			PPBD60	-0.02	+0.4	+2.6	-63.8	-71.8	+28.7	+47.6			

10.2 Application Potential

Petroleum diesel fuel is the most efficient of all internal combustion power systems because of the superior engine efficiency and higher energy content of the fuel. Most of the diesel engines are commonly used in commercial light and heavy duty trucks, buses, railroad locomotives, marine vehicles, all construction and farming equipments, and countless industrial applications. Palm oil methyl ester (POME) is to use biodiesel made from palm oil to run a CI engine without making any changes in engine, but fuel modification is required to improve the major technical problem of higher viscosity, density. These could be effectively eliminated by the combined effects heating biodiesel and blending with diesel before injecting it into combustion chamber. It has been found that the combined preheating biodiesel and blending with diesel showed a remarkable improvement in kinematic viscosity, density and calorific value which dominantly improved injection of biodiesel by contributing to better fuel atomization at the elevated temperature of the biodiesel. Moreover, intake air preheating improves the ignition problem by decreasing the ignition delay time. From the experimental investigation, it is realized that up to 60% blends of preheated POME with diesel could replace diesel in CI engines for getting better performance and would also help in achieving energy economy. It also contributed to reduction in CO, HC emissions of biodiesel than that of the pure diesel while the NO_x emission increased due to increase in combustion temperature. The optimal combinations of operating condition (CR-IP-IT) showed the better engine characteristics with optimum blends of fuel at 90% load condition as compared to reference diesel. The results of present study outline the potential preheated biodiesel blends in various probable applications are discussed as follows:

- From the experimental and thermodynamic analyses, it was noticed that preheated biodiesel blends run in a VCR DI CI engine at optimal combinations of CR 18, 212 bar fuel IP at 27° bTDC of IT of operating conditions showed a better engine performance and emission parameters at 90% engine loads without any base diesel engine parameters modifications. Hence, the higher percentage of preheated palm biodiesel blends with intake air preheating modes can be employed for passenger cars, railroad locomotives, stationary diesel engine applications of farming and construction use, electrical generation and back-up power production, and combined heat and power applications.
- The capability of the palm oil as a practical, modest and effective hotspot for the generation of biodiesel. Palm oil is a readily available and affordable liquid fuel for people in a nation like India. For the farming and power generation purpose, rural people mostly use small or low power single cylinder diesel engines. However, they encounter problems of liquid fuels shortage due to living in remote locations. The preheated blends with intake air preheating mode in a CI engine operation can fit best for this type of problems for the rural need.

10.3 Limitations of the Study

During the execution of the research project, the following limitations are encountered:

- Regulating/maintaining preheating temperature: In order to arrive at the accurate experimental results, there was a requirement to control, maintain and monitor the temperature of fuel and intake air during the conduct of the experiment. The required device, though identified could not be procured because its cost limitation.
- Thermal stability/storage of biodiesel: The integrity of thermal stability of the blended fuel is an essential requirement to mitigate the degradation of biofuel because of its exposure to external environment like sunlight, temperature and some substances present in the biodiesel. Thermal stability provides a fair idea about the sensitivity of biofuel composition to environmental conditions for its storage and utilization as engine fuels. Design of a proper storage container ensure the storage ability of the fuel. In the absence of specifically designed container, normal container was used for storage of test biodiesel.
- Engine Operational Problems: Biodiesel can be blended and used in many different concentrations. The most common blends are, BD5-BD20 (5% to 20% biodiesel). While using high-level blends, many issues should be considered. It requires special handling and may require equipment modifications, to avoid engine operational problems. Pure biodiesel

contains less energy on a volumetric basis than fossil diesel fuel. Hence, the higher the percentage of biodiesel (above 20%), the lower the energy content per kg. High-level biodiesel blends can also impact engine warranties, and may present unique storage issues. BD100 use could also increase nitrogen oxides (NO_x) emissions, although it greatly reduces other toxic emissions (CO and HC).

- Electric vehicle revolution poses threat to biodiesel: Most governments (like, USA, China, and India) are committed to switch all automobiles sales into Electric Vehicle, partly to avoid severe air pollution and reduce the high demands of fossil fuels. The oil consumption will reduce drastically for electric vehicles even from most efficient internal combustion engines. Thus, the demand because of the efficiency of self-driving, or autonomous, cars will cause the amount transportation fuel to drop. Electric vehicles could quickly grow in number in the upcoming years and this can significantly reduce the demand for petroleum fuel as well as biofuel products in the internal combustion engine application.

10.4 Future Work

Although considerable work on the engine characteristics of preheated biodiesel blends has been completed and reported in this investigation, additional research is necessary to further improve the performance of preheated biodiesel blends operated in a VCR DI compression ignition engine. Specifically, the following extensions are suggested as future work for the investigations on the use of preheated biodiesel-diesel blends.

- A detailed CFD analysis for helical coiled heat exchanger, and shell and tube heat exchanger needs to be undertaken for greater understanding and explanation of the results.
- The usage of preheated biodiesel added with additives as an alternative fuel and evaluates its performance for various blends with different loads. Preheated Biodiesel is added with Additives for rapid combustion as for the biodiesel, the cetane number is high that results in shorter delay of ignition and the mixture is preheated to raise its temperature to improve the combustion process.
- The possibility of using combination of waste cooking oil biodiesel with vegetable oil biodiesel may be carried out to obtain its suitability as an alternative to diesel fuel in internal combustion engine.

- Implementing pyrolysis process along with preheating technique to decrease viscosity and density of the neat biodiesel used a fuel in a diesel engine.
- Implementing a combined effects of preheating biodiesel with ethanol-diesel blends in a diesel engine application.
- The preheated palm biodiesel-diesel blends can be tested in a diesel engine as a pilot fuel and gaseous fuel as a main fuel.
- Adopting water spraying NOx emission control techniques to bring the emission results closer to that of diesel operation.
- Durability test is to be done for checking the longevity of the VCR engine fueled with preheated biodiesel-diesel blends.
- Engine parts deposit should be considered for the future study. This is important as it will provide information whether biodiesel PPBD60 has any effect on engine parts when used the fuel for a long time.
- Numerical analysis of different blends (biodiesel-diesel) engine combustion in a variable compression ratio direct injection compression ignition engine can be undertaken by implementing the CFD simulation software using KIVA4-CHEMKIN or CONVERGE code for greater understanding and explanation of the results.

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APPENDIX-A

Equations for Performance and Combustion Analysis

The analytical equations for computing performance and combustion parameters are calculated as follows:

(I) Performance Analysis:

(a) Brake power (bp):
$$bp = \frac{2\pi \times N \times W \times r}{(60 \times 1000)}, (\text{kW}) \quad (\text{A1})$$

Where N , W and r are the speed of the engine (rpm), engine load (kg-m/s^2) and dynamometer arm radius (m), respectively.

(b) Fuel flow rate, $F_1 = (\rho_f \times X \times 10^{-6} \times 60)$, kg/hr (A2)

Where, X is the cc fuel consumption of engine (cc/ min), ρ_f is the density of fuel (kg/m^3).

(c) Brake thermal efficiency (BTHE):

$$\text{BTHE} = (bp \times 3600) / (F_1 \times \text{LHV}_{\text{fuel}}) \times 100, \quad \% \quad (\text{A3})$$

Where, F_1 is fuel mass flow rate (kg/s), and LHV_{fuel} is the lower heating value (kJ/kg).

(d) Brake specific fuel consumption (BSFC):

$$\text{BSFC} = (F_1 / bp), \quad \text{kg/kW.hr} \quad (\text{A4})$$

(e) Air Flow rate (F_2):

$$F_2 = C_d \times \rho_{\text{air}} \times (\pi d_o^2 / 4) \sqrt{2gh (\rho_w / \rho_a)} \times 3600, \quad \text{kg / hr} \quad (\text{A5})$$

Where, C_d , d_o , h are coefficient of discharge of orifice of 0.61, diameter of the orifice of air flow (0.02 m), h is manometer reading across orifice (m), ρ_w , ρ_a are water density (1000 kg/m^3) and the density of ambient air (kg/m^3), respectively. The density of air ($\rho_a = P/RT$), P is atmospheric pressure in kgf/m^2 (1 standard atm = 10332.27 kgf/m^2), R is a gas constant of 29.27 kgf m/kg K , and T is atmospheric temperature of 300 K.

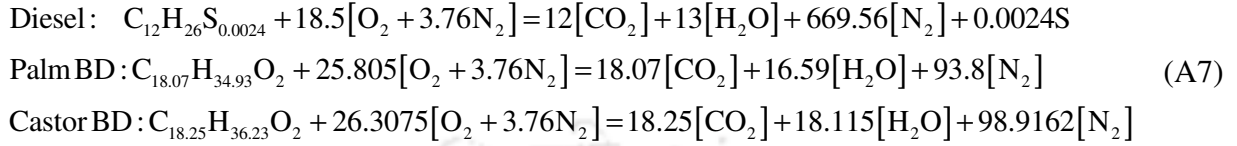
(f) Volumetric efficiency, $\eta_{\text{volumetric}} = F_2 / [(\pi/4) \times B^2 \times l \times (N/n) \times 60 \times k \times \rho_a] \times 100, \quad \% \quad (\text{A6})$

where, B , l , n , and k are engine cylinder diameter (m), engine stroke length (m), number of

revolution per cycle (2 for four stroke engine), number of cylinders, respectively.

(g) Stoichiometric air-fuel ratio of diesel fuel, POME and COME i.e. $\lambda=1$ (Debnath et al., 2013b):

The complete combustion of diesel, POME biodiesel and COME biodiesel with air generates following equations:



II. Combustion Analysis:

(a) Pressure Smoothing:

The DAC can record cylinder pressure variation with each degree of crank angle change. Differentiating the raw pressure data shows a noisy trend between successive values. Therefore, after treatment of these pressure data in the form of smoothing becomes necessary (Stone, 1999). For this the smoothing algorithm for $(2b+1)$ value is used as follows:

$$P_n = \frac{1}{b^2} \left[P_{n-(b-1)} + 2P_{n-(b-2)} + 3P_{n-(b-3)} + \dots + bP_n + \dots + 3P_{n+(b-3)} + 2P_{n+(b-2)} + P_{n+(b-1)} \right] \quad (A8)$$

The terms in Eq. (A9) are only evaluated when the part of the subscript in bracket is not negative.

This is illustrated by simplest case when $b = 2$

$$P_n = \left[(P_{n-1}) + 2(P_n) + (P_{n+1}) \right] / 4 \quad (A9)$$

where, 'P' is the instantaneous pressure data. The above equation is used for smoothing the instantaneous pressure data.

(b) Rate of Pressure Rise: The rate of pressure rise calculated from the smoothed pressure data by using first order finite difference equation with fourth order accuracy (Stone, 1999).

$$dP/d\theta = \left[(P_{n-2}) - 8(P_{n-1}) + 8(P_{n+1}) - (P_{n+2}) \right] / 12(\Delta\theta) \quad (A10)$$

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

(c) Net Heat Release Rate: The crank angle encoder, connected to the engine shaft, detects each degree rotation of the crank for each cycle. Hence, for a particular cycle, a total of 720 data for both cylinder pressure and volume are recorded at each load and CR-IP-IT combination. Therefore, to avoid annoying calculations of huge data, at least three cycles are considered for the purpose of heat release calculations at each load and each CR-IP-IT combination. The equation used for net heat release rate is obtained from the first law analysis by implementing rate of pressure rise and rate of volume change, which is given below.

$$d\theta_n/d\theta = \left[\gamma/(\gamma-1) \times P(dV/d\theta) \right] + \left[1/(\gamma-1) \times V(dP/d\theta) \right] \quad (A11)$$

where, $dQ_n/d\theta$ is the net heat release rate, γ is the ratio of specific heats, P the cylinder pressure, V the instantaneous volume of the cylinder.

The ranges of γ varies from 1.3 to 1.35. However, in this study the value of γ is considered as 1.35. In order to select the constant value of γ a short analysis has been performed as follows. Although combustion in a CI, DI engine is quite heterogeneous, the contents of the combustion chamber are assumed to be homogeneous in the heat release calculations. Some investigators (Hansen, 2013) have developed more complex, quasi-dimensional models, but these are only slightly more accurate than these model. The value of γ varies with temperature and the gas temperature is also needed in calculating heat transfer to the wall. The ideal gas law is used to calculate the spatially averaged temperature in the combustion chamber, i.e.:

$$T_j = (P_j V_j / M R_g), \quad K \quad (A12)$$

Where, T_j is bulk gas temperature at point j (K), R_g is idea gas constant, M is mass of charge. It can be shown that the value of γ can be calculated from the following equation:

$$\gamma = \left[1 - (R_g / C_p) \right]^{-1} \quad (A13)$$

According to Goering (1998), the value of C_p/R_g , can be calculated from:

$$C_p/R_g = a_0 + a_1 T_j + a_2 T_j^2 + a_3 T_j^3 + a_4 T_j^4 \quad (A14)$$

Where, $a_0 = 3.04473$, $a_1 = 1.33805e-3$, $a_2 = -4.88256e-7$, $a_3 = 8.55473e-11$, $a_4 = -5.70132e-15$

The results indicate that, there is negligible difference between the NHRR values obtained for constant and variable γ . Hence throughout the result analysis γ is maintained as 1.35.

(d) Ignition Delay: Ignition delay can be defined as the duration between the start of fuel injection (θ_{IN}) and the crank angle at which combustion starts (θ_{IG}). Therefore,

$$\theta_R = \theta_{IG} - \theta_{IN}, \quad ^\circ CA \quad (A15)$$

The standard IT is obtained from the manufacturer specifications, 23° bTDC. The ITs, other than the standard one are obtained from the fuel pressure data stored during data acquisition. The crank angle at which combustion starts is obtained from the ($dP/d\theta$) diagram as it changes its concavity when combustion starts.

APPENDIX-B

Equations for Thermodynamic and Thermo-economic Potential Study

(I) Thermodynamic analysis

1) Energy Analysis

The first law of thermodynamics states that energy is conserved in every device and process. It deals with the quantitative energy balance in the engine in the form “heat or work”. Thus, energy may exist in many different forms and may be converted from one form to another within a thermodynamic system (Heywood, 1988). While considering energy input, various test samples of fuels are considered along with their calorific values (Caliskan and Mori, 2017). The following assumptions are made; i.e. engine run at steady state, negligible changes in potential and kinetic energy, ideal gas mixture. The entire system of the engine is modelled as a control volume for steady state of the energy balance is shown below;

$$\sum \dot{H}e_{in} = \sum \dot{H}e_{out} \quad (B1)$$

Where, $\sum \dot{H}e_{in}$ and $\sum \dot{H}e_{out}$ are the total input and output energy rates (kW).

In compression ignition (CI) engine, the fuel energy supplied per unit time ($\dot{H}e_{in}$) is transferred in its different processes (**Figure 9.1**), namely, shaft power ($\dot{H}e_{sh}$), energy in cooling water per unit time ($\dot{H}e_{cw}$), energy in exhaust gas per unit time ($\dot{H}e_{exh}$) and uncounted energy losses per unit time ($\dot{H}e_{untd}$) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipments, etc. Hence, the following empirical relation is used as cited in the reference (Caliskan and Mori, 2017).

$$\dot{H}e_{in} = \dot{H}e_{sh} + \dot{H}e_{cw} + \dot{H}e_{exh} + \dot{H}e_{untd} \quad (B2)$$

In the present study, it may be emphasized that the densities and kinematic viscosities of different test fuels varies with percentage fraction of blends at a particular preheating temperature. These different forms of energies are calculated according to the following analytical expressions (Al-Najem and Diab, 1992; Debnath et al., 2013b; Heywood, 1988).

(a) The input energy ($\dot{H}_{e_{in}}$) to the diesel engine is the amount of fuel energy content in the supplied diesel fuel and it is given by,

$$\dot{H}_{e_{in}} = F_1 \times \text{LHV}_{\text{fuel}}, \text{ (kJ/hr)} \quad (\text{B3})$$

The lower heating value (LHV_{fuel}) of different fractions of biodiesel blend fuels can be computed;

$$\text{LHV}_{\text{fuel}} = \left\{ \left[246.25 - 0.167 \rho_{\text{fuel}} - 12.88 \times \ln(v_{\text{fuel}}) \right] \times 10^3 / 2 \right\}, \text{ kJ/kg} \quad (\text{B4})$$

Where, ρ_{fuel} , v_{fuel} is fuel density (kg/m^3) and the kinematic viscosity of fuel (mm^2/s), respectively.

(b) Shaft power ($\dot{H}_{e_{in}}$): The heat equivalent to useful brake power in kW, is

$$\dot{H}_{e_{sh}} = bp \times 3600, \text{ kJ/hr} \quad (\text{B5})$$

(c) Energy in cooling water per unit time (\dot{H}_{cw}): The heat loss from the engine block to the cooling water in (kJ/hr), is given by,

$$\dot{H}_{cw} = F_3 \times C_{p_{cw}} (T_{cw,oe} - T_{cw,ie}), \text{ kJ/hr} \quad (\text{B6})$$

Where, F_3 is the mass flow rate of cooling water in kg/s passing through engine jacket. $C_{p_{cw}}$ is the specific heat of water in kJ/kg K. $T_{cw,ie}$ and $T_{cw,oe}$ are the inlet and outlet temperature of cooling water passing through engine jacket in K.

(d) Energy in exhaust gas per unit time (\dot{H}_{exh}): The energy carried away by exhaust gases in (kJ/hr), is evaluated by,

$$\dot{H}_{exh} = \dot{m}_{exh} \times C_{p_{exh}} (T_{exh,ic} - T_{atm}), \text{ kJ/hr} \quad (\text{B7})$$

Where, $\dot{m}_{exh} = \dot{m}_{fuel} + \dot{m}_a$ is the mass flow rate of exhaust in kg/s, \dot{m}_a is the mass flow rate air in kg/s. $C_{p_{exh}}$ is the specific heat of exhaust gas in kJ/kg K is obtained from the energy balance of the flows passing through the calorimeter as follows:

$$C_{p_{exh}} = \left[F_3 \times C_{p_{cw}} (T_{cw,oe} - T_{cw,ie}) \right] / \left[(\dot{m}_{fuel} + \dot{m}_a) (T_{exh,ic} - T_{exh,oc}) \right], \text{ kJ/kgK} \quad (\text{B8})$$

Where F_3 is the mass flow rate cooling water, and $T_{exh,ic}$ and $T_{exh,oc}$ are the inlet and outlet temperatures of exhaust gas passing through calorimeter.

(e) Uncounted energy losses per unit time (\dot{H}_{untd}): The amount of the uncounted losses in (kJ/hr), is determined by performing an energy balance,

$$\dot{H}_{untd} = \left[\dot{H}_{e_{in}} - (\dot{H}_{e_{sh}} + \dot{H}_{e_{cw}} + \dot{H}_{e_{exh}}) \right], \text{ kJ/hr} \quad (\text{B9})$$

(f) Thermal efficiency of the engine (energy percentage), is usually determined as the ratio of the power output (network) to the fuel energy input, expressed as follows:

$$\eta_{th} = \frac{\dot{H}e_{sh}}{\dot{H}e_{in}} \times 100, \% \quad (B10)$$

2) Exergy Analysis

The quality of the fuel is very essential to evaluate the merit of each test fuel under study while evaluating their performance in the engine. The second law analysis called, exergy or availability analysis. The availability can be described as the ability to perform useful amount of work by the supplied energy (Heywood, 1988). The exergy balance for steady-state control volume (**Figure 9.1**) is given (Caliskan and Mori, 2017);

$$\sum \dot{E}x_{in} = \sum \dot{E}x_{out} + \sum \dot{E}x_{destr} \quad (B11)$$

Where, $\sum \dot{E}x_{in}$, $\sum \dot{E}x_{out}$, and $\sum \dot{E}x_{destr}$ are the summation of input exergy rate, output exergy rate, and exergy destroyed rate (kW), correspondingly.

In the compression ignition engine, the exergy rate of fuel ($\dot{E}x_{in}$) supplied is converted into different types of exergy rates and can be described as namely, useful exergy ($\dot{E}x_{sh}$), exergy of cooling water ($\dot{E}x_{cw}$), exergy of exhaust gas ($\dot{E}x_{exh}$) and destroyed exergy ($\dot{E}x_{destr}$) in the form of friction, radiation, heat transfer to the surrounding, operating auxiliary equipment's, etc.

$$\dot{E}x_{in} = \dot{E}x_{sh} + \dot{E}x_{cw} + \dot{E}x_{exh} + \dot{E}x_{destr} \quad (B12)$$

The amount of each of these exergy transfers on the basis of the second law of thermodynamics are calculated according to the following analytical expressions as described in the literature (Debnath et al., 2013b).

(a) Input exergy rate ($\dot{E}x_{in}$): It is calculated according to the following analytical expressions as described in the literature (Flynn et al., 1984);

$$\dot{E}x_{in} = F_1 \times LHV_{fuel} \times \epsilon_{fuel}, kW \quad (B13)$$

Where, ϵ_{fuel} is a chemical exergy factor of fuel and “h, c, o and s” are the percentage composition of hydrogen, carbon, oxygen and sulphur of the fuel.

$$\epsilon_{fuel} = [1.0401 + 0.1728(h/c) + 0.0432(o/c) + 0.2169(s/c)\{1 - 2.0628(h/c)\}] \quad (B14)$$

where, h , c , o and s are the mass fractions of hydrogen, carbon, oxygen and sulfur contents (Lyn, 1963).

(b) Useful exergy rate ($\dot{E}x_{sh}$) in kW is;

$$\dot{E}x_{sh} = \dot{H}e_{sh} \times 1/3600, \text{ kW} \quad (\text{B15})$$

(c) Exergy rate of cooling water ($\dot{E}x_{cw}$) in kW is;

$$\dot{E}x_{cw} = \frac{\dot{H}e_{cw}}{3600} - \left[F_3 \times C_{p_{cw}} \times T_{amb} \times \ln(T_{cw,oe}/T_{cw,ie}) \right], \text{ kW} \quad (\text{B16})$$

(d) Exergy rate of exhaust gas ($\dot{E}x_{exh}$):

$$\dot{E}x_{exh} = \frac{\dot{H}e_{exh}}{3600} + \left[(F_1 + F_2) \times T_{amb} \times \left\{ C_{p_{exh}} \times \ln(T_{amb}/T_{exh,ic}) - R_{exh} \ln(P_{amb}/P_{exh,ic}) \right\} \right], \text{ kW} \quad (\text{B17})$$

where, R_{exh} is the specific gas constant of the exhaust gas in kJ/kg K. P_{amb} is the ambient pressure, N/m². $P_{exh,ic}$ is the final pressure, N/m², T_{amb} is the ambient temperature in K.

(e) Destroyed exergy rates ($\dot{E}x_{destr}$) in kW is;

$$\dot{E}x_{destr} = \dot{E}x_{in} - (\dot{E}x_{sh} + \dot{E}x_{cw} + \dot{E}x_{exh}), \text{ kW} \quad (\text{B18})$$

(f) Exergy efficiency (%) is the ratio of total availability recovered from the system to the total availability input into the system ($\dot{E}x_{sh}$, $\dot{E}x_{cw}$ and $\dot{E}x_{exh}$). Therefore,

$$\eta_{exergy} = \frac{\dot{E}x_{sh}}{\dot{E}x_{in}} \times 100, \% \quad (\text{B19})$$

(g) The entropy generation rate; $S_{gen} = \dot{E}x_{destr}/T_{amb}$, kW/K (B20)

(h) sustainability index (SI) is defined from exergetic efficiency as given by:

$$SI = 1/(1 - \eta_{exergy}) \quad (\text{B21})$$

(II) Thermo-economic analysis

It is the exergy based economic analysis in which exergy costs are distributed among outputs (Meisami et al., 2018). In this analysis, relation between different exergy streams such as, fuel input, useful work, cooling water, exhaust gases, destruction and capital investments are included. The exergy balances in the control volume of compression ignition engine as shown in **Figure 9.2**, can be modelled by;

$$\begin{aligned}\dot{E}x_{ht} + \sum \dot{m}_i e_i &= \dot{E}x_{sh} + \sum \dot{m}_e e_e + \dot{E}x_{destr} \\ \text{Here, } \sum \dot{m}_i e_i &= \dot{E}x_{in} \\ \sum \dot{m}_e e_e &= \dot{E}x_{exh}\end{aligned}\quad (B22)$$

The economic balance the control volume of CI engine as shown in **Figure 9.2**, can be modelled by;

$$\begin{aligned}\sum \dot{C}_e + \dot{C}_{sh} &= \dot{C}_{ht} + \sum \dot{C}_i + \dot{Z} \\ \text{Here, } \dot{C}_i &= c_{in} \dot{E}x_{in} = c_{in} (\dot{E}x_{fuel} + \dot{E}x_{air}) \\ \dot{C}_e &= c_{exit} \dot{E}x_{exit} = c_{exh} \dot{E}x_{exh} \\ \dot{Z} &= Z_{engc} + Z_{omc}\end{aligned}\quad (B23)$$

Here, \dot{C}_{sh} and \dot{C}_{ht} are the cost rate exergy of effective power and heat transfer (cooling water), and \dot{Z} is the capital investment, \dot{C}_i and \dot{C}_e are the cost rate of inlet fuel and outlet exhaust gases streams. In the present study, it is assumed that the cost rate associated with the inlet air induced into the engine is neglected, the thermo-economic balance equation can be defined as follows:

$$\begin{aligned}\dot{C}_{sh} + \dot{C}_{exh} &= \dot{C}_{ht} + \dot{C}_{fuel} + \dot{C}_{destr} + Z_{engc} + Z_{omc} \\ \text{Here, } \dot{C}_{sh} &= c_{sh} \dot{Q}_{sh} \\ \dot{C}_{exh} &= c_{exh} \dot{E}x_{exh} \\ \dot{C}_{ht} &= c_{ht} \dot{E}x_{ht} \\ \dot{C}_{fuel} &= c_{fuel} \dot{E}x_{fuel} \\ \dot{C}_d &= c_{destr} \dot{E}x_{destr}\end{aligned}\quad (B24)$$

Here, \dot{C}_{fuel} , \dot{C}_{sh} , \dot{C}_{ht} , \dot{C}_{exh} and \dot{C}_{destr} are the cost rate related with the input exergy, useful exergy, exhaust gas, heat loss, and destruction exergy, respectively in \$/hr. The cost balance applying to the components presented in **Figure 9.2**, has been stated by Eq. (B25). The terms involve the sum of cost rates linked with all exiting useful exergy stream (\dot{C}_{sh}) and all entering exergy streams (\dot{C}_{fuel}) plus the cost rates because of the investment of the compression ignition engine (Z_{engc}), and operating and maintenance expense (Z_{omc}). Thus, the cost balance equation for the engine can be rewritten as follows;

$$c_{\text{exh}} \dot{E}x_{\text{exh}} + c_{\text{sh}} \dot{E}x_{\text{sh}} = c_{\text{ht}} \dot{E}x_{\text{ht}} + c_{\text{fuel}} \dot{E}x_{\text{fuel}} + c_{\text{d}} \dot{E}x_{\text{d}} + Z_{\text{engc}} + Z_{\text{omc}} \quad (\text{B25})$$

Here, c_{fuel} , c_{sh} , c_{exh} and c_{destr} are cost per unit exergy of inlet fuel, useful brake power, heat loss, exhaust gas, and destruction, respectively. The cost rate associated exhaust gases loss ($c_{\text{exh}} \dot{E}x_{\text{exh}}$) and heat loss due to cooling water ($c_{\text{ht}} \dot{E}x_{\text{ht}}$) were neglected. Since, the inlet fuel represents only resources, which are expended for generating useful brake power, the cost of heat loss and destruction can be correlated in terms of fuel costs i.e. $c_{\text{fuel}} = c_{\text{loss}} = c_{\text{destr}}$. Thus, the cost rates associated with different blends of fuel are finally expressed as follows:

$$c_{\text{sh}} \dot{E}x_{\text{sh}} = c_{\text{fuel}} \dot{E}x_{\text{fuel}} + c_{\text{fuel}} \dot{E}x_{\text{d}} + Z_{\text{engc}} + Z_{\text{omc}} \quad (\text{B26})$$

Hence, the basic equation for the cost per useful power exergy and exergy destruction can be interpreted from Eq. (B27) as below;

$$c_{\text{sh}} = \frac{(c_{\text{fuel}} \dot{E}x_{\text{fuel}} + c_{\text{fuel}} \dot{E}x_{\text{destr}} + Z_{\text{engc}} + Z_{\text{omc}})}{\dot{E}x_{\text{sh}}}, \text{ \$ / MJ}$$

$$c_{\text{destr}} = \frac{(c_{\text{sh}} \dot{E}x_{\text{sh}} + c_{\text{fuel}} \dot{E}x_{\text{fuel}} + Z_{\text{engc}} + Z_{\text{omc}})}{\dot{E}x_{\text{d}}}, \text{ \$ / MJ} \quad (\text{B27})$$

Exhaust gas recirculation system with engine have been developed for testing and measuring the parameters of selected test fuels to calculate performance characteristics and emission characteristics with use of this EGR system. The EGR system is developed to reduce the amount of oxides of nitrogen (NO_x) created by the engine during operating periods that usually results in high combustion temperatures. EGR system consist of the exhaust gas that is recirculated back in to the intake manifold where it mixes with air and gets diluted with intake charge which in turn act as a diluents and reduces peak combustion temperature. The R_{EGR} (%) is evaluated based the following relation (Agarwal et al., 2011). It is defined as the mass percent of the recirculated exhaust (\dot{m}_{EGR}) in the total intake mixture (\dot{m}_i).

$$R_{EGR} (\%) = \frac{\dot{m}_{EGR}}{\dot{m}_i} \times 100 \quad (C1)$$

$$\dot{m}_{EGR} = C_d \times \left(\frac{\pi}{4} \times d_{exo}^2 \right) \times \rho_{ex} \times \left(\sqrt{2 \times 9.81 \times \left(\frac{\rho_w}{\rho_{ex}} \right) \times h_{ex}} \right) \times 3600 \quad (C2)$$

$$\dot{m}_i = \dot{m}_a + \dot{m}_{EGR} + \dot{m}_{fuel} \quad (C3)$$

Where,

R_{EGR} = Percentage Mass of EGR being recirculated,

C_d = Coefficient of discharge of orifice, 0.61,

d_{exo} = Diameter of orifice for exhaust gas flow, 0.02m

ρ_{ex} = Density of exhaust gas, kg/m³,

ρ_w = Density of water, 1000 kg/m³,

h_{ex} = height of manometer water level difference, m

APPENDIX-D

Experimental Uncertainties

The uncertainties of the parameters are calculated by using the universally known theory of sequential perturbation technique brought by Kline and McClintock (1953). The method is described as follows. If N is a dependent measuring parameter which is a function of the independent variables $x_1, x_2, x_3, x_4, \dots, x_n$. Therefore,

$$N = N(x_1, x_2, \dots, x_n) \quad (D1)$$

If ΔN is the uncertainty created due to the individual uncertainties of the independent parameters termed as $\Delta N_1, \Delta N_2, \Delta N_3, \Delta N_4, \dots, \Delta N_n$. Then the uncertainty of the dependent variable can be written as,

$$\Delta N = \left[\left(\frac{\partial N}{\partial x_1} \Delta N_1 \right)^2 + \left(\frac{\partial N}{\partial x_2} \Delta N_2 \right)^2 + \dots + \left(\frac{\partial N}{\partial x_n} \Delta N_n \right)^2 \right]^{1/2} \quad (D2)$$

The estimated relative errors of each measured independent parameters for both pure diesel and PPBD20 fuel modes are summarized in

Table D.1 in Appendix D. For each performance parameters the overall uncertainties measurements error is established in **Table D.2**. The overall measurement uncertainties of emission constituents are tabulated in **Table D.3**.

(I) Uncertainty Calculations

1) Uncertainties in Performance Parameters

The uncertainty calculation shown here are carried out correspond to CR 17.5, IP 200bar, IT 23° bTDC for both fuels fuel (diesel and PPBD20). The uncertainties of the dependent performance parameters are described below. A sample calculation provided here for measurement of uncertainty. The full load (100%) is taken as 12 kg and low load (20%) is taken as 2.4 kg.

(a) Uncertainty in brake power

In the above equation N and W are independent variable, because r has a constant value. The uncertainties of N and W , are 3.33%, 0.83% (full load) and 0.42% (20% load) respectively (for PPBD20 blend fuel).

The bp of the engine is given by (A1), $bp = (2\pi N \times W \times r) / (60 \times 1000)$

$$\begin{aligned} \text{At 12 kg loading condition : } \Delta bp/bp &= \sqrt{(\Delta N/N)^2 + (\Delta W/W)^2} \\ \Delta bp/bp &= \sqrt{(50/1500)^2 + (0.1/12)^2} = 3.43\% \end{aligned} \quad (D3)$$

$$\text{At 2.4 kg loading condition : } \Delta bp/bp = \sqrt{(50/1500)^2 + (0.1/2.4)^2} = 5.33\%$$

(b) Uncertainty in BSFC. The BSFC of the engine is given by (A4), $BSFC = (F_1/bp) \times 3600$

$$\begin{aligned} \text{At 12 kg loading condition : } \Delta BSFC/BSFC &= \sqrt{(\Delta N/N)^2 + (\Delta W/W)^2 + (\Delta \dot{m}_f/\dot{m}_f)^2} \\ \Delta BSFC/BSFC &= \sqrt{(50/1500)^2 + (0.1/12)^2 + (0.01/1.01)^2} \\ &= 3.57\% \end{aligned} \quad (D4)$$

$$\begin{aligned} \text{At 2.4 kg loading condition : } \Delta BSFC/BSFC &= \sqrt{(50/1500)^2 + (0.1/2.4)^2 + (0.01/1.01)^2} \\ &= 5.43\% \end{aligned}$$

Here, the independent variables are N, W, \dot{m}_{fuel} .

The error in measurement of N, W, \dot{m}_{fuel} are 3.33%,

(c) Uncertainty in BTHE. It is given by (A3), $BTHE = bp/(F_1 \times LHV_{fuel}) \times 100$

At 12 kg loading condition :

$$\begin{aligned} \Delta BTHE/BTHE &= \sqrt{(\Delta N/N)^2 + (\Delta W/W)^2 + (\Delta \dot{m}_f/\dot{m}_f)^2 + (\Delta LHV_{fuel})^2} \\ \Delta BTHE/BTHE &= \sqrt{(50/1500)^2 + (0.1/12)^2 + (0.01/1.01)^2 + (0.011)^2} = 3.74\% \end{aligned} \quad (D5)$$

At 2.4 kg loading condition :

$$\Delta BTHE/BTHE = \sqrt{(50/1500)^2 + (0.1/2.4)^2 + (0.01/1.01)^2 + (0.011)^2} = 5.54\%$$

Similarly, uncertainty volumetric efficiency and other dependent parameters are calculated.

2) Uncertainty in Emission Constituents

Uncertainty in emission constituents are obtained by computing its Resolution/Range.

$$\begin{aligned} \Delta CO/CO &= (\text{Resolution/range})_{CO} \times 100 \\ \Delta CO_2/CO_2 &= (\text{Resolution/range})_{CO_2} \times 100 \\ \Delta HC/HC &= (\text{Resolution/range})_{HC} \times 100 \\ \Delta NO_x/NO_x &= (\text{Resolution/range})_{NO_x} \times 100 \end{aligned} \quad (D6)$$

3) Total Percentage of Uncertainty

Using the principle of propagation of errors, the total percentage of the uncertainty of the

experimental trail can be computed:

$$\text{TPU} = \text{Square root of} \left[\begin{array}{l} (\text{uncertainty of BP})^2 + (\text{uncertainty of BSFC})^2 + (\text{uncertainty of BTHE})^2 + \\ (\text{uncertainty of CO})^2 + (\text{uncertainty of CO}_2)^2 + (\text{uncertainty of HC})^2 + \\ (\text{uncertainty of NO}_x)^2 \end{array} \right] \quad (\text{D7})$$

Table D.1: The independent variables and its error in measurement.

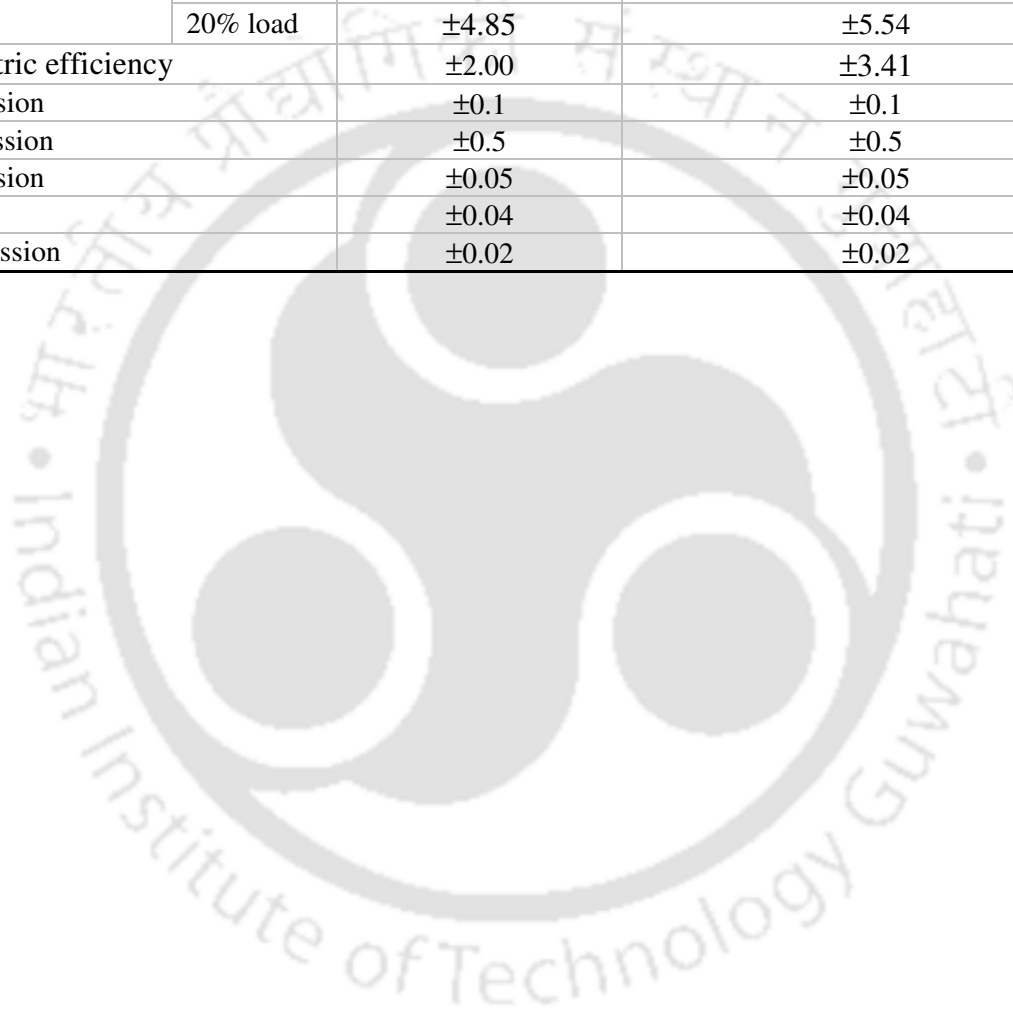
S.N	Independent variable	Measuring ranges	accuracy	Relative errors (%)
1	Engine speed	1500 rpm	±50 rpm	±3.33
2	Engine load	Full load	±0.1 kg	±0.83
		20% load	±0.01 kg	±0.42
3	Liquid fuel flow rate	1.01 kg/hr.	±0.01 kg.hr	±0.99
	Air flow rate	23.11kg/hr.	±0.01 kg.hr	±0.99
5	LHV of liquid fuel	-	-	± 1.0
6	Temperature	0 – 900 °C	±1°C	± 0.11
7	Cylinder pressure	0 -110 bar	±1 bar	±1.0
9	Burette fuel measurement	10 cc	±0.1cc	±1.0
10	Time	50 s	±0.1s	±0.2
11	Manometer	100 mm	±1mm	±1.0
12	Crank angle encoder-crank angle	0–360°	±1°	±1°

Table D.2: The emission variables and its error in measurement.

S.N	Measured parameter	Measuring ranges	Resolution	Relative errors (%)
1	CO emission	0–10 % vol.	± 0.01% vol.	±0.1
2	CO ₂ emission	0–20 % vol.	± 0.1% vol.	±0.5
3	HC emission	0–2000 ppm	±1 ppm	±0.05
4	O ₂	0-22 % vol.	± 0.01% vol.	±0.04
5	NO _x emission	0–5000 ppm	±1 ppm	±0.02

Table D.3: Overall measurement uncertainty for performance and emission parameters.

Performance parameter		Diesel mode uncertainty (%)	Preheated Blend mode (PPBD20) uncertainty (%)
bp	Full load	±2.16	±3.43
	20% load	±4.62	±5.33
BSFC	Full load	±2.38	±3.57
	20% load	±4.73	±5.43
BTHE	Full load	±2.62	±3.74
	20% load	±4.85	±5.54
Volumetric efficiency		±2.00	±3.41
CO emission		±0.1	±0.1
CO ₂ emission		±0.5	±0.5
HC emission		±0.05	±0.05
O ₂		±0.04	±0.04
NO _x emission		±0.02	±0.02



APPENDIX-E

Design and Dimensioning of Fuel and Intake Air Preheater

In this study, two different preheaters are integrated with the test engine setup – one for heating biodiesel and the other for heating the intake air prior to fuel-air mixing and subsequent entry to the combustion chamber. Both the heat exchangers are indigenously designed with an assumption to meet the requirements with specific intended purpose. In the present study, a recuperator type of heat exchanger with counter flow arrangement, is chosen for which the heat transfer occurs between two fluid streams separated by a heat transfer surface. The most common problems of in heat exchanger design are thermal analysis (i.e. rating and sizing calculation).

(I) Rating Calculation

The “rating problem” is concerned with determination of the heat transfer rate and the fluid temperatures based on Kern (1950), that involves “trial and error calculations” by using Equation (E1).

- (a) Heat transfer rate: The actual heat transfer rate, q can be determined by energy balance equation:

$$q = \dot{m}_h C_{p_h} (T_{in} - T_{out})_h = \dot{m}_c C_{p_c} (T_{out} - T_{in})_c \quad (E1)$$

- (b) Heat carried by hot fluid (exhaust gases);

$$q_h = \dot{m}_h C_{p_h} (T_{in} - T_{out})_h \quad (E2)$$

- (c) Heat carried by cold fluid (biodiesel);

$$q_c = \dot{m}_c C_{p_c} (T_{out} - T_{in})_c \quad (E3)$$

- (d) The average of two reading q_{avg} ,

$$q_{avg} = (q_h + q_c) / 2 \quad (E4)$$

- (e) Heat Exchanger effectiveness (ϵ)

$$\epsilon = q_{avg} / q_{max} \quad (E5)$$

Where,

$$q_{max} = \dot{m}_{min} C_{p_{min}} (T_{in,h} - T_{in,c}) \quad (E6)$$

- (f) Log mean temperature difference (ΔT_{LM})

$$(\Delta T_{LM})_{cc} = (\Delta T_1 - \Delta T_2) / \ln(\Delta T_1 / \Delta T_2) \quad (E7)$$

Where, $\Delta T_1 = (T_{in,h} - T_{out,c})$ and $\Delta T_2 = (T_{out,h} - T_{in,c})$.

(g) The overall heat transfer coefficient, U_o , is calculated from the temperature data and the flow rates using the following equations (Bergman et al., 2011):

$$\begin{aligned} U_o &= (q/A_o \Delta T_{LM}) \\ \Rightarrow A_o &= \pi d_o L \end{aligned} \quad (E8)$$

Where, A_o is the outside surface area of coiled tube, q is the heat transfer rate and ΔT_{LM} is the log mean temperature difference, based on inlet temperature difference, ΔT_1 , and the outlet temperature difference, ΔT_2 , using the following equation (Bergman et al., 2011):

(II) Sizing Calculation

The “sizing calculation” focuses on the evaluation of the dimensions of the two heat exchangers (Hx1 as shown in **Figure 3.13**). For designing this heat exchanger, the important parameters of coil and shell geometry such as geometric effects of curvature ratio and pitch ratio, are considered. All the geometrical details are listed in **Table E.1**. The important design calculations are governed by lists of equations in Equation (E9–E13) as described in the references (Patil et al., 1982; Salimpour, 2009).

1) Design procedure for helical coiled heat exchanger (Fuel Preheater)

$$\begin{aligned} L_c \text{ (m)} &= N_{ct} \sqrt{[(\pi D_c)^2] + P^2} \\ D_h &= \left[D_o^2 - \pi D_c d_o^2 (b/\pi D_c)^{-1} \right] / \left[D_o + \pi D_c d_o^2 (b/\pi D_c)^{-1} \right] \\ V_c \text{ (m}^3\text{)} &= \pi/4 (d_o)^2 L_c \\ V_s \text{ (m}^3\text{)} &= \pi/4 (D_o^2 - D_i^2) \gamma \times N_{ct} \\ V_f \text{ (m}^3\text{)} &= V_s - V_c \\ Re_i &= \rho v_i d_i / \mu; \quad Re_o = \rho v_o D_h / \mu \\ De &= Re_i \sqrt{(d_i/D_c)}; \quad He = De \sqrt{(1 + \gamma^2)} \\ V_i &= \dot{m}_b / \rho_b A_i \\ h_i &= C \times V_i^2 \end{aligned} \quad (E9)$$

Where, The length of coil, L_c needed to make N turns, Hydraulic diameter (D_h), Volume occupied by the coil (V_c), Volume of the shell (exhaust gases) for the same length (V_s), Volume available for flow of exhaust gas (V_f), Reynolds Number (Re_i), dean number (De), helical number (He),

velocity of biodiesel inside tube (V_i). Here, \dot{m}_b is the mass flow rate of biodiesel, ρ_b is the density at coil side fluid temperature (biodiesel) $[(T_{in,c} + T_{out,c})/2]$, $A_i = (\pi/4)d_i^2$. convective heat transfer coefficients of coil side fluid (h_i), here C is constant, obtained from V_i and R_{ov} , graph $y = mx + c$, n is constant which is approximate ($n = 0.8$).

$$\begin{aligned}
 R_{ov} &= 1/U_o A_o ; C = (1/m) A_i \\
 (1/h_o) &= (1/U_o) - \left\{ (d_o/d_i h_i) + [d_o \ln(d_o/d_i)/2\pi k_c L_c] \right\} \\
 Nu_i &= (h_i d_i / k_b), Nu_o = (h_o D_h / k_{exh}) \\
 N_{act} &= A_o / (\pi d_o L / N_{ct}) \\
 \Delta P &= (2f L \rho_b V_i^2) / d_i \\
 \text{Here, } f &= (7.2 / Re_i^{0.5}) \times (d_i / D_c)^{0.25}
 \end{aligned} \tag{E10}$$

Where, heat transfer coefficients (h_o) and is obtained from overall heat transfer coefficient (U_o), Nusselt number (Nu_i), actual number of turns (N_{act}), pressure drop from coiled side fluid (ΔP), and f is friction factor.

2) Design procedure for shell and tube heat exchanger (intake air Preheater)

In case of intake air preheating, a simple “counter flow, single pass, shell and tube heat exchanger (Hx2 as shown in **Figure 3.13**)” is preferred to extract the maximum amount of heat from a given waste heat exhaust gas. Same materials are used (i.e. mild steel for shell part and copper tube) and the design calculations are based on the lists of equations in Equation (E11-E13). The following parameters (**Table E.1**) are assumed such as tube diameter and length, fouling factor based on inside and outside tubes, fixed baffle cut of 25%, three known temperatures (ambient temperature; inlet and exit temperature of exhaust gas), overall heat transfer coefficient etc.

$$\begin{aligned}
 N_t &= A / (\pi d_o \times L) \\
 P_t &= 1.25 d_o \\
 L_B &= 0.4 D_s ; N_B = [L_s / (L_B + t_w)] - 1 \\
 D_e \text{ (square pattern)} &= 4(P_t^2 - \pi d_o^2 / 4) / \pi d_o
 \end{aligned} \tag{E11}$$

Where, Number tubes (N_t), Pitch (P_t), Baffle length (L_B) and number of baffles (N_B), Equivalent diameter for square pitch (D_e).

Shell side fluid

$$\begin{aligned}
 \text{Mass flow rate; } M_s &= \dot{m}_h / (P_t - d_o) L_B D_s / P_t \\
 \text{Reynolds number; } Re_s &= D_e M_s / \mu \\
 \text{prandtl number; } Pr &= c \mu / k \\
 \text{Heat transfer coefficient; } h_o D_e / k &= 0.36 Re_s^{0.55} \cdot Pr^{0.33} (\mu / \mu_w)^{0.14} \\
 \text{Friction factor; } f (Re_s > 500) &= 1.728 Re_s^{-0.188} \\
 \text{Pressure drop; } \Delta p_s &= f \left[(N_B + 1) D_s / D_e (M_s^2 / 2 \rho_s) (\mu / \mu_w)^{0.14} \right]
 \end{aligned} \tag{E12}$$

Tube side fluid

$$\begin{aligned}
 \text{Mass flow rate; } M_t &= \dot{m}_c / (N_t \pi d_i^2 / 4 n_t) \\
 \text{Reynolds number; } Re_i &= d_i M_t / \mu \\
 \text{Prandtl number; } Pr &= c \mu / k \\
 \text{Heat transfer coefficient; } h_i (\text{La min ar flow}) &= 1.86 (Re_i \cdot Pr \cdot d_i / L)^{0.33} (\mu / \mu_w)^{0.14} \\
 h_{io} &= h_i d_i / d_o \\
 1/h_o &= 1/U_o - [(1/h_{io}) + R_s + R_t]^{-1} \\
 \text{Friction factor; } f (\text{La min ar flow}) &= 16 / Re_i \\
 \text{Pressure drop; } \Delta p_t &= 4 (f \times n_t) (L / d_i) (M_t^2 / 2 \rho_t) (\mu / \mu_w)^{0.25} \\
 \Delta p_r &= 4 n_t (M_t^2 / 2 \rho_t); \Delta p_T = \Delta p_t + \Delta p_r
 \end{aligned} \tag{E13}$$

Table E.1: Dimensional parameters of Helical coil, and shell and tube heat exchanger.

Fuel Preheater		Intake Air Preheater	
Heat exchanger type	Helical coiled counter flow	Shell and tube counter flow	
Parameters	Dimension	Parameters	Dimension
Shell length, L	284.0 mm	Shell outer Diameter, D _o	140 mm
Shell outer diameter, D _o	160.0 mm	Shell wall thickness, s _w	3.0 mm
Shell wall thickness, s _w	3.0 mm	Tube outer diameter, d _o	28.58 mm
Coil tube outer diameter, d _o	12.7 mm	Tube wall thickness, t _w	1.63 mm
Coil tube wall thickness	1.2446 mm	Length of tube, L _t	460 mm
Length of coil tube, L _c	4373 mm	Number of tubes, N _t	5 no's
Number of turns in coil, N _{ct}	12 (turns)	Number of Baffles, N _B	6 no's
Number of Baffles, N _B	04	Baffle space, L _B	64 mm
baffle space, L _B	56 mm	Baffle Cut, B _c	40.2 mm
Curvature radius, R _c	58 mm	Pitch of the Tubes, P _t	35.72 mm
Coil pitch, c	24 mm	No. of Shell passes, n _s	01 no's
Curvature ratio, δ	0.088	No. of tube passes, n _t	01 no's
Pitch ratio, γ	0.066		

Table E.2: Operating parameters range of helical coil, and shell and tube heat exchangers.

S.N	Parameters	(Fuel Preheater)		(Intake air Preheater)	
		Cold fluid (biodiesel)	Hot fluid (exhaust gases)	Cold fluid (intake air)	Hot fluid (exhaust gases)
1	Mass flow rate, (kg/hr)	22.58 (calculated)	109.08 (measured)	26.23 (calculated)	127.65 (measured)
2	Inlet temperature, °C	30	118 (measured)	33	108
3	Outlet temperature, °C	75 (measured)	80 (calculated)	61	38

Table E.3: Properties of exhaust gases and biodiesel fluid.

S.N	Properties	(Fuel Preheater)		(Intake air Preheater)		
		Working fluid	Cold fluid (biodiesel)	Hot fluid (exhaust gases)	Cold fluid (intake air) at 30°C	Hot fluid (exhaust gases)
1	Specific heat (Cp), kJ/kg. K		4.69	1.15	1.007	1.15
2	Kinematic viscosity (ν), m ² /s		8.71 x 10 ⁻⁶	0.0083	1.608 x 10 ⁻⁵	0.0083
3	Dynamic viscosity (μ), kg/m.s		124.956 kg/mh	2.12417 x 10 ⁻⁵	1.872 x 10 ⁻⁵	19.002*10 ⁻⁶
4	Density (ρ), kg/m ³		943 at 27 °C	1.008 at 114 °C	1.164	1.0156 at 61 °C
5	Thermal conductivity (k), W/m.K		400.26	0.024	0.02588	0.02872
6	Fouling factor (R), m ² K/W		0.00053	0.001776	-	-

Table E.4: Optimized heat exchanger technical specification (fuel and intake air preheaters).

Fuel Preheater		Intake Air Preheater	
Heat exchanger type	Helical coiled counter flow	Heat exchanger type	Shell and tube counter flow
Parameters	Dimension	Parameters	Dimension
Shell length, L	284.0 mm	Shell outer Diameter, D _o	140 mm
Shell outer diameter, D _o	160.0 mm	Shell wall thickness, s _w	3.0 mm
Shell wall thickness, s _w	3.0 mm	Tube outer diameter, d _o	28.58 mm
Coil tube outer diameter, d _o	12.7 mm	Tube wall thickness, t _w	1.63 mm
Coil tube wall thickness	1.2446 mm	Length of tube, L _t	460 mm
Length of coil tube, L _c	4373 mm	Number of tubes, N _t	5 no's
Number of turns in coil, N _{ct}	12 (turns)	Number of Baffles, N _B	6 no's
Number of Baffles, N _B	04	Baffle space, L _B	64 mm
baffle space, L _B	56 mm	Baffle Cut, B _c	40.2 mm
Curvature radius, R _c	58 mm	Pitch of the Tubes, P _t	35.72 mm
Coil pitch, c	24 mm	No. of Shell passes, n _s	01 no's
Curvature ratio, δ	0.088	No. of tube passes, n _t	01 no's
Pitch ratio, γ	0.066	-	-

APPENDIX-F

List of Publication

Journals

1. Mekonen, M. W., and Sahoo, N. (2018). Effect of fuel preheating with blended fuels and exhaust gas recirculation on diesel engine operating parameters. *Renewable Energy Focus*, Vol. 26, pp. 58-70.
2. Walle Mekonen, M., and Sahoo, N. (2018). Combined effects of fuel and intake air preheating for improving diesel engine operating parameters running with biodiesel blends. *Journal of Renewable and Sustainable Energy*, Vol. 10, No. 4, 043103.
3. Walle Mekonen, M., and Sahoo, N. (2018). Combined effects of fuel injection pressure and injector nozzle holes on the performance of preheated palm oil methyl ester used in a diesel engine. *Biofuels*, pp. 1-17.
4. Menelik Walle Mekonen, N. Sahoo and S. K. Hotta (2017). Effect of Preheating and Fuel Injection Pressure on Performance Parameters of Diesel Engine Running with Biodiesel. *Book chapter, Springer Proceedings (Accepted)*.
5. Menelik Walle Mekonen and N. Sahoo, Effect of Compression Ratio and Injection Pressure on Diesel Engine Performance Parameters Running with Preheated Biodiesel-Diesel Blends of Fuel. *International Journal of Green Energy (Under Review)*.
6. Menelik Walle Mekonen and N. Sahoo, Thermodynamic and thermos-economic analysis of preheated and blended castor oil methyl ester in a compression ignition engine. *International Journal of Energy Research (Under Review)*.

Conferences

1. Menelik Walle Mekonen and Sahoo N., Santosh K. H., 2017. Effect of preheating and fuel injection pressure on performance parameters of diesel engine running with biodiesel. *6th International Conference on Advances in Energy Research 2017 December 12–14, 2017, IIT Bombay, India*.
2. Menelik Walle Mekonen, Niranjana Sahoo, Santosh Kumar Hotta (2017), “Effects of injector nozzle geometry on engine parameters for a diesel engine fueled with preheated pure biodiesel”. *Proceedings of ICTACEM 2017 International Conference on Theoretical, Applied, Computational and Experimental Mechanics December 28-30, 2017, IIT Kharagpur, India*
3. Menelik Walle Mekonen, Niranjana Sahoo, Santosh Kumar Hotta (2018), “Effects of intake air and fuel preheating for improving diesel engine operation parameters running with methyl ester biodiesel blends”. *Proceedings of the International Conference on Sustainable Energy and Environmental Challenges (SEEC-2018), IISc Bangalore, India, 31 December 2017 – 3rd January 2018*
4. Menelik Walle Mekonen, N. Sahoo, S. Hotta (2017), “Improving fuel economy and NOx emission of diesel engine using preheating and exhaust gas recirculation”. *Preceding of the National Conference on Sustainable Mechanical Engineering: today and beyond (SMETB 2017) March 25–26, 2017 Department of Mechanical Engineering, Tezpur University, India*.
5. Menelik Walle Mekonen, Niranjana Sahoo, Santosh Hotta (2017), “Evaluations of combustion characteristics of a compression ignition engine running with castor oil methyl ester - diesel blends”. *Proceedings of the 7th International and 45th National Conference on Fluid Mechanics and Fluid Power (FMFP). December 10-12, 2018, IIT Bombay, Mumbai, India*.