SOME INVESTIGATIONS ON USE OF RENEWABLE FUEL IN COMPRESSION IGNITION ENGINE

A thesis submitted to the Delhi Technological University, Delhi in the fulfillment of the requirements for the award of the degree of

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by

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DECLARATION

I hereby declare that the thesis titled **"Some Investigations on use of Renewable Fuel in Compression Ignition Engine"** is an original work carried out by me under the supervision of Dr. Naveen Kumar, Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi. This thesis has been prepared in conformity with the rules and regulations of Delhi Technological University, Delhi. The research work reported and results presented in the thesis have not been submitted either in part or full to any other University or institute for the award of any other diploma or degree.

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CERTIFICATE

This is to certify that the work embodied in the thesis titled **“Some Investigations on use of Renewable Fuel in Compression Ignition Engine”** by Kirat Singh, **(**Roll No.: **2K16/PHD/ME/02**) in partial fulfillment of requirements for the award of Degree of **Doctor of Philosophy in Mechanical Engineering**, is an authentic record of student’s own work carried by him under my supervision.

This is also certified that this work has not been submitted to any other University or institute for the award of any other diploma or degree.

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Dedicated to my Family & Friends whose constant support helped me in completing this Research

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ABSTRACT

Compression ignition engines are a primary source of power in the transportation sector, which, unfortunately, generate greenhouse gases, posing a significant threat to both the environment and human health. As a response to this concern, the exploration of alternatives to diesel has gained prominence due to their potential to mitigate greenhouse gas emissions and enhance energy security. Vegetable oils are abundantly available in India and has garnered the attention of the Indian government for conversion into biofuels. Biofuels are generally produced by the transesterification process. By using this process, the raw vegetable oil is converted to its methyl/ethyl ester and its viscosity is significantly reduced. However, the process posses some challenges like increasing the cost of biofuel production, and low utilization of the by-products. Low viscosity vegetable oils that do not need to be trans-esterified have been previously used for operating the diesel engine. Pine oil and eucalyptus oil are two such fuels that have the potential to replace diesel. This research evaluates the viability of using the two oils as alternatives for diesel engine by examining their impact on combustion, performance, and emission in comparison to traditional diesel. The aim of the study is to optimize the ratio of these oils so as to create an ideal blend with enhanced fuel qualities. Hence, P5E5D90 (5% pine oil, 5% eucalyptus oil and 90% diesel), P10E10D80 (10% pine oil, 10% eucalyptus oil and 80%diesel), P15E15D70 (15% pine oil, 15% eucalyptus oil and 70%diesel), P50E50 (50% pine oil, 50% eucalyptus oil), E20D80 (20% eucalyptus oil and 80% diesel), P20D80 (20% pine oil and 80% diesel), ED20D80 (20% distilled eucalyptus oil and 80% diesel), P20D80 (20% distilled pine oil and 80% diesel) and PD10ED10D80 (10% distilled pine oil, 10% distilled eucalyptus oil and 80%diesel) fuel samples were selected.

In the first phase, the research delves into the physicochemical characteristics of the fuel samples such as calorific value, density, flash point, surface tension and kinematic viscosity.

The properties of the fuel meet the relevant ASTM standards. To analyse the composition of the fuel samples and their relationship with engine performance, GC-MS analysis was carried out. The major constituents of pine oil were carene (46.53%), alpha-Terpineol (14.79%), and limonene (9.6%). While, eucalyptus oil consists of carene (15.69%), cyclopentene (25.28%), Limonene (6.68%), Bicyclo (3.68%), and Citronellene (2.24%). Then the shelf life of the fuel samples was tested. The fuel samples underwent a one-year stability assessment, during which the changes in calorific value, viscosity, and density were monitored. These parameters showed minimal alterations over time, and the rate of change was low.

The second phase involved efforts to fine-tune the combination ratios of plant-based oils, aiming for an optimal diesel blend with enhanced fuel properties. The brake thermal efficiency with blend of only pine oil was lowest. With equal blending of pine and eucalyptus oil, the thermal efficiency was lower than diesel fuel engine operation. The lowest efficiency was observed with P20D80. With equal blend of only pine and eucalyptus oil the efficiency was lower than diesel at all loads. When the equal blends of pine and eucalyptus oil were mixed with diesel, the thermal efficiency increases with an increasing proportion of the biofuels in the blend. The thermal efficiency was highest with P10E10D80 blend at all the load conditions. These fuel blends exhibited an earlier onset of heat release compared to baseline diesel oil due to their lower kinematic viscosity and flash point. Furthermore, the blends showed a higher peak heat release than diesel. Test blends demonstrated reduced HC, CO, and smoke emissions, ranging from 23% to 48% less than the baseline, with higher blend proportions resulting in lower emissions. In all test samples, NOx emissions surpassed diesel, with the P10E10D80 blend showing an 8.60% increase above the baseline.

With the blend of distilled biofuels and diesel, the thermal efficiency was found to increase with increase in load. Also, the efficiency was higher than the non-distilled blends. PD10ED10D80 demonstrated the highest engine efficiency due to the presence of all seven major constituents, with Carene being the most abundant at 30.52%. The inclusion of Citronellene and bicyclo [2.2.1] heptane in the PD10ED10D80 blend resulted in increased brake thermal efficiency. PD10E10D80 biofuel blend stands out as a superior fuel offering advantages in terms of combustion characteristics (max. HRR: 71.94 J/°CA, max. in-cylinder pressure: 79.63 bar), emissions reduction (HC emission decreased by 14.70%, CO emission decreased by 13%, smoke opacity reduced by 20.14%, and NOx emission increased by 8.79%). It is thus seen that the refined biofuels represent a promising solution for cleaner and more efficient energy in the transportation sector, surpassing the non-distilled alternative in multiple critical aspects.From the previous experiments, it was observed that the distilled biofuel blend PD10ED10D80 performed better than P10E10D80 blend in terms of combustion, performance and emission. However, the oxides of nitrogen emission were higher with the distilled biofuels blend. Also, the cost of distillation would further increase the cost of the blend. Hence, P10E10D80 blend was used for further exploration.

In the last phase of the experimentation, for further substitution of diesel with bio-oils, methanol was fumigated in the intake manifold of the engine. The flow rate of methanol was varied from 10% to a maximum of 56%. The results reveal that higher percentages of methanol substitution at full load resulted in improved thermal efficiency and combustion characteristics. The energy consumption of the experimental engine reduced by 17% when 39% methanol substitution was carried out at full load condition. Simultaneously a reduction in emissions except NOx emission was also observed. NOx emission was found to increase by approximately 16% at 40% substitution levels. The smoke opacity was also found to decrease by approximately 10% at maximum methanol substitution as compared to engine operation with the blend in single fuel mode. In conclusion, this study shows that up to 50% substitution of diesel at full load condition with pine oil, eucalyptus oil and methanol is possible without much affecting the engine performance.

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NOMENCLATURE

A/F	Air to Fuel Ratio
ASTM	American Society for Testing and Materials
ATDC	after Top Dead Center
BMEP	Brake mean effective pressure
BSEC	Brake specific energy consumption
BTE	Brake thermal efficiency
BTDC	before Top Dead Center
^{0}CA	°CA Degree Crank Angle
CA50	Crank Angle At Which 50% of Heat is Released
CO	Carbon monoxide
CI	Compression Ignition
CN	Cetane Number
C.V	Calorific Value (MJ/Kg)
CO	Carbon Monoxide Emission
CO_2	Carbon Dioxide
EGR	Exhaust Gas Recirculation
EGT	Exhaust Gas Temperature
FAME	Fatty Acid Methyl Ester
FFA	Free Fatty Acids
F.P	Flash Point
GC-MS	Gas Chromatography-Mass Spectrometry
HC	Unburned Hydrocarbon
IMEP	Indicated Mean Effective Pressure
K.V	Kinematic Viscosity
LHV	Lower heating value
MToe	Millions of Tonnes of Oil Equivalent
NOx	Oxides of Nitrogen Emission
NO	Nitric Oxide
PAH	Polynuclear Aromatic Hydrocarbons
PM	Particulate Matter

RPM	Revolutions per Minute
SMD	Sauter Mean Diameter
SOx	Oxides of Sulphur
TDC	Top Dead Center
D100	Diesel 100% (Neat oil)
D100	100% Diesel
P100	100% Pine oil
E100	100% Eucalyptus oil
P5+E5+D90	5% Pine oil 5% Eucalyptus oil and 90% Diesel
P10+E10+D80	10% Pine oil 10% Eucalyptus oil and 80% Diesel
P15+E15+D70	15% Pine oil 15% Eucalyptus oil and 70% Diesel
P50+E50	50% Pine oil and 50% Eucalyptus oil
E20+D80	20% Eucalyptus oil and 80% Diesel
P20+D80	20% Pine oil and 80% Diesel oil
PD10+ED10+D80	10% Distilled Pine oil, 10% Distilled Eucalyptus oil and 80% Diesel
ED20+D80	20% Distilled Eucalyptus oil and 80% Diesel
PD20+D80	20% Distilled Pine oil and 80% Diesel

CHAPTER 1
INTRODUCTION

1.1 Motivation

Current international energy issues and the need for sustainable solutions and environmental considerations provide the motivation behind this research. Specifically, this research aims to address the increasing dependence on finite fossil fuels and the risks associated with global energy security, by focusing on compression-ignition engines. The urgency to diversify the energy sources and reduce reliance on geopolitically sensitive resources is highlighted by the volatility of oil prices and the imminent depletion of fossil fuel reserves. Furthermore, the environmental impact of fossil fuel usage, resulting in climate change, air pollution, and health concerns, highlights the necessity for eco-friendly alternatives. It is thus imperative to explore renewables as a solution to the pertinent challenges and provide a cleaner and sustainable energy alternative. Adopting alternative fuels offers enhanced combustion efficiency and reduced emissions, contributing significantly to improved air quality and economic stability.

Within the scope of this alternative, the potential of plant-derived fuels can be explored for compression-ignition engines. Thus, a comprehensive investigation of a unique blend of pine oil, eucalyptus oil, and methanol aiming to development of cleaner and more sustainable fuel solutions was carried out in this study.

1.2 Global Energy scenario

The global energy landscape is confronted with a complex and interlinked set of challenges and opportunities. The increasing need for energy worldwide has resulted in a notable dependence on non-renewable fossil fuels, including coal, oil, and natural gas. These limited resources currently account for over 85% of the world's primary energy supply. However, this heightened dependence on fossil fuels raises concerns regarding energy security, particularly due to the volatility of oil prices and the finite nature of these resources. In 2023, global energy consumption experienced a slowdown in growth, with an increase of 2.4% compared to the previous year [1]. The current growth rate exceeds the average growth rate observed between 2010 and 2023, which stood at 2.2% per year. China, the largest energy consumer globally, saw its energy consumption growth rate slow down from 5.2% in 2021 to 3.4% in 2023. Similarly, the United States experienced a decrease in growth from 4.9% in 2021 to 2.1% in 2023. Some countries demonstrated significant growth in energy consumption in 2023. India saw a growth rate of 7.5%, Conversely, Saudi Arabia and Indonesia saw growth rates of 8.6% and 22%, respectively. More moderate growth rates of 4.2% and 3.1% are seen in Canada and Latin America, respectively; Brazil, Mexico, and Argentina are major contributors to these growth rates. In contrast, Europe saw a decline in primary energy consumption by 4.1%, with a 4.6% decrease in the European Union [2]. In the context of the global energy landscape, various regions have experienced fluctuations in energy consumption in recent years. The United Kingdom and Turkey, for instance, registered a decline of around 3.8%. This was caused by a number of things, such as increased energy prices, milder weather, and worries about an approaching recession in the wake of Russia's invasion of Ukraine. Consequently, there was a notable reduction in energy demand from both residential and industrial consumers.

Similarly, the Commonwealth of Independent States (CIS) recorded a 3.2% decrease in energy consumption, triggered largely by the war in Ukraine and sanctions directed at Russia. Meanwhile, In OECD-Asia, energy consumption stayed mostly consistent, with stable rates seen in nations like Australia and South Korea. Conversely, Japan saw a slight decline of roughly 1.3%. It is pertinent to note that these energy consumption patterns are invariably driven by various factors, including economic conditions, geopolitical events, and policy decisions.

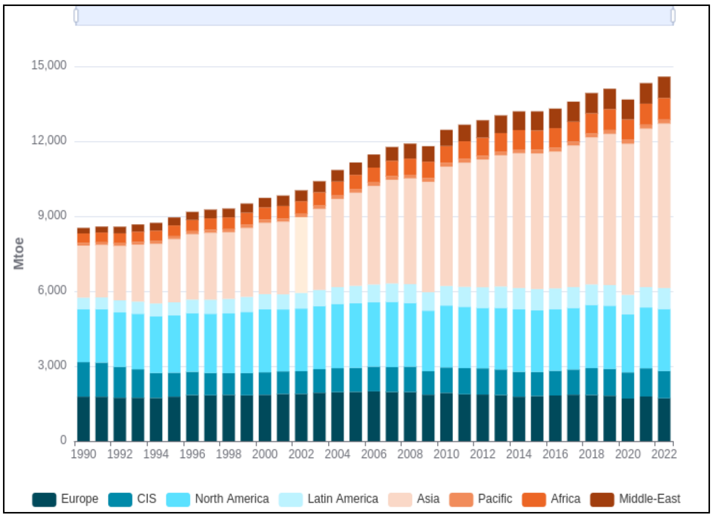


Fig. 1.1 Primary Energy Consumption of Top Global Energy Consumers (Millions of Tonnes of Oil Equivalent, 2023) [2]

1.3 India's Energy Outlook

India's rapidly expanding economy heavily relies on its energy resources. Although India does have some domestic oil reserves, they are relatively small in comparison to the country's substantial energy demands. Since gaining independence in 1947, India's crude oil production has surged from a meagre 0.28 million tons to approximately 29.2 million metric tons (MMT) by 2022-23. However, to meet its energy requirements for industrial, transportation, and agricultural sectors, India heavily relies on imported crude oil. In the fiscal year 2022-23, India's imports surged to around 232.7 MMT. This substantial surge in imports imposed a massive financial burden on the national treasury, amounting to a staggering USD157.2 billion. It is noteworthy that these imported oils constitute a significant (90%) of the energy consumed within India's transportation and agricultural sectors. The demand for diesel in India has witnessed a significant increase over the years, rising from approximately 53 MMT in the 2007-2008 periods to a substantial 85.9 MMT by 2022-2023 [3]. Recent statistics reveal that there was a 6.2% increase in crude oil imports during 2023 compared to the previous year [4]. Given this surge in energy requirements, it is imperative for India to strategize ways to reduce its dependence on oil and develop sustainable and renewable energy alternatives. The proposed research aligns with this objective and aims to explore ways of reducing India's reliance on non-renewable energy sources and promoting sustainable energy practices.

1.4. Challenges in the Crude Oil Market and Escalation of Petroleum Fuel Costs

Regardless of a country's financial situation, energy is essential to its development both economically and socially. However, both developed and developing countries have significantly increased their reliance on fossil fuels over the years. the Fig 1.2, it revealed evident that global energy demand has surged by 127%, rising from 6101 million tonnes of oil equivalent in 1973 to 13864 by 2022-23 [5, 6]. Together, the three main energy sources-oil, natural gas, and coal-account for 85.18% of the world's total energy consumption. The exponential increase in energy demand, coupled with the depletion of non-renewable fossil fuel reserves, has put immense pressure on the environment, economies, and geopolitics of nations worldwide. This underscores the urgent need to transition towards renewable energy sources that offer a cleaner, more sustainable, and eco-friendly alternative. The proposed research aligns with this global objective and aims to explore innovative solutions that promote sustainable energy practices and address the pressing challenges facing the energy and environmental sectors.

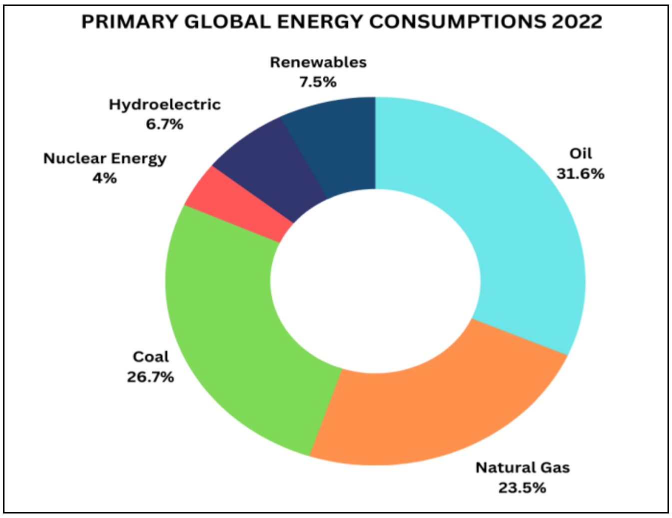


Fig. 1.2 comparative analyses for energy supply 2023 [5, 6]

1.5 Need of alternatives

Fossil fuels continues to fulfill more than half of global energy requirements, with escalating growth, the gap between energy supply and demand is growing. According to projections, the remaining life of natural gas is 36 years, coal is 106 years, and oil is expected to last 34 years. This suggests that coal might be the sole surviving in fossil fuel by 2042 and could potentially be available up to 2112 [7].

1.6 Environmental Issue

The methodologies involved in the extraction, transit, and refinement of fossil fuels have significant adverse effects on the environment. At every stage, including extraction, transportation, and storage, there is a risk of leaks or spillages that can result in the contamination of air and water resources. Furthermore, the refining process itself contributes to environmental damage.

Fossil fuels, are mostly composed of carbon, hydrogen, and trace amounts of sulfur. When they burn, a variety of gases, ash, soot, and other organic compounds are generated that have a big impact on air quality. Particulates, carbon monoxide (CO), hydrocarbons (HC), sulfur oxides (SOx), nitrogen oxides (NOx), benzene, polyaromatic hydrocarbons (PAH), and aldehydes are some of the most hazardous emissions. Methane (CH_4) and carbon dioxide (CO_2) are two of the greenhouse gases released during the burning of fossil fuels. These emissions contribute to global warming, and climate anomalies and increased occurrences of wildfires. Research findings indicate that under specific conditions, these pollutants can transform into harmful aerosols or acids, posing dangers to both aquatic and terrestrial ecosystems [8-10]. Given these challenges, there's a global inclination towards alternative energy sources.

1.7 Alternative Fuels and Compression Ignition (CI) Engine

Compression Ignition (CI) engines play a crucial role in both transportation and agriculture. Moreover, they are utilized in off-road vehicles and power generation applications. Their durability, user-friendliness, and low maintenance requirements make them popular choices. These engines are also recognized for their high thermal efficiency, substantial torque, and efficient fuel consumption. The CI engine's characteristics like leaner air-fuel mixtures and minimized pumping losses and elevated compression ratio contribute to its notable thermal efficiency. Nevertheless, these engines release emissions that negatively impact the environment and pose health risks to humans.

Diesel engines in particular emit particulates in minute form, exceeding concentrations of 10 million particulates per cm^3. Primary carbon particles have diameters ranging between 0.01 to 0.08 microns, while agglomerated particles measure between 0.08 to 0.25 microns, with a significant 90% of these particles measuring less than 1 micron. Numerous authoritative bodies have labeled these particulates as either "known carcinogens" or "potential carcinogens" for humans. Such emissions amplify risks associated with cardiovascular and respiratory conditions, including asthma and bronchitis. Furthermore, these particles contain hydrocarbons known as polynuclear aromatic hydrocarbons (PAHs), which have been linked to cancer. PAHs are hydrocarbons that have numerous benzene rings. Especially, the soluble organic components of these particulates, possessing four to five rings, are among the most noxious compounds [11]. A sustainable remedy to these challenges lies is substituting the fossil fuel by renewable fuels, either partially or wholly. For these alternative fuels to be a viable long-term solution, a substantial resource base is crucial to ensure year-round availability and validate the investment. It's essential that such fuels are congruent with the engine's associated components like hoses, tanks, and pumps, as well as align with the storage, current transport, and commercial infrastructure.

1.8. Biofuels: A Viable Alternative for Diesel Engines

In response to the growing concerns of energy stability, ecological conservation, and sustainability, focus has been shifted on to renewable fuels [12-14]. Recent years have witnessed the emergence of biofuels as a highly compelling substitute for conventional diesel fuels [15, 16]. Biofuels typically refer to fuels derived from plant-based materials and other organic matter, with the potential to fill the gap left by petroleum-based fuels in the near future [17]. A projection by the USD [18, 19] aims at enabling the US to meet 30% of its current gasoline demands with biofuels by 2030. the need to reallocate existing farmlands. The forecast also underscores the potential to utilize 2.6 billion tons of biomass for generating 3.8 billion gallons of ethanol by 2025. Different countries have distinct strategies and frameworks for the advancement and utilization of biofuels.

Over the last decade, substantial progress has been achieved in uncovering innovative biofuels and their compatibility with diesel engines, spurred by the growing demand for conventional petroleum fuels. These bio-based fuels span diverse categories, including vegetable oils, alcohols, esters (biodiesel), carbonates, ethers, and acetate derivatives [20-22]. These advancements aim to address the challenge of sustainable fuel supply. By delving deeper one can find viable solutions to sustainably meet the energy needs and foster a cleaner and greener future.

Notably, the intrinsic oxygen content in these biofuels, which is absent in traditional fuels, presents significant advantages in engine combustion, efficiency, and emissions. Being renewable and decomposable, biofuels have significantly lower emissions like CO, HC and smoke [23-25] when deployed in diesel engines. Crucially, the emissions of CO_2, have also been reported to decrease with the use of biofuels [26].

Stationary diesel engines, can benefit tremendously from the exploration of alternative fuel options. Consequently, conducting research into these alternative fuels holds significant potential in addressing these diverse challenges [27-29].

1.9 High viscous biofuels

Over an extended period, researchers worldwide have championed the viability of vegetable oil as a feasible alternative to diesel fuel [30, 31]. Vegetable oil has some qualities that make it a desirable substitute fuel for diesel engines [32]. Researchers are continuing to address challenges associated with vegetable oil, including blending with diesel, pre-warming, and taking engine components or operational parameters like pistons or injectors [33]. Yet, despite these modifications, issues like heightened carbon buildup and lubrication oil contamination remained [34].

Addressing these concerns, the scientific community introduced biodiesel [35-37], an ester derived from vegetable oils employing alkaline KOH or acid catalysts (H_2SO_4). [38-40]. By transforming triglycerides into methyl esters, the molecular weight and viscosity are notably decreased to about a third and an eighth, respectively, of their original values [41]. Post this trans-esterification process, biodiesel properties become more suited for diesel engine usage. Contemporary research has seen an influx of studies on biodiesel-diesel blends, and with its enhanced attributes, a consensus emerged to mix diesel with 20% biodiesel [35]. This consensus subsequently ignited a quest for suitable vegetable oil sources for biodiesel production. Recent engine performance evaluations have highlighted noticeable improvements with biodiesel usage compared to regular diesel [42, 43]. Additionally, emissions like smoke, CO, HC and CO_2 have been shown to decrease, albeit at the cost of a minor uptick in NO_X emissions [44-46]. however, biodiesel also poses few challenges viz., higher raw material costs, suboptimal storage and oxidation resistance, low calorific value, performance issues at colder temperatures, and NO_X emissions [47]. From an economic standpoint, even though biodiesel offers environmental advantages, it hasn't been as cost-efficient as other petroleum-based fuels [48], which contributed to its limited early adoption. Yet, current challenges in the petroleum sector, combined with the ever-growing energy demand, have rejuvenated interest in vegetable oil-derived biodiesel development. Contemporary guidelines support biodiesel synthesis from non-edible oils, as using edible variants might strain food resources, and are generally costlier [35, 49, 50].

1.10 Less viscous biofuels

Though the esters derived from vegetable oils as alternative renewable diesel fuel gain momentum, however, researchers are increasingly exploring less viscous fuels, such as camphor oil, orange oil, eucalyptus oil and, pine oil etc [51-54]. These biofuels primarily derived from plant components, distinct from triglycerides like vegetable oils, commonly exhibit a lower cetane number and viscosity. Although their ignition quality is somewhat lower, blending them with diesel serves as a straightforward approach for incorporating such fuels in diesel engines [55]. Yet, numerous challenges emerge when blending these fuels with diesel, prompting recommendations to limit the blend ratio [56, 57]. Furthermore, the miscibility of these low-viscosity fuels with diesel can be problematic, risking separation at colder temperatures. This separation issue can be addressed using emulsifiers or specific additives that promote effective blending of fuels like ethanol and diesel [58, 59]. Nevertheless, these less viscous fuels are not without drawbacks, such as reduced lubrication capabilities and an elevated ignition temperature [60, 61]. The advantages highlighted by researchers encompass superior atomization, improved fuel-air mixture, and enhanced combustion, emissions, and performance in diesel engines without the need for modifications [62, 63]. In a vein similar to alcohols, there's been interest in substances like diethyl ether or dimethyl ether as potential diesel fuels [64-71]. A recent study has mentioned DME as a cleaner and more efficient alternative for compression ignition due to its emission-reducing properties [65]. While DME helps address the NOx-soot dilemma, engines require a revamped fuel storage and delivery infrastructure to accommodate it. A recent study reported DEE derived from ethanol dehydration as a valuable component for compression ignition engines [72].

1.11 Utilizing Compression Ignition Engines with Alternative Fuels

To facilitate effective utilization of alternative fuels in diesel engines, it is imperative that these substitutes exhibit characteristics closely resembling those of conventional diesel fuel. especially vegetable oils, have emerged as promising alternatives given their alignment with established diesel properties, including cetane number, heat of vaporization, and energy density. However, a noteworthy disparity exists in the form of their considerably higher viscosity when compared to diesel. This heightened viscosity can influence crucial factors like spray atomization, vaporization, and the thorough blending of fuel and air.

Utilization of vegetable oils in diesel engines not only offers significant and sustainable prospects for harvesting, and processing of oil-yielding plants but also contributes energy independence. This pursuit encompasses a multifaceted strategy, encompassing policies related to both oil and natural gas production. It also promotes hybrid and electric vehicles, and the exploration of economical alternative fuels to reduce dependence on imported oil. In the case of a country like India, which accommodates 17% of the global population, a balanced approach is essential, drawing from both renewable and non-renewable energy sources. Grassroots, agriculture-centric energy production has the potential to not only fortify community-driven economies but also bolster overall energy security [73, 74].

1.12 Problem and Research Goals

From the previous discussions, it is evident that there has been significant past research focusing on the potential of various biofuels for diesel engines. While there has been a significant transition towards biomass-based fuels, the supply may not be adequate to meet the petroleum fuel demands of a country. For example, in a big country like India, where 200–230 billion gallons of petroleum are needed annually [75], Jatropha also as source isn't practical. As such, diversifying into multiple novel biofuels can address these supply concerns, promoting national self-reliance and also curbing detrimental emissions. Depending on geographical and climatic nuances, different regions within a country could harness and develop their unique biofuels. This decentralized approach would not only reduce dependence on a single biofuel type but also mitigate transportation challenges across vast distances.

However, the challenge doesn't end with simply discovering new biofuels. Making sure these fuels can be used effectively in diesel engines is necessary. Given that diesel engines have traditionally been designed for diesel fuel, introducing biofuels can raise durability concerns due to their unique properties. This means that adaptations, either in the engine's design or its operational parameters, may be essential. Looking ahead, more research is pivotal to adapt diesel engines to these emerging biofuels. Though

20% biofuels blend with diesel is currently deemed acceptable for standard diesel engines, further investigations are crucial to enhance this blend ratio. This research's primary goal is to delve into innovative biofuels suitable for diesel engines. Additionally, to preempt potential longevity issues associated with new plant oil.

To date, a wide array of vegetable oil-derived fuels, notably biodiesel, has been recognized as sustainable fuel alternatives, and their potential in diesel engines has been established. It's evident that these vegetable oil fuels possess a higher viscosity, leading to challenges in atomization and combustion when utilized in diesel engines. Furthermore, scientists have pinpointed other fuels with reduced viscosity, which unlike vegetable oil-based fuels; don't necessarily have high FFA content. These alternatives have superior atomization capabilities, to boost engine combustion and overall performance. Alcohols and ethers are some of the biofuels derived from plants.

Despite the known advantages of these less viscous plant-derived fuels, such as enhanced fuel atomization and improved air/fuel mixing, there seems to be limited exploration in this category. Consequently, there's a pronounced need to further research in the domain of less viscous plant-based fuels. This will not only address the previously mentioned challenges but also has the potential to amplify engine efficiency and emissions with these newly conceptualized biofuels.

CHAPTER 2

LITERATURE REVIEW

2.1. Introduction

The motivation to assess the practicality of employing lower-viscosity fuels in diesel engines emerged from the recent advancements in their production and utilization within this engine type. Despite possessing unique fuel properties, these less viscous fuels have increasingly penetrated the alternative fuel sector as a viable substitute for conventional petroleum diesel. Their popularity stems from attributes such as renewability, wide availability and biodegradability. Generally, fuels with viscosities lower than that of standard diesel are categorized as less viscous fuels. They can also be classified as light biofuels, distinguishing them from biodiesel, which is derived from vegetable oils. Significantly, these low-viscosity fuels are made from plants or biomass other than seeds, setting them apart from conventional biodiesel. Moreover, fuels with lower viscosity are anticipated to exhibit improved properties in terms of evaporation, atomization, and spray behaviour when compared to standard diesel. Typically, these low-viscosity fuels possess lower cetane index due to the presence of relatively shorter carbon chain lengths. Methanol, ethanol, butanol, eucalyptus oil, pine oil, diethyl ether, diethyl ether, and carbonate-based fuels serve as examples of widely recognized low-viscosity fuels. Interestingly, fuel additives such as carbonates and ether-based fuels are used with biodiesel or diesel to improve the fuel properties. Among various alternatives, alcoholic fuels have garnered significant attention due to their perception as cost-effective and produced from waste renewable raw materials. In addition to alcohols, pine oil and eucalyptus oil are becoming more and more viable alternatives to diesel engine fuel. While alcohols are produced through the fermentation of starch or sugar, pine oil and eucalyptus oil are obtained through the distillation of leaves from pine and eucalyptus trees, respectively. It is essential to emphasize that although these fuels share the common attribute of lower viscosity, their origins and compositions differ substantially. Researchers have thoroughly documented the properties of alcohol-diesel blends for diesel engine applications in their comprehensive review studies. Moreover, numerous scientists have conducted research on subjects concerning energy security, the swift rise in petroleum fuel costs, and environmental impacts [76-78]. Hansen et al. [79] reviewed diesel-ethanol mixes, providing information on engine performance, durability, emissions, and blend characteristics. Agarwal et al. conducted a study that examined the usage of alcohols as fuel for internal combustion engines [27]. Providing details on ethanol production, properties, and the performance and emission characteristics of diesel-ethanol blends, Agarwal presented noteworthy insights. In particular, the author's evaluation stressed how important it is to conduct research on material compatibility with mixtures of diesel and alcohol. Giakoumis et al. [80] reviewed the literature to find out how mixtures of ethanol and n-butanol affected the emissions from a diesel engine, both regulated and unregulated, particularly when operating temporarily. Three distinct alcohol-diesel blends, diesel-butanol, diesel-ethanol, and diesel-methanol-are investigated by Kumar et al. [81]. Their analysis provided crucial details on the characteristics of fuel, the economics of manufacturing, and how these factors affect emissions and engine performance. Based on a compilation of previous review works, it is clear that the main emphasis has been on alcohol-diesel blends; no thorough reviews have addressed the other ways in which alcohol fuels can be used, including fumigation, dual injection, and, to a lesser degree, using them as the only energy source. In a comparable context, none of these operational modes have undergone assessment regarding the operational viability of another category of low-viscosity fuel, specifically pine oil and eucalyptus oil. In light of this context, an extensive review was undertaken with the aim of offering valuable insights to emerging researchers regarding the utilization of two primary low-viscosity fuels, namely eucalyptus oil and pine oil, in diesel engines. It is worth noting that although both of these fuels exhibit lower viscosity compared to traditional counterparts, they possess distinct compositions and unique properties. Thus, thorough examination of the thermal and physical properties of different fuels was carried out. Aligned with the extensive body of research on the engine characteristics of alcohol-diesel blends, which has been thoroughly documented. Hence, this review does not reiterate this aspect. Instead, the focus shifted towards exploring the feasibility of employing alcohol-diesel blends in alternative operational modes, such as ethanol fumigation by carburetor and as the sole source of fuel. Conversely, when it comes to the lesser-explored less viscous fuel, pine oil, and eucalyptus oil, an in-depth analysis of its operational viability in diesel engines across all conceivable modes of operation was conducted. Thus, providing a comprehensive coverage of this relatively uncharted territory.

2.2 Less viscous biofuels composition and property analysis

The physical and thermal characteristics of fuels are of utmost importance when evaluating its suitability for application in diesel engine, alongside the consideration of different engine operational and design variables. When engaging in research concerning the creation, examination, and application of alternative fuels in a diesel engine, it becomes essential to perform a thorough assessment of the properties inherent to these fuels.

In this context, when conducting experimental investigations involving biofuels in a diesel engine, the initial step involves determining their physical and thermal properties. Furthermore, it is essential to ensure that these biofuels conform to international standards before introducing them into diesel engine applications.

Table 2.1: Physicochemical properties of less viscous fuels [14, 79, 81, 82, 83]

Property	Diesel	Eucalyptus oil	Pine oil	Methanol	Ethanol	Butanol
Kinematic viscosity (10^{-6} m^2/s)	3.6	2	1.3	0.59	1.2	3.7
Gross calorific value (kJ/kg)	42700	43270	42800	19700	26800	33070
Density (kg/m^3)	822	890	875	790	789	809.8
Boiling point (°C)	180-340	175	175	64	78	111.7
Flash point (°C)	74	54	52	12	13	35
Cetane index	54	54	11	3.8	7	25
Self-ignition temperature (°C)	250	325	77.78	463	423	397
Latent heat of vaporization (kJ/kg)	265	305	300	1170	900	578.4

As alcohol-based fuels gain prominence for internal combustion engines, several researchers have committed their efforts to investigate the physical and thermal properties of these fuels [27, 84-87]. In a parallel manner, subsequent to the introduction of pine oil and eucalyptus oil as alternative fuels for diesel engines, while performing trials in a diesel engine, a certain group of researchers has started to evaluate their properties [54, 88]. A thorough and detailed comparison of the properties of the fuels and their effects on engine characteristics is undertaken in this study. To facilitate this comprehensive analysis, an extensive summary of the thermal and physical properties of all the low viscosity fuels under consideration is compiled and is presented in Table 2.1.

It is commonly known that a fuel's physical and thermal properties are greatly influenced by its chemical makeup as well as the effects of functional group characteristics. In alignment with this recognition, it is pertinent to draw attention to the research conducted by Anand et al. [89] and An et al. [90]. To offer a comprehensive understanding of this subject, the crucial information, including molecular lights, group contribution parameters, molecular formulas, functional groups, critical properties, and chemical structures of the less viscous fuels under examination have been assembled and presented in Table 2.2 for reference.

2.3 Critical properties

Table 2.2 provides valuable insights into the chemical compositions of the considered fuels. Pine oil structure elements –CH_2– >CH–, =CH–>, C<, =C< and Eucalyptus oil, for instance, incorporates structural elements like -CH_2-, >CH–, and >C< within its ring groups, along with an additional oxygen atom. In contrast, the shorter carbon chains of methanol, ethanol, and butanol classify them as lower alcohols. In these alcohols, the functional group -OH is attached to a main or secondary carbon atom. It is noteworthy that these group contribution parameters exert distinct influences on the critical properties of these fuels. Important characteristics for each of the fuels under examination, as established by Jobac's technique, including temperature, pressure, and volume, are included in Table 2.2. While methanol, butanol, and ethanol as a group have lower critical temperatures, eucalyptus oil exhibits a significantly higher critical temperature. Upon careful inspection of these fuels' boiling ranges, it can be seen that methanol has the lowest value of all, with ethanol, butanol, and eucalyptus oil following in progressive order. Curiously, the boiling points and critical temperatures of these fuels coincide, suggesting a tight connection between the critical characteristics and the typical boiling point. These crucial characteristics are known to affect the fuels' thermal and physical characteristics, a topic shall cover in more detail in the section that follows, along with the fuels' molecular weight.

Table 2.2 Characteristics of less viscous fuel composition

Qualitative characteristics	Pine oil	Eucalyptus oil	Methanol	Butanol	Ethanol
Another name	Turpentine oil	Cineole	Methyl alcohol	Butyl alcohol	Ethyl alcohol
Critical volume (cm^3 /mol) -	504	509.5	-	278.05	166.5
Critical pressure (bar)	9.6	-	63	80.84	45
Oxygen group	-O	-O	-OH	-OH	-OH
Non-ring elements	$-CH_3$	$-CH_3$,	$-CH_3$,	$-CH_3$, $-CH_2-$	$-CH_3$, $-CH_2-$
Ring elements	$–CH_2–$, >CH–, =CH–>, C<, =C<	$-CH_2-$ >CH-, >C<	-	-	-
C/H ratio	0.33	0.55	0.25	0.4	0.33
Critical temperature (°C)	210	422	237	272	226
Stoichiometric air to fuel ratio	14	14	6.5	11.1	9
Molecular formula	$C_{10}H_{18}O$	$C_{10}H_{18}O$	CH_3OH	C_4H_9OH	C_2H_5OH
Molecular weight (g/mol)	154.28	154.24	32.04	74	46.06

2.4 Ignition properties

The cetane number is a crucial parameter that gauges the ignition quality of a fuel, and it can be influenced by various factors. Generally, as the carbon chain length of a fuel increases, its cetane number is expected to rise, while it exhibits an inverse correlation with the presence of double bonds (degree of un-saturation) [91]. In the case of methanol, ethanol, and butanol, there are no double bonds within their molecular structures. However, their carbon chain lengths are relatively short, resulting in lower cetane values for these fuels. Similarly, pine oil has a lower cetane index. The molecular makeup of eucalyptus oil is remarkably different from that of methanol and ethanol. Despite containing ten carbon atoms, eucalyptus oil can be classified as an alicyclic hydrocarbon with double bonds in its structure. Table 2.2 indicates that the greater cetane value of the substance can be attributed to its unique structure and the inherent presence of double bonds. Due to the lower cetane numbers of these less viscous fuels, they have a longer ignition delay and accumulate air-fuel mixture while they burn in a diesel engine. Consequently, during the premixed combustion phase, this causes their sudden burning [92,93]. Furthermore, all of these fuels have self-ignition temperatures higher than that of diesel, this influences their behaviour upon auto-ignition and necessitates the supply of ignition assistance when utilizing them in a traditional diesel engine.

2.5 Evaporation properties

Although, methanol, ethanol, pine oil, butanol and all share the characteristic of having a lower cetane number and eucalyptus oil has higher cetane index, they differ in other attributes that impact engine performance and combustion. Ethanol and Methanol, for instance, have higher latent heats of vaporization compared to other low-viscosity fuels. This quality makes them harder to evaporate and leads to a cooling effect within the engine [65]. Studies have also shown that as the concentration of these fuels in a blend increase, difficulties with cold engine starts become more apparent [94, 95]. When used in ethanol, diesel engines, and methanol tend to decrease the internal cylinder temperature, particularly at lower loads, affecting fuel evaporation. However, this issue is mitigated at full engine load due to elevated cylinder temperatures [96, 97]. Nevertheless, in contrast to methanol and ethanol, butanol possesses a lower latent heat of vaporization, reducing the impact of cold start issues. Due to its larger latent heat of vaporization than diesel, its evaporation qualities are affected when coupled with diesel. The significantly lower latent heat of vaporization of pine and eucalyptus oil, compared to ethanol and methanol, mitigates the cooling effect and presents fewer challenges in this regard. In conclusion, the lower flash temperatures and lower boiling points of these low-viscosity fuels encourage superior evaporation when compared to diesel.

2.6 Atomization and spray properties

Fuel surface tension and viscosity are well-recognized factors that influence how the fuel atomizes and mixes with air. Data indicates that the less viscous fuels under consideration all exhibit lower viscosity compared to diesel. Lower-viscosity fuels are more likely to disperse into finer droplets within the combustion chamber, as demonstrated by Hazar et al. [98,99]. This improves both the evaporation and combustion processes, resulting in enhanced engine performance compared to traditional diesel and high-viscosity biodiesel options. Beyond viscosity, surface tension is another crucial parameter influencing fuel dispersion. For the less viscous fuels considered, surface tension is generally lower. This has an impact on spray properties, which have been shown to be lower for certain fuels, such as spray tip penetration and droplet size [100,101]. In a related study, Wang et al. [102] discovered that owing to their lower viscosity and surface tension, the spray characteristics of ethanol and methanol closely resemble those of gasoline.

2.7 Fuel consumption properties

When evaluating the use of low viscosity fuels in diesel engines, energy density is a key factor, as it informs the amount of fuel needed to achieve a specified power output. Within the spectrum of less viscous fuels examined, ethanol and methanol demonstrate markedly lower calorific values compared to diesel, thereby constraining their potential for substantial diesel replacement. [57]. Butanol, however, has a higher calorific value than ethanol, making it less likely to result in increased fuel consumption in engines. Conversely, pine oil and eucalyptus oil's calorific value are comparable to that of diesel [103,104], meaning there's little to no loss in engine power when it's used. This allows for greater flexibility in blending it with diesel fuel without affecting performance. Another critical factor to take into account is the stoichiometric fuel-to-air ratio, especially under lower engine loads. Based on available data, methanol exhibits the lowest stoichiometric fuel-to-air ratio, attributed to a partially oxidized condition, followed by ethanol and butanol. This may affect engine performance at lower loads by making methanol more prone to lean combustion [105]. Due to presence of their natural oxygen content, pine and eucalyptus oils have a stoichiometric fuel-to-air ratio that is somewhat lower than diesel but higher than alcohol-based fuels.

2.8 Characteristics of stability and storage

The lowest temperature at which a fuel begins to evaporate and combine with air, allowing for ignition, is known as its flash point. This is an important fuel handling and storage element that affects insurance and fire safety laws. Since, these low-viscosity fuels typically have lower flash points than diesel, extra care must be taken when storing them. Notably, pine oil and eucalyptus oil are safer for storage as they have a higher flash point than ethanol and methanol. Conversely, alcohols have a higher self-ignition temperature than eucalyptus and pine oil, mitigating their lower flash point and minimizing storage hazards. To prevent combustion dangers, it is imperative to exercise caution when storing any less viscous fuels that have higher self-ignition temperatures and lower flash points than diesel.

When it comes to the stability of low-viscosity fuels, particularly in blends with biodiesel or diesel, some issues emerge over extended storage periods. For instance, alcohol-diesel blends may suffer from instability over time, leading to phase separation. Methanol, with its polar properties, is particularly prone to this issue [106]. In contrast, butanol mixes well with diesel and can be blended in almost any ratio. Ethanol also faces challenges related to phase separation due to its hydrophilic qualities [86,107], and extensive research recommends the use of surfactants and emulsifiers to combat this problem [108,109]. While the stability of blends and the possibility of phase separation have not been thoroughly investigated, engine tests have shown that pine and eucalyptus oils mix well with diesel.

2.9 Compositional attributes

Research has demonstrated that the inclusion of oxygen atoms in fuel can improve the quality of combustion. For example, biodiesel, despite being more viscous, has been found to improve combustion due to its 10-11% oxygen content by weight [22,44]. Higher levels of oxygen in fuel are associated with reduced smoke emissions, as they help oxidize soot at elevated combustion temperatures [110]. In the domain of low-viscosity fuels, methanol takes the lead with an impressive 50% oxygen content by weight. Following closely, ethanol and butanol exhibit 34.7% and 21.6% oxygen content, respectively. While, pine oil and eucalyptus oil contain 10.3% oxygen by weight. With its elevated oxygen levels, methanol is expected to yield lower smoke emissions in comparison to other low-viscosity fuels discussed in this review. Delving deeper into the carbon to hydrogen (C/H) ratios of these fuels reveals additional insights. Methanol boasts a C/H ratio of 0.25, while pine oil and eucalyptus oil exhibit a ratio of 0.55. Generally, fuels with higher C/H ratios are more prone to soot formation, thereby leading to increased smoke emissions. In this context, methanol, with its lower carbon to hydrogen (C/H) ratio in this context, methanol, with its lower carbon to hydrogen (C/H) ratio, is anticipated to generate fewer smoke emissions compared to the other fuels considered in this study.

2.10 Viability evaluations of low-viscosity fuels in blended fuel mode

2.10.1 Alcohol-based blended fuel operation

As the global community has witnessed the successful integration of alcohols in gasoline engines, there has been a growing exploration and advancement of their utilization in diesel engines. Alcohols, distinguished by their oxygen-rich composition and reduced viscosity, have demonstrated the ability to decrease hydrocarbon (HC) and carbon monoxide (CO) emissions. This reduction is attributed to their superior combustion efficiency and engine performance compared to traditional diesel fuel. Typically, a fuel containing approximately 30% oxygen by weight can achieve nearly smoke-free combustion [112], making alcohols particularly effective in this regard. Nevertheless, employing alcohols as the exclusive fuel in diesel engines poses practical challenges due to their higher self-ignition temperature and lower calorific value. However, these ignition challenges can be mitigated by preheating the engine's intake air, thereby enhancing combustion [113]. Additionally, incorporating alcohols into diesel engines may necessitate modifications to engine configurations, especially in the context of fuel injection mechanisms [114]. Adapting diesel engines to accommodate alcohols is resource-intensive, given that these engines are primarily designed for diesel fuel, and introducing a new fuel type requires comprehensive engine adjustments. To address the aforementioned challenges, recent research has indicated the potential of blending ethanol with diesel, often referred to as "diesohol." This approach not only refines the air-fuel mixture but also enhances combustion efficiency. The decreased viscosity and lower boiling point of ethanol contribute to enhanced atomization, promoting improved air-fuel mixing, optimal fuel evaporation, and complete combustion [115, 116]. Several investigations focusing on ethanol-diesel blends have demonstrated reductions in hydrocarbon (HC), carbon monoxide (CO), and smoke emissions attributed to improved combustion processes. [108, 117]. Nonetheless, a drawback is the prolonged ignition delay, leading to a noticeable accumulation of the air/fuel mixture during this delay. The in-cylinder temperature and peak heat release rate rise as a result, increasing NO_X emissions [118, 119]. A detailed analysis of the performance and emission characteristics of engines using ethanol-diesel and methanol-diesel mixes was carried out by Kumar et al. [83]. It is crucial to highlight that the researchers have not limited their experiments solely to blending alcohols with diesel[120-125]. There have been efforts to combine alcohols with biodiesel, with the aim of potentially phasing out diesel entirely[126-129]. The blending of ethanol with biodiesel enhances the blend's lubricating properties while optimizing other physical attributes[130-134]. Furthermore, tri-blends of alcohol, biodiesel, and diesel have been explored for engine applications. Comprehensive information on alcohol-biodiesel [66, 91, 111,135-145] and alcohol-biodiesel-diesel mixtures [64,101,146-154]. While numerous studies have delved into the use of alcohol-diesel and alcohol-biodiesel mixtures, challenges arise due to the less-than-ideal compatibility of alcoholic fuels with either biodiesel or diesel. Prolonged use of low-viscosity alcohol can lead to issues like injector leakage. To address these complications, certain experts have suggested incorporating specific lubricants, such as including 2% castor oil in the blend. [155]. Additionally, the blending of ethanol with diesel presents challenges due to potential phase separation in colder environments. To address this separation issue, researchers have proposed the use of emulsifiers or specific additives like do-decanol and surfactant 80, which facilitate improved blending of ethanol and diesel [59, 156]. Furthermore, ethanol and methanol's auto-ignition characteristics are influenced by their lower cetane numbers. Researchers have explored various approaches to modify the blend and enhance the ignition properties of alcohol-diesel blends. Some of the investigated methods involve the addition of cetane boosters like ethyl hexyl nitrate, peroxides, and diethyl ether, followed by an analysis of combustion properties such as ignition delay and peak heat release [157-162]. These studies widely concur that the inclusion of a cetane enhancer significantly improves the ignition properties of the blend. Notably, additives have been effective in reducing the NO_X emissions typically associated with fuels having lower cetane values. The utilization of alcohol-diesel mixtures with these additives has been comprehensively reviewed in works such as that by Ribeiro et al. [163]. Nonetheless, despite the solutions available for addressing separation and low cetane number issues, both methanol and ethanol possess relatively lower calorific values, which can impact overall engine efficiency. Due to their reduced calorific value, larger fuel volumes are required to achieve the desired power output, which is why ethanol's inclusion in diesel typically does not exceed 25% [164]. Therefore, while the advantages of alcohols as potential diesel substitutes are evident, constraints exist regarding the extent to which they can be blended with diesel.

2.10.2 Pine oil-based blended fuel operation

Previous research [165] has provided evidence that the combination of pine oil with diesel produces a blend with properties that position it as an intermediate between pure diesel and other alternative fuels. The viability of pine oil as a sustainable fuel for diesel engines gained recognition through the efforts of Valliagam et al. [166], who explored its viability when blended with kapok oil methyl ester for use in diesel engines. This study diverged from the traditional use of diesel and instead focused on two alternative fuels. Their findings revealed a notable reduction of 12.5% in smoke emissions, 8.1% in hydrocarbon (HC) emissions, and 18.9% in carbon monoxide (CO) emissions, while nitrogen oxide (NO_X) emissions remained comparable to those of diesel at maximum load. In a similar pursuit of exclusive renewable fuels, Vallianayagam et al. [167] compared pine oil, characterized by its lower viscosity and resembling alcohols, with high-viscosity biodiesel derived from kapok. Notably, pine oil demonstrated a calorific value similar to that of diesel, establishing a significant advantage and distinctive feature of the proposed fuel [168]. Furthermore, an investigation into evaporation and spray characteristics was conducted, revealing superior attributes for pine oil when compared to diesel. Additionally, the research introduced ignition promoters in the form of selective catalytic reduction and a catalytic converter to enhance engine performance [169]. Diesel was injected via the primary fuel injection system, and pine oil was added to the inlet manifold. Prior to starting engine tests, the research used suspended droplet studies to investigate the properties of pine oil evaporation while it was fumigated in the engine's inlet manifold. This provided important information about the evaporation of pine oil droplets at different temperatures [170]. Similar to this, Saravanan et al. investigated the impact of elevated fuel injection pressure on a diesel engine using a 30% pine oil biodiesel blend (P30%) [171]. P30 initially outperformed pure diesel in terms of brake thermal efficiency (BTE) at the typical 200 bar injection pressure. Nonetheless, better BTE and combustion characteristics resulted by raising the injection pressure by 50 bar increments from 200 to 350 bar. At 350bar, the P30 mix's peak BTE was 26.1%, 6.9% more than the diesel blends at 200 bars. A higher injection pressure led to better fuel atomization, quicker evaporation, and improved air-fuel mixture formation, which all decreased exhaust emissions (CO, HC, and smoke). In comparison to pure diesel, the P30 blend demonstrated the greatest decrease in CO and HC emissions under 100% load at 300bar, as well as combustion characteristics, with reductions of 16.6% and

13.1% at 200bar injection pressure. On the other hand, compared to diesel at 200bar, NOx emissions for the P30 blend at 350bar increased by about 9.4%. Consequently, raising the fuel injection pressure holds promise for optimizing engine performance and emissions. However, Venkatesan et al. findings indicate that as engine load increases, specific fuel consumption decreases by up to 4% for 100% pine oil [172]. At full load, for 50% Pine and 75% pine blends, they exhibited 8% and 10% higher brake thermal efficiency, respectively, compared to diesel. With a minor increase in NOx emissions, biofuel blends outperformed petro-diesel in terms of smoke, HC, and CO emissions. Diesel engines can safely use up to a 50% blend of pine oil biodiesel (50% Pine and 75% Pine) without experiencing any degradation in performance, emissions, or combustion characteristics.

2.10.3 Eucalyptus Oil-Based Blended Fuel Operation

The examination of eucalyptus oil in the context of its potential as a sustainable fuel for diesel engines has garnered attention due to its distinctive properties that bridge the characteristics of diesel and gasoline, as indicated in prior research [54]. Nonetheless, its relatively low cetane number presents a challenge, rendering it unsuitable as a standalone fuel for unmodified diesel engines. A full substitution of diesel with eucalyptus oil is unfeasible, primarily due to its delayed start of combustion (SOC), which can significantly impact engine efficiency, combustion quality, and emission profiles. However, blending eucalyptus oil with either diesel or biodiesel presents a viable alternative. It is noteworthy that eucalyptus oil, unlike alcohols, boasts a calorific value closer to that of diesel, making it more amenable for a substantial replacement ratio. For detailed insights into the utilization of eucalyptus oil blended with diesel or biodiesel. The exploration of eucalyptus oil as a sustainable fuel option for diesel engines gained prominence through the research of Devan et al. [54], who combined eucalyptus oil with the methyl ester of paradise oil to test a diesel engine. Their research departed from conventional diesel use, focusing instead on two alternative fuels. The outcomes of their study showcased a substantial reduction of 49% in smoke emissions, 34.5% in hydrocarbon (HC) emissions, and 37% in carbon monoxide (CO) emissions. Notably, a blend consisting of equal parts of paradise oil methyl ester and eucalyptus oil (Me50–Eu50) experienced a modest 2.7% increase in nitrogen oxide (NO_X) emissions at maximum load, while exhibiting combustion characteristics closely resembling those of diesel and a notable 2.4% increase in brake thermal efficiency (BTE). Rao et al. [173] have similarly endeavoured to use only renewable fuels. The authors examined the efficacy of methyl ester-based palm kernel oil and eucalyptus oil in a single-cylinder diesel engine. In this scenario, biomass-derived eucalyptus oil took the lead role, supplemented by the methyl ester of palm kernel oil to enhance its ignition quality. Their findings highlighted the improved fuel properties of eucalyptus oil, resulting in reduced gaseous emissions and heightened engine efficiency. Following the positive outcomes of blending eucalyptus oil with biodiesel, subsequent experiments involved blending it with traditional petroleum diesel for testing in diesel engines. Anandavelu et al. study [174] demonstrated that diesel could be blended with eucalyptus oil up to 40% to achieve enhanced engine performance and reduced emissions, including smoke, HC, and CO. However, the lower cetane value of eucalyptus oil led to an extended ignition delay, resulting in an elevated peak heat release rate and in-cylinder temperatures, which subsequently contributed to increased NO_X emissions. Various methods have been explored to mitigate NO_X emissions, with Anandavelu et al. [103] advocating for Exhaust Gas Recirculation (EGR) as a practical solution. Their study used eucalyptus oil-diesel blends to power a mono-cylinder diesel engine with exhaust gas recirculation (EGR), recirculating 15% of the exhaust gas.

They discovered that this had little effect on the engine's performance, emissions, or combustion characteristics. Remarkably, even with EGR engaged, there was no appreciable drop in performance; in fact, the thermal efficiency increased by 3.8% when compared to regular diesel. Moreover, because eucalyptus oil naturally contains oxygen, it naturally produced less smoke; this reduction in smoke was further enhanced by adding more eucalyptus oil to diesel. The study also shows that the blend's NO_X emissions are significantly lower than those of regular diesel, by 65%. Eucalyptus oil has many benefits, but when combined with diesel in large quantities, its low cetane number causes problems even if its energy content is similar to that of diesel. In response to this challenge, Tamilvendhan et al. [175] endeavoured to determine the optimal blend ratio of eucalyptus oil to diesel. They employed the Taguchi method to optimize engine operation parameters, with a specific focus on injection timing and pressure. After pinpointing the optimal parameters, an Analysis of Variance (ANOVA) was conducted to validate the preferred blend composition's consistency across these parameters. The Taguchi method results indicated that an injection pressure of 180bar, an injection time of 29° before top dead center (BTDC), and a blend of 40% eucalyptus oil and diesel are the optimal parameters for running a diesel engine on eucalyptus oil. Subsequent engine tests conducted under these optimized conditions revealed that a blend containing 40% eucalyptus oil resulting in a 2.5% enhancement in BTE in comparison to fossil diesel.

2.11 Fumigation Process Viability with Low-Viscosity Fuels

2.11.1 Alcohol base Fumigation Process

While the use of alcohol blends is recognized as a straight-forward method for integrating alcohol-based fuel in diesel engines, their long-term performance and dependability remain uncertain. Techniques such as emulsification and the inclusion of lubricants have been proposed as solutions to prevent separation and leakage issues. However, the use of these blends in higher ratios is restricted, primarily because their reduced calorific value leading to a decline in engine power. This challenge is particularly pronounced with alcohol-biodiesel combinations, given that both fuels possess lower energy content than diesel. In response to these challenges, scholars have proposed altering engine designs as a method, even if it necessitates more extensive changes, to efficiently utilize these fuels. This section will give an overview of the methods that researchers have used to use alcohol fuels in diesel engines other than just mixing them with diesel or biodiesel. Since, full insights on using alcohol fuels through engine modifications have not yet been consolidated.

When operating on two fuels, the principal injection system is used to deliver diesel, which serves as the pilot fuel. The inlet manifold is used to introduce alcohol fuel, which has a lower cetane number, as the major fuel. The dual fuel strategy employing alcohols has thus far been documented through two distinct mechanisms: fumigation and dual injection. Traditional methods including carburetion, continuous injection, and sequential injection are used in historical research on alcohol fumigation in diesel engines to distribute the fuel in the input manifold [176]. On the other hand, dual fuel injection incorporates an independent fuel injection system, comprised of elements like a fuel pump, filter, damper, nozzle, and pressure regulator [177].

2.12 Dual fuel mode of operation for pine oil

Extending their series of investigations Vallinayagam et al. [169,170] research assesses the emission-reducing capabilities of pine oil as a biofuel in a single-cylinder diesel engine. While pine oil's environmental benefits as a bio-derived fuel are evident, its comprehensive utility in diesel engines remains unexplored. Eucalyptus oil and ethanol exemplify biofuels that exhibit similar characteristics to pine oil, including low viscosity and boiling point. However, due to its low cetane number, it requires aid with ignition, and because of its viscosity, the fuel injection system must be adjusted. On the other hand, pine oil inductance into the inlet manifold through fumigation, utilizing diesel as the primary ignition source rather than blending it with the fuel. The findings show that at low and full loads, pine oil could replace 60% and 36% of diesel, respectively. With just slight increases in NO_X, smoke emissions are 64.2% lower than those of ordinary diesel. At maximum load, CO and HC emissions experienced a notable decrease of 67.5% and 47.8%, respectively, but exhibited an increase at lower loads. Nevertheless, there was an overall reduction in the average emissions of smoke, CO, and HC.

Upon closer examination, both fumigation and dual fuel injection bring to the table a plethora of benefits owing to their uncomplicated design and the versatility they offer in transitioning between standard and fumigation operations[178-179]. While there are minor differences between the injection strategies and the way lower cetane index fuels are introduced into the combustion chamber for fumigation and dual fuel injection, they all share one important benefit: they can replace a large amount of diesel fuel. This level of diesel replacement has not been fully achieved using conventional alcohol-diesel blends.

2.13 Methanol based Fumigation Process

In addition to safflower oil biodiesel (B100), Thiyagarajan et al. [180] investigated the effects of pre-injecting methanol and pentanol into the input manifold as a diesel substitute. A single-cylinder CI engine's performance, emissions, and combustion are evaluated through experiments. B100 was used to inject methanol and n-pentanol at 10% and 30% mass ratios, respectively. Results reveal higher brake thermal efficiency compared to pure biodiesel when using methanol and n-pentanol. For 10% mass ratio, n-pentanol outperforms methanol, but the trend reverses at 30% mass ratio due to methanol's higher heat release. Alcohol infusion resulted in higher emissions of HC, CO, and NOx, while lower emissions of smoke because of better mixture formation and less biodiesel. Popa et al. [181] proposed two distinct methodologies to employ methanol in fumigation mode, taking into account its inferior cetane value and its implications on spontaneous ignition. Their initial method incorporated a carburetor alongside the primary diesel injection system. Conversely, their second strategy utilized separate injection systems for methanol and diesel. Through their experiments on methanol fumigation employing these two techniques, they observed enhanced power outputs. Furthermore, at elevated loads, both NO_X and smoke emissions declined.

Kumar et al. [182] embarked on introducing methanol into the inlet manifold of a mono-cylinder diesel engine via fumigation. Recognizing that methanol could induce a cooling effect when sprayed into the inlet manifold, they integrated a heat exchanger to elevate the temperature of the incoming air. Consequently, methanol was dispersed near the inlet manifold aided by the heat exchanger. Comparing their trial results to conventional diesel, they observed a slight improvement in performance along with significant reductions in both smoke and NO_X emissions. Liu et al. [183] investigated a dual-fuel system using methanol and diesel. They used a methanol injector that was electronically controlled and installed in the inlet manifold around 250 mm in front of the inlet valve. In the interim, the primary injection system used diesel as the pilot fuel. To guarantee efficient fuel vaporization in this study, an electric heater was incorporated into the input manifold with the intention of preheating the incoming air. They reported increased levels of HC and CO emissions and suggested refining fuel distribution techniques to reduce them, which was in line with previous study findings.

In a related study, Song et al. [184] pursued the dual-fuel approach using methanol, building on the methodology introduced by Liu et al. [180]. They advised administering methanol predominantly during full load scenarios, as opposed to low-load situations. This was attributed to methanol's inherent propensity for lean combustion and its extended ignition delay. Even though reductions in NO_X and smoke emissions are recorded, there was a rise in HC and CO emissions. Concluding their findings, the team posited that the methanol-diesel dual-fuel arrangement might be optimal for stationary diesel engines, like generators. This is attributed to the capability of these engines to operate under consistent high-load conditions, maximizing their potential advantages.

A DMCC system was created by Yao et al. especially for naturally aspirated and turbocharged diesel engines [185]. The engine was configured to function in two modes: conventional diesel at low loads and dual fuel at high loads. Methanol was injected into the intake manifold using an electronic fuel injection system that had a relatively low-pressure threshold of 0.3MPa for dual fuel operation. The outcomes showcased simultaneous reductions in NO_X and smoke emissions. However, there was an elevation in CO and HC emissions, attributed to the cooled combustion environment resulting from the lower inlet air temperature.

In a subsequent investigation, with the aim of mitigating HC and CO emissions, Yao et al. [186] brought back their DMCC system with an oxidation catalytic converter installed. Surprisingly, there was a notable decrease in HC, CO, smoke, and NO_X emissions when the oxidation catalyst and DMCC system are paired, all without negatively impacting engine efficiency. Delving deeper into combustion dynamics, the DMCC-fuelled engine exhibited a more dominant premixed combustion stage and a subdued diffusion combustion phase, a result of the extended ignition delay.

While there have been significant strides in harnessing the advantages of fumigation and effectively controlling HC and CO emissions, Zhang and his team [187] addressed a persistent uncertainty concerning the impact of methanol fumigation on particulate emissions. They used a four-cylinder, naturally aspirated engine fitted with a Diesel Oxidation Catalyst (DOC) for their research. Specifically, the emphasis was on evaluating the amount of particulate matter in the exhaust during the fumigation procedure. Their findings were a drop in smoke, CO, HC, and NO_X emissions. Notably, they observed a reduction in particulate matter, characterized by a decline in both mass and number concentrations of particles. However, there was a noticeable uptick in the mean size of these particles, an effect not seen when the DOC was excluded in a follow-up research endeavour. Zhang et al. [188] further scrutinized the implications of methanol fumigation by exploring both conventional and non-conventional emissions. The study shed light on the possible adverse consequences arising from the release of volatile hydrocarbons and minute particle emissions when diesel engines operate on oxygenated fuels. This underscored the importance of a thorough examination of such emissions, including those typically not regulated. Their comprehensive analysis unveiled that the use of methanol in fumigation led to an increase in emissions such as xylene, toluene, benzene, formaldehyde and unburnt methanol, while emissions of substances like ethyne, ethene, and 1,3-butadiene saw a decrease.

Extending their series of investigations, Zhang's group [189], in a subsequent study, delved into the combustion patterns associated with methanol fumigation in engines. Their assessments indicated that fumigating with methanol led to an extended ignition delay, culminating in a heightened heat release rate. Intriguingly, there are variances in the internal pressures of the cylinder: while there was a decrease at lower to mid-range loads, a surge was evident at higher loads.

Cheung et al. [190] conducted pilot fuel experiments with biodiesel made from leftover cooking oil in an attempt to find a way to avoid using diesel altogether. The inlet manifold was charged with methanol using an electronically controlled low-pressure injector. The thermal efficiency of methanol fumigated with biodiesel as the pilot fuel decreased noticeably at lower loads. However, BTE remained relatively constant at medium and higher loads. The utilization of biodiesel, distinct from methanol fumigation blended with diesel, led to a discernible increase in NO_X and particle emissions. CO and HC emissions also rose at the same time.

To curb the rising HC and CO emissions, Cheung and collaborators incorporated a Diesel Oxidation Catalyst (DOC) in subsequent methanol fumigation experiments [191]. Their research ambit was broadened to include not only standardized emissions but also non-standardized emissions. Their results shed light on the DOC's influence over both types of emissions. The consensus from their findings was affirmative: the DOC effectively mitigated emissions like HC, CO, unburnt methanol, and formaldehyde. Aligning with Cheung's discoveries, Hosoya et al [192] also proved that using a Diesel Oxidation Catalyst (DOC) during methanol fumigation reduced the emissions of HC and CO. In an effort to discern the impacts of utilizing methanol in blended form versus fumigation, Cheng and colleagues [193] investigated the emission characteristics of diesel engines operating on mixtures of methanol and biodiesel in comparison to inlet manifold methanol fumigation using biodiesel as the pilot fuel. The researchers identified distinct operational outcomes for the two methods: the fumigation technique resulted in superior brake thermal efficiency during medium and high load operations, while the blend approach proved more effective at lower loads. However, observable differences are noted in primary emissions, including HC, CO, smoke, and NO_X. The fumigation technique revealed increased smoke, CO, HC, and

NO_2 emissions. This increase was explained by the methanol's cooling action in the inlet manifold, which led to a drop in in-cylinder temperature. The analysis of combustion dynamics revealed a clear prioritization of the premixed combustion phase over the diffusion combustion phase in both operating modes, indicating similar combustion characteristics between the two modes. Lu and colleagues [194] used a different strategy to use methanol in a dual fuel setup, adding methanol to the input port and supplying gas-to-liquid (GTL) fuel through the primary injector. To effectively orchestrate dual fuel injection, an Engine Control Unit (ECU) that managed methanol injection was harmonized with a crankshaft encoder and an array of sensors. The investigation reveals that the decreased cetane number of methanol causes a noticeable prolongation in ignition delay. Consequently, there was a rise in the heat release rate, coupled with a decrease in in-cylinder pressure. Furthermore, it was observed that the maximum pressure growth rate increased with the premixed ratio during operations at low to medium equivalency ratios, reaching a peak and then declining. However, notwithstanding this peak, the maximum heat release rate continued to grow steadily as premixed ratios increased. The use of GTL also produced a simultaneous drop in smoke and NO_X emissions, even as emissions of HC and CO increased. This is comparable to the use of diesel or biodiesel in conjunction with methanol fumigation.

2.14 Fumigation operation for Ethanol

In a study by Abu-Qudais et al. [164], ethanol was introduced as a fine mist into the inlet manifold for fumigation. This was accomplished using a single-hole nozzle with a diameter of 0.25mm, positioned 50mm before the inlet valve to ensure optimal air/fuel mixing. Intriguingly, they integrated an optical entry point in the inlet manifold, offering a view of ethanol's vaporization. Ethanol's flow rate for injection was moderated by an air compressor. Notably, they observed an uptick in volumetric efficiency stemming from the drop in air temperature, attributed to ethanol's pronounced latent heat of vaporization. When comparing, fumigation replaced 20% of diesel, whereas only a 15% replacement was noted with diesel-ethanol blends.

Chauhan et al. [195] decided to permit ethanol fumigation in a smaller diesel engine using a constant volume carburetor, after which it was assessed for emissions. They managed the volume of ethanol delivered by the carburetor using a butterfly valve, achieving mixes of 3% and 48% ethanol. The engine was tested under varied load conditions; as the load increased, only diesel consumption shifted, keeping ethanol volume consistent to sustain engine speed. For different ethanol injection rates, the engine manifested enhanced performance, registering reduced emissions like NO_X, CO, and CO_2, though HC emissions surged. The authors pinpointed elongated combustion duration at lower and intermediate loads relative to higher loads. They found the ethanol-air blend's interaction with diesel to be most effective at lower loads, given ethanol's diminished surface tension. While achieving a peak ethanol injection of 48%, they identified 15% as the optimal value after assessing overall engine emissions and performance.

Ajay et al. [196], fumigated ethanol using a Solex down-draught carburetor placed in the inlet manifold. The inlet air was warmed to 50°C to aid in the evaporation of ethanol inside the input manifold. The ethanol flow rate was controlled during low-load operations by taking the fuel pump's rack position, altering the quantity of replaced diesel from low to high load. As a result, the authors witnessed an increase in both Brake Power (BP) and brake thermal efficiency, accompanied by reduced exhaust emissions, as vaporized ethanol/air mixtures are introduced into the engine.

In a study by Babu et al. [197], when ethanol and air are introduced into an engine's inlet manifold, the authors saw a simultaneous decrease in NO_X and smoke emissions.

Their findings highlighted a 20% ethanol to diesel ratio as the most effective, taking into account engine performance and emission outcomes. The differences in combustion between dual-fuel engines and their gasoline and diesel counterparts are underscored. While combustion in gasoline and diesel engines is localized, it takes place throughout the combustion chamber in dual-fuel engines. This extensive combustion leads to reduced peak in-cylinder temperatures, thereby mitigating NO_X emissions. Moreover, the uniform distribution of the air-fuel mixture within the chamber contributed to a decrease in soot emissions. The authors identified a 49% reduction in NO_X and a 62% reduction in smoke emissions at a 20% ethanol to diesel mix, substantiating their claims. Tsang et al. [96] confirmed these results, highlighting similar outcomes when experimenting with different ratios of ethanol in diesel engines.

Hebbar et al. [198] looked into the use of ethanol-diesel blends in a single-cylinder diesel engine fitted with an exhaust gas recirculation (EGR) system. The findings indicated that the use of EGR decreased the thermal efficiency and increased emissions of CO and HC. This was explained by the fact that exhaust gas dilutes intake air, which always affects power production. Interestingly, BTE improved when ethanol was fumigated rather than mixed. They determined that 10% ethanol injection was the most effective in achieving a balanced combination of engine performance and emission outcomes. Bo et al. [199] adopted a unique methodology, instead of direct ethanol fumigation, the authors heated ethanol to initiate pyrolysis and introduced the resultant gas into the engine's inlet manifold. They achieved this by routing ethanol through a screw pipe connected to the exhaust pipe. When necessary, external heating was employed, and the resultant gas was cooled prior to integration into the air intake system. They praised this approach for its simplicity and minimal engine modifications, coupled with a significant 16% reduction in brake specific fuel consumption.

Goldsworthy et al. [176] investigated an electronically controlled Cummins engine using aqueous ethanol fumigation. The introduction of ethanol occurred post-turbocharger, utilizing the hot compressed air to enhance evaporation. Leveraging ethanol's inherent traits, such as extended ignition delay and high peak heat release rates, a two-stage injection process was employed to mitigate NO_X emissions. This modification effectively alleviated the ignition delays associated with ethanol, enhancing the thermal efficiency through optimized injection strategies. Remarkably, the presence of water in the ethanol contributed to reduced NO_X emissions due to lowered in-cylinder temperatures. However, there was a corresponding rise in CO and smoke emissions with an increasing ethanol proportion. Remarkably, they observed a phenomenon akin to knocking when ethanol injections surpassed 30%. With the prior injection of ethanol before the main diesel injection, the combustion of the ethanol-air mix began prematurely. Due to the rapid combustion of the ethanol-air mixture following the delay, there was a notable surge in in-cylinder pressure observed at 31% and 34% ethanol injections. Two separate heat release peaks were observed, the second of which was noticeably lower due to oxygen scarcity that occurred before to the liquid fuel jet combustion.

Zhang et al. [200] conducted a comparative analysis of the impact of methanol and ethanol fumigation on a single-cylinder diesel engine's performance, as well as its gaseous and particulate emissions. Their research revealed that methanol fumigation influenced several variables, including air-fuel ratio, combustion processes, in-cylinder temperature, and subsequent emissions. In comparison to ethanol fumigation, methanol fumigation resulted in higher brake thermal efficiency and reduced NO_X and smoke emissions. However, methanol fumigation had its drawbacks, leading to elevated HC and CO emissions when compared to ethanol fumigation.

2.15 Feasibility of low-viscosity fuels in exclusive fuel mode

Using low-viscosity fuels exclusively in a diesel engine presents several challenges. Nevertheless, researchers have explored various methods to integrate these fuels into diesel engines by modifying the engine's design and operational parameters. The diverse approaches employed by scholars in their work with alcohols, pine oil, and eucalyptus oil are detailed in the following sections.

2.16 Alcohol-based exclusive fuel mode operation

Although using ethanol as the primary fuel in a diesel engine presents certain operational challenges, Nagarajan et al. [201] explored its potential. They recommended the use of an injector nozzle with a larger surface area and a high-speed injection pump to address the calorific value disparity between diesel and ethanol. Their findings indicated a reduction in smoke and NO_X emissions, coupled with an enhancement in brake thermal efficiency. However, compared to diesel, there was a greater release of carbon monoxide and hydrocarbons. An extended ignition delay was observed with ethanol, attributed to its lower cetane number, resulting in increased maximum pressure rise and peak heat release rates.

In a similar vein, Maurya et al. [202] enabled exclusive use of ethanol in a diesel engine through the application of homogeneous charge compression ignition (HCCI) technology. Their method included changing the fuel-to-air equivalency ratio and preheating the intake air to a temperature between 120°C and 150°C in order to stabilize combustion. Ethanol was introduced upstream of the intake valve, and a heater was placed in the intake manifold, with a separate circuit regulating fuel injection timing and duration. According to the results, ethanol in HCCI mode had an indicated mean effective pressure (IMEP) of 4.3 bar, or 97.45% gas efficiency. Moreover, the suggested thermal efficiency was 44.78% at a relative air fuel ratio of 2.5 with an input air temperature of 120°C. Notably, emissions of CO and HC increased, but emissions of NO_X decreased dramatically.

The usage of 100% ethanol in a single-cylinder diesel engine with port injection was studied by Zhang et al. [203] and Xie et al. [204]. The HCCI mode exhibits a distinctive combustion process even if ethanol functions in two fuel modes with distinct combustion characteristics. HCCI mode exhibits its distinct combustion process. For example, Fig. 2.1 shows how fuel-air mixing procedures and combustion zones differ in engines with HCCI, dual fuel injection, CI, and SI. When diesel auto-ignites, a premixed air/fuel mixture ignites in dual fuel conditions. On the other hand, HCCI mode works differently from gasoline engine ignition in that fuel is spread throughout the combustion chamber to guarantee even combustion.

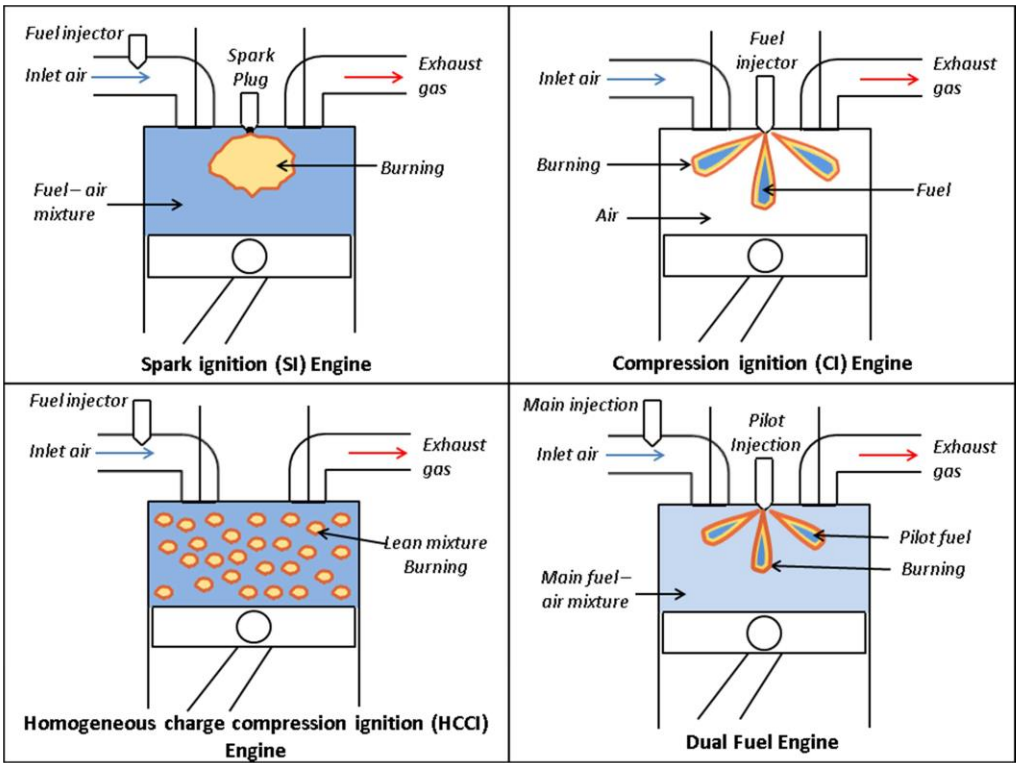


Fig. 2.1 Illustrates a comparative analysis of the combustion processes in dual fuel engines, fumigation with SI, CI and HCCI engines[16].

Liu et al. [205] designed a constant-volume combustion chamber to simulate conditions similar to a diesel engine. This chamber incorporated an electronic injector, pressure transducer, flame brightness monitor, electric heater, and other necessary components.

Their goal was to study the operation of butanol and soy biodiesel as primary fuels. During this research, they maintained ambient temperatures at either 800K or 1000K. Furthermore, at specific air densities, oxygen levels are adjusted to 21%, 16%, or 10.5%, which corresponded to zero, medium, and heavy EGR conditions, respectively. The results show that soy biodiesel had a lower peak pressure than butanol. Interestingly, even with its lower cetane number, butanol's auto-ignition remained consistent despite oxygen level variations between 16% and 10.5%. At 800K, soy biodiesel's soot production reduced as oxygen levels decreased, while butanol produced negligible soot. However, at 1000K, declining oxygen concentrations led to increased soot formation for both fuels.

In addressing the challenges associated with using ethanol as the sole fuel in diesel engines, Karthikeyan et al. [206] tackled two primary issues: the elevated self-ignition temperature and the low cetane index of ethanol. To overcome the former challenge, they introduced a glow plug, ensuring that a portion of the injected ethanol interacted with the heated surface for ignition aid. To mitigate the latter, engine parts are coated with zirconia, known for its limited thermal conductivity. This approach conserved the heat from the combustion process, reducing the ignition delay, thereby facilitating better combustion.

2.17 Eucalyptus oil in exclusive fuel mode operation

Utilizing eucalyptus oil in engines presents challenges due to its lower cetane value, akin to alcohols. This means its ignition isn't straightforward, and when employed at 100% load, an auxiliary ignition source becomes essential. To harness the full benefits of eucalyptus oil and run it as the primary fuel, Tamilvendhan et al. [90] improved the fuel's auto-ignition by integrating an air pre-heater into the input manifold. Notably, elevating the air temperature has a significant positive impact on the combustion of eucalyptus oil. Experimental trials with diesel engines operating solely on eucalyptus oil demonstrated a 50% reduction in smoke emissions across various load conditions, supporting this observation. Meanwhile, the brake thermal efficiency remained comparable to diesel. It is essential to acknowledge that as air temperature increases, its density tends to decrease, potentially affecting the engine's volumetric efficiency; an aspect that warrants further investigation.

2.18 Conclusions of literature review

Fuels with high viscosity often raise concerns about long-term issues like contamination of lubricating oil and accumulation of soot. However, blended fuels that incorporate alcohols don't present these challenges, primarily because of their reduced C/H ratio. Moreover, these alcohol blends showcase improved atomization and spray properties, resulting in enhanced engine performance and more efficient combustion. As oxygen-rich fuels, alcohols facilitate better fuel oxidation, leading to more favourable emission profiles. Given these benefits, alcohols show promise as potential alternatives for diesel engines, provided challenges such as low cetane number and high cetane number, elevated ignition temperatures, and lubrication concerns can be addressed. Nevertheless, their lower energy content restricts the extent to which they can be blended. Looking ahead, solutions will need to be developed to navigate this limitation.

At this point, it's vital to amplify research efforts exploring other plant-derived biofuels, especially those similar to pine and eucalyptus oil. This is because they can potentially be used in greater volumes due to their superior energy content compared to alcohols. Interestingly, there haven't been reports indicating separation problems between pine oil, eucalyptus oil and diesel, as seen with alcohol-based fuels. Despite the chemical differences between pine oil and eucalyptus oil and substances like ethanol and methanol, there's a need to study the blending compatibility and bond strength of pine oil, eucalyptus oil with diesel.

In the comprehensive review of the application of low-viscosity fuels in dual-fuel mode, encompassing a synthesis of pertinent experimental studies, it is noteworthy that while ethanol and methanol are frequently utilized in fumigation or dual-fuel injection strategies, the exploration of butanol in this context has been relatively limited. Given that alcohol-fuel mixtures tend to burn lean in typical methanol and ethanol conditions, engines often encounter cold start problems at lower loads. While the natural oxygen content of alcohols can enhance efficient combustion at higher loads, it may dilute the air-fuel mixture at lower loads, potentially impeding combustion. Furthermore, alcohol's prolonged ignition delay might impede combustion at lower loads, lowering the thermal efficiency. This could negatively impair fuel efficiency and combustion, especially when combined with the cooling effect brought on by the increased latent heat of vaporization. As a result, many experts think that the best course of action is to use alcohols as diesel substitutes at high loads and to stop injecting alcohol into the engine at reduced loads so that it runs just on diesel. With the aid of an electronic injection system, diesel injection can be varied along with progressively increasing alcohol injection to replace more diesels. Nevertheless, precautions must be in place to prevent knocking as alcohol proportions rise.

The exploration of pine oil and eucalyptus oil as a viable dual fuel option is still in its infancy. Moreover, various fumigation methods such as carburation, nozzle spraying, and split injection haven't been extensively tested with it. Thus, when juxtaposed with lower alcohols like ethanol and methanol, there's a noticeable gap in methanol fumigation by carburettor research for pine oil and eucalyptus oil blend from main injector, even though numerous studies have explored its different types of blending with diesel. Given this, it becomes crucial to devise strategies for the methanol fumigation by carburettor of pine oil and eucalyptus oil blend, be it through fumigation or dual-fuel injection. In conclusion, opting for methanol fumigation through a carburettor presents several advantages over blend operations. This approach effectively circumvents challenges such as fuel pump leaks and wear of injection equipment, attributed to the diminished lubricity of these low-viscosity fuels.

2.19 Research gap analysis

Taking into account the literature review and the results covered in the previous sections, the following research gaps are identified:

- Few studies have been done on the engine properties of pine oil, eucalyptus oil, and their combination thus far.
- The technique of blending and characterizing pine and eucalyptus oils is crucial in creating distinctive mixes. Nevertheless, there is a dearth of literature that describes the methods used in the manufacture and characterization of mixed pine and eucalyptus oils.
- One major problem when working with blended pine oil and eucalyptus oil is the agglomeration of the oils over time. There is not enough in-depth research in the studies that highlight storage stability.
- There is a dearth of research on the igniting properties and evaporation duration of blended dispersed fuels containing pine and eucalyptus oils.
- The impact of blending pine and eucalyptus oils on the macro and micro spray properties of diesel-biofuels blends is still unknown. The majority of studies have used animal fat and vegetable oil to produce biodiesel, but there is still a problem with the viscosity of these feedstocks.

- Insufficient research has been done on the combination of alcoholic fuels, like pine oil and eucalyptus, with CI engines.
- These vegetable oils can be used straight as fuel in a CI engine without the need for modification.
- Higher NOx emissions are a result of FAME's oxygenation properties. Here, the use of pine and eucalyptus oil mix in an equal ratio (1:1) and find out the oxygenated and lesser oxygenated constituents present in biofuels blends with the help of GC-MS analysis.
- Insufficient research has been done on the droplet size (by Malvern spray tec) of the combination of pine and eucalyptus oil in the CI engine. An attempt has been made to link the oil ingredients with the blend's efficiency because qualitative and quantitative evaluations of these constituents are said to offer greater understanding.
- The auto-ignition temperature of standard diesel has not been investigated through methanol fumigation using a 1:1 blend of pine and eucalyptus oil.
- The majority of the literature does not adequately explain the mechanism underlying the enhancement of various engine properties by the addition of edible and non-edible biofuels.
- There aren't many researches that identify the ideal dosage of a blend of pine and eucalyptus oils to be diluted in base fuels like blends, biofuels, and diesel.

2.20 Research objectives

- To determine unutilized biofuels oil that can be directly used as a fuel in the CI engine.
- To prepare the blends of pine oil and eucalyptus oils with petroleum diesel and higher fuels.

- To determine the physicochemical properties of all test fuels.
- To determine sauter mean diameter of pine oil and eucalyptus oil mix with diesel.
- To determine separate as well as blend analysis of pine oil and eucalyptus oil constituents by GC-MS analysis.
- To determine performance emission and combustion characteristics of CI engine using prepared test fuels and compare with DI taking baseline data.
- To determine the effect of methanol fumigation by various percentages in engine combustion, performance, and emission in CI engine.
- To determine the clean combustion and effect of HC, CO, EGT, NOx, smoke opacity increase the BTE and reduce the BSFC.

2.21 Research plan and outline

The extensive literature review led to a unique blend of pine and eucalyptus oils, derived from pine tree resins. Notably, this blend hadn't been acknowledged as a viable substitute for diesel engines in prior research. Consequently, this research introduces this blend of pine and eucalyptus oils as a potential substitute for diesel, examining its operational viability. Given practical considerations such as drivability, user comfort, and other logistical aspects, a stationary diesel engine was selected for this study. This choice was made due to its operational adaptability and its suitability for methodical research. Beyond introducing the novel pine and eucalyptus oil blend, this study's objectives also extend to thoroughly testing its physical and thermal attributes, assessing evaporation and spray features, and conducting engine characterizations using three distinct approaches. A computational analysis of engine combustion and emissions also forms part of the approach. The subsequent sections will include an extensive breakdown of research with thesis layout.

Since this blend of pine and eucalyptus oil is a pioneering biofuel, with no precedent in scholarly literature, a GC-MS analysis was initiated to discern its components. Before delving into engine tests, the physicochemical attributes of the proposed blend are evaluated and contrasted against standard diesel and other existing biofuels. Assuredly categorizing the blend within the realm of less viscous fuels, foundational studies concerning fuel spray and evaporation are also undertaken to understand the underlying fuel dynamics more effectively. To this end, experiments involving suspended droplets are carried out, exploring evaporation features like droplet regression rates and evaporation timelines for the biofuels blend. Subsequent to this, analyzed spray attributes, such as the spray's reach and its conical spread, comparing the pine and eucalyptus oil blend with conventional diesel.

2.22 Thesis organization

The thesis entitled “Some Investigations on use of Renewable Fuel in Compression Ignition Engine” includes Five chapters which are as follows:

CHAPTER1: INTRODUCTION- The initial chapter delves into the global energy landscape, focusing on India's energy trajectory. The escalating energy consumption in India underscores the pressing need to identify alternative energy sources as fossil fuels pose severe environmental threats. Given the centrality of compression ignition engines to India's economy, there's a necessity to find substitutes for diesel. This chapter highlights the potential of vegetable oil as a viable alternative to diesel. Certain unmodified oils are found suitable for running diesel engines, paving the way for their transformation into diesel-compatible fuels. Distillation emerges as a method that can transform vegetable oil mixtures into fuels similar to pine oil and eucalyptus oil.

CHAPTER 2: LITERATURE REVIEW- Chapter two provides a comprehensive review of existing academic works. The discussion bifurcates into two main segments.

The initial segment explores the process of transforming pure vegetable oils into diesel-analogous fuels through distillation and GC-MS techniques, examining the impact of these methods on conversion efficiency. The latter segment evaluates the performance of engines powered by various biofuels blends. This chapter identifies research voids and delineates the objectives of the current study.

CHAPTER 3: SYSTEM DEVELOPMENT AND METHODOLOGY- Chapter three presents an exhaustive description of the processes involved in producing pine oil and eucalyptus oil. It further discusses how the production of these oils can be optimized. Instruments and techniques used to determine the properties attributes of the developed fuel are also covered. The chapter gives an insight into the experimental setup, including the Spraytec system to assess the Sauter mean diameter (SMD). It also elaborates on the engine test procedures, culminating in a discussion on measurement precision and uncertainties.

CHAPTER 4: RESULTS AND DISCUSSION-Chapter four scrutinizes the outcomes from tests conducted on the specially developed engine setup. Initially, tests involved a blend of pine and eucalyptus oil with diesel, followed by introduction of a distilled mix of both oils into diesel to observe its influence on engine efficacy. Tests were carried out with various biofuel blends along with the methanol fumigation via the inlet port combined with a blend of pine and eucalyptus oil. Detailed results are evaluated and juxtaposed against existing scholarly works.

CHAPTER 5: CONCLUSSION- In the concluding fifth chapter, pivotal research discoveries are consolidated, and potential avenues for future exploration are suggested in last chapter.

CHAPTER 3

SYSTEM DEVELOPMENT AND METHODOLOGY

3.1 Introduction

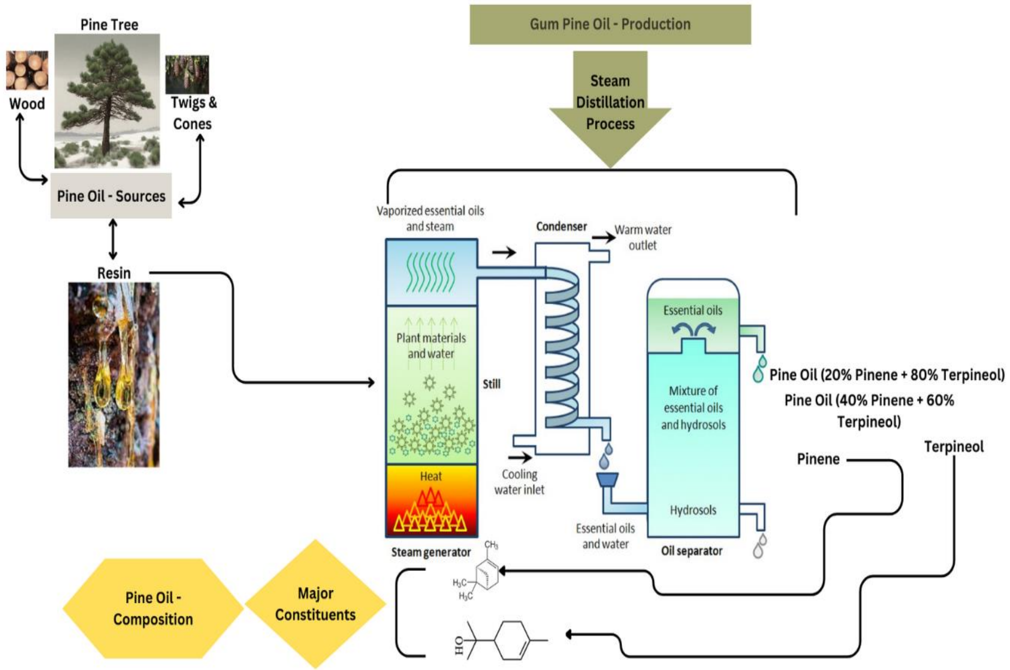
A wide range of literature was assessed to identify the optimal blend of low-viscosity fuel. The physicochemical properties of these low-viscosity fuels, such as density, kinematic viscosity, cetane number, calorific value, surface tension and flash point were assessed. Subsequent to obtaining data on these properties, the combustion, performance and emissions profile of both unmodified and modified engines was scrutinized. The modification of the conventional diesel engine was carried out to enable its operation with flexible fuels, encompassing both liquid and alcoholic options. This chapter delivers a comprehensive explanation of the instruments used for finding the physicochemical properties, engine components, and the innovations implemented in the engine to enable this fuel flexibility. The selection of two oxygenated biofuels was a crucial aspect of this exploration. The decision was grounded in crucial combustion characteristics, encompassing flame speed, auto-ignition temperature, adiabatic flame temperature, and calorific value. Additionally, factors such as accessibility, storage, and safety precautions were taken into account when opting for alcoholic fuels. Among the choices explored in the literature, pine oil and eucalyptus oil displayed promising combustion properties compared to other low-viscous alternative fuels. The final decision regarding feedstock selection was influenced by factors like cost, availability, and environmental considerations. As a result, pine oil and eucalyptus oil are selected as the primary feedstock for the current study.

3.2 Pine oil production

Pine oil is a notable renewable biomass-derived fuel source with a unique characteristic. It is derived from pine trees and can be blended with fossil diesel fuel.

Pine trees are produced for their bark, timber, tar, and essential oil, among other uses. They can grow up to 40 meters in height. Pine oil is a pale-yellow essential oil that is extracted from pine trees and has a forest-fresh scent. Pine oil output was estimated to be 30,000 tons year worldwide [207], and 953,894 tons are expected to be needed by 2025 [208]. Gum, wood, and sulfate are the three main types of pine oil; each is derived from a distinct component of the pine tree and has special qualities. Among these, gum pine oil is particularly significant. Pine oleoresin is the basic material present in it. It is astounding to consider that 2.75 kg of pine oléoresin, or about 65% rosin and 20% turpentine, may be obtained from a single pine tree. Pine oil is later made by processing the turpentine into oleoresin. Fig. 3.1 depicts the pine oil production process. Oleoresin, extracted by tapping pine trees, is the main ingredient used to produce pine oil. The oleoresins are cleansed and then added to a reactor that is encircled by coils of cylindric wire in which steam is passed. As the steam passes through, the oleoresins separate into fumes of turpentine and rosin.

The oleoresins are split into rosin and turpentine vapours as the steam flows through; the latter are sent into a condenser, where liquid turpentine is collected. With low-boiling fraction chemicals like β-pinene and α-pinene as its main ingredients, this turpentine functions as a biofuel oil on its own. The residue that remains after the low-boiling fraction compounds are isolated from the oleoresins is called rosin and it looks like camphor. To produce pine oil, the turpentine undergoes a reaction with ortho-phosphoric acid, ultimately leading to the collection of pine oil as an essential oil.



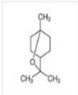
Major Constituent	3,7-dimethylocta-1,6-diene (Citronellene) ; 1,8 Cineole ; 1,4-dimethyl-5-(1-methylethyl)-cyclopentane ; bicyclo[2.2.1]heptane ; 3,7,7-trimethylbicyclo[4,1,0] hept-3-ene (Carene) ; Cyclohexane (Limonene) ; 2-oxabicyclo[2.2.2]octane ; (+)-alpha-terpineol p-menthl-1-en-8-ol
Chemical Formula	$C_{10}H_{18}O$
Chemical Structure	

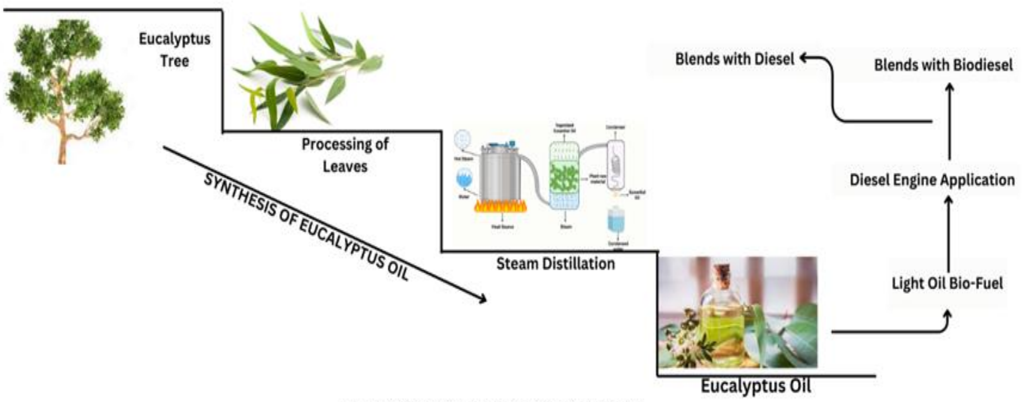
Fig. 3.1 Pine oil- Source, production and composition.

3.3 Eucalyptus oil production

Eucalyptus oil, a renewable plant-based fuel source, has gained recognition as an alternative fuel, leading to extensive research into its optimization for utilization in diesel engines, alongside other biofuels such as methanol and ethanol. As eucalyptus oil is taken from different portions of the eucalyptus tree, it is classified as a light biofuel, in contrast to triglycerides, which are usually derived from seeds. The eucalyptus tree belongs to the Myrtaceae plant family, and its oil is obtained through the process of steam distillation from its leaves. This study offers insight into the production process of eucalyptus oil from the leaves and highlights its potential as an effective co-solvent, owing to its chemical similarity to naphthenic bases. This property facilitates the blending of eucalyptus oil with other oils or solvents. In the context of diminishing petroleum fuel resources, researchers have explored the adaptability of eucalyptus oil as a viable alternative fuel. In addition to its historical use as a substitute for gasoline, recent studies have revealed its potential as a replacement for diesel fuel, with several researchers successfully demonstrating its operation in diesel engines.

The physicochemical properties of eucalyptus oil, as detailed in Fig. 3.2, aligns with its suitability for use in diesel engines. The principal component of eucalyptus oil, cineole ($C_{10}H_{18}O$), inherently contains oxygen within its molecular structure. Since, eucalyptus oil has a lower boiling point and viscosity than diesel, it facilitates fuel atomization, evaporation, and subsequent combustion processes. Eucalyptus oil is a possible alternative fuel source as it has a higher calorific value than diesel, which is crucial for ethanol's viability in diesel engines.

Despite the benefits of having better fuel qualities, it's crucial to remember that eucalyptus oil has a higher cetane number, which indicates better ignition quality.



EUCALYPTUS OIL - COMPOSITION DETAILS

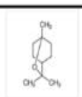
Major Constituent	3,7-dimethylocta-1,6-diene (Citronellene) ; 1,8 Cineole ; 1,4-dimethyl-5-(1-methylethyl)-cyclopentane ; bicyclo[2.2.1]heptane ; 3,7,7-trimethylbicyclo[4,1,0] hept-3-ene (Carene) ; Cyclohexane (Limonene) ; 2-oxabicyclo[2.2.2]octane ; (+)-alpha-terpineol p-menthl-1-en-8-ol
Chemical Formula	$C_{10}H_{18}O$
Chemical Structure	

Fig. 3.2 Synthesis of eucalyptus oil and its application in diesel engine

3.4 Test fuels blend

A blend of pine oil and eucalyptus oil, extracted from trees through a distillation process, was mixed with diesel in different proportions as part of this research. The research investigated various biofuels blend ratios, including 5%, 10%, 15%, 20% and 50% mixed with diesel. The test fuel samples are formulated using a volume-based approach. For example, in a one-litre test fuel batch, 100ml of pine oil and eucalyptus oil are mixed with 900ml of diesel, this resulted in a 10% blend of pine oil and eucalyptus oil with diesel. Similar methods are used to prepare other blend ratios. The prepared fuel samples are visually represented in Plate 3.1.

Plate 3.1 Blend Samples of Test Fuel

Furthermore, blends were formulated that incorporated pine oil and eucalyptus oil biofuel mixed with diesel. For a comprehensive overview of the test fuels used, including their names and composition, please refer to Table 4.2. Achieving homogeneity in the samples involved the use of a hand blender, which was operated at high speed with vigorous agitation. The test samples were monitored for a 60-day period to evaluate their uniformity and the possibility of phase separation. During this time, no signs of separation was detected.

Table 3.1 Test Fuel Names and Composition

Test Fuel Composition	Nomenclature
100% Diesel	D100
100% Pine oil	P100
100% Eucalyptus oil	E100
5% Pine oil, 5% Eucalyptus oil and 90% Diesel	P5E5D90
10% Pine oil, 10% Eucalyptus oil and 80% Diesel	P10E10D80
15% Pine oil, 15% Eucalyptus oil and 70% Diesel	P15E15D70
50% Pine oil and 50% Eucalyptus oil	P50E50
20% Eucalyptus oil and 80% Diesel	E20D80
20% Pine oil and 80% Diesel oil	P20D80
10% Distilled Pine oil, 10% Distilled Eucalyptus oil and 80% Diesel	PD10ED10D80
20% Distilled Eucalyptus oil and 80% Diesel	ED20D80
20% Distilled Pine oil and 80% Diesel	PD20D80

3.5 Techniques for Determining Physicochemical Properties

Table 3.2: Instrument used for determining the physicochemical properties

Properties	Manufacturer	Techniques	Operating Range	ASTM Standard	Measuring Device
Surface Tension	LAUDA	Du-Nouy ring method	0.75mN/m to 300mN/m (999mN/m with plate)	D971	Tensiometer-3 standard specified by
Density	Anton Paar	Oscillating U-tube	0 g/cm^3 to 1.5 g/cm^3	D4052	Density meter
Kinematic Viscosity	Petrotest	Time for the flow of the fuel sample in the capillary tube	+5°C to +150°C	D445	Capillary Tube - High Temperature
Calorific Value	Parr	Combustion of fuel at constant volume in the presence of oxygen	52 to 12000 calories	D240	Bomb Calorimeter
Flash Point	Pensky-Martens	Heating of the sample took place in a specialized test cup	up to 405°C	D93	Automatic Flashpoint Tester

The examination of the properties of the test samples is an essential part of developing a deeper understanding of the trends witnessed in the engine trials. To achieve this, a careful and thorough preparation process was employed to create 500ml of test samples for the detailed analysis of the test samples physicochemical properties. The following subsections will provide a detailed account of the various properties and the methodologies used for testing them.

3.5.1 Density

To determine the density of the test fuels, an Anton Paar density meter (DMA 4500 model) was utilized. The test equipment is illustrated in Plate 3.2. The device maintains a steady temperature of 15°C while measuring the test fuel's specific gravity in accordance with ASTM D-4052. To begin the measurement, 10 millilitres of toluene are injected via the sample injection port to rinse the test fuel line. The test sample is then injected into the same port in an amount of 10ml. The same specimen was evaluated three times for quality control purposes for repeatability, and the findings are deemed adequate. The final value was then calculated by averaging the three values.

Plate 3.2 The density-finding instrument

3.5.2 Kinematic viscosity

For measuring the kinematic viscosity of the test samples, a Petrotest viscometer, as depicted in Plate 3.3, was employed. The viscosity of the samples was carried in accordance with ASTM D-445 standard at a consistent temperature of 40°C. This involved filling a capillary tube with the test sample, marking both upper and lower level on the tube. The precise duration it took for the fuel to travel between the upper and lower-level marks was carefully recorded with the aid of a stopwatch. The kinematic viscosity of the specimen was computed by applying the time in seconds to the capillary constant, following the formula outlined in Equation 3.1.

$$v = k \times t \tag{3.1}$$

v =kinematic viscosity in mm^2/s and t is in seconds, Where k = 0.005675mm^2/s^2

Plate 3.3 Equipment was employed to measure kinematic viscosity

3.5.3 Calorific Value

A Parr 6100 oxygen bomb calorimeter, as indicated in Plate 3.4, was used to determine a liquid fuel's calorific value, which is a measure of the heat released during burning with oxygen. The procedure followed ASTM D-240 standards. It begins with placing a specific amount of fuel in a crucible, which is situated between electrodes. A nichrome wire bridges the electrode ends to complete the circuit. Afterward, the electrodes are hermetically sealed, and oxygen fills the bomb. The bomb is submerged in a water-filled container, and the entire system is positioned within the calorimeter to calculate the fuel's C.V.

Plate 3.4 Instrument for finding the calorific value

3.5.4 Flash Point

The fuel's flash point is the temperature at which a mixture of gasoline vapours and air can briefly ignite when exposed to a little flame. It employed an ASTM D-93 compliant Pensky Martens Automatic Flash Point. Apparatus in this investigation is depicted in Plate 3.5. The procedure involves placing the sample in a test cup, heating it at a regulated rate, and swirling it continuously. A tiny flame is directed in the direction of the cup at predetermined intervals, momentarily stopping the churning action.

Plate 3.5 Instrument for measuring the flash point

The term "flash point" refers to the lowest temperature at which, when a test flame is introduced close to the cup, the vapours generated above the sample ignite instantly.

3.5.5 Distillation

In line with the ASTM D-86 test method, a 100ml of the test sample was distilled using the equipment shown in Plate 3.6. The process involves placing the sample in a distillation flask and conducting distillation under specified conditions. The temperature at the moment was carefully recorded when the first condensate drop fell from the condenser's lower end. Temperature measurements are obtained in relation to the condensate volume at predetermined intervals, and the highest temperature at which complete fuel condensation occurred was also noted. Following that the temperature readings are translated into levels of conventional atmospheric pressure.

Plate 3.6 Device used to distil the test fuel

3.5.6 Cetane Index

To establish the Cetane Index, it follows the calculations described in ASTM D-4737, utilizing information about the density and boiling point temperatures of the test fuel. This approach works quite well, especially when the test engine is unavailable or the sample size is limited. The calculation process is based on equation 3.2.

$$\textbf{Cetane Index} = 45.2 + 0.0892 * T_{10X} + (0.131 + 0.901 * Y) * T_{50X} + (0.0523 - 0.42 * Y) * T_{90X} + 0.00049 * ((T_{10X})^2 - (T_{90X})^2) + 107 * Y + 60 * Y^2 \quad (3.2)$$

where,

$$T_{10X} = T_{10} - 215 \quad (3.3)$$

$$T_{50X} = T_{50} - 260 \quad (3.4)$$

$$T_{90X} = T_{90} - 310T_{10} \quad (3.5)$$

T_{10} represents the distillation recovery temperature at 10% (V/V) in °C, T_{50} corresponds to the distillation recovery temperature at 50% (V/V) in °C, and T_{90} indicates the distillation recovery temperature at 90% (V/V) in °C

$$Y = [\exp(-0.0035 * D_X)] - 1 \quad (3.6)$$

$$D_x = D_f - 850 \quad (3.7)$$

D_f is the fuel density at 15°C, in kg/m^3

3.6 Gas Chromatograph and Mass Spectrometer (GC-MS)

Gas-liquid chromatography is the most widely used technique for figuring out the chemical composition of volatile organic mixtures. In this method, a capillary column basically, a narrow tube meant for continuous flow within the instrument is used. Since, each component of the sample has unique physical and chemical characteristics, they move through the column at different rates in a gaseous environment. An inert gas, such as nitrogen or helium, is the mobile phase, which is what moves the sample up the column. There is a stationary phase inside the column where a liquid is chemically bonded or adsorption-immobilized on the surface of a solid support. By partitioning between them, the gaseous and immobilized liquid phases work together to break the sample down into its component parts. Electronic differentiation is made possible by the stationary phase's separation of the components, which forces them to exit the column at different times.

Plate 3.7 GC-MS Experimental setup

In a gas chromatography setup, essential components are carrier gas cylinder with a regulator, a gas flow controller, a sample injection port, the chromatographic column, a detector, and a recorder. Precisely heated to predetermined temperatures are the injection port, column, and detector. For the sample to pass through the column efficiently, its constituent parts must be in vapour form because the mobile phase is gaseous. The selection of the right column and the accurate determination of component temperatures are pivotal for the successful identification of mixture components using this method.

As shown in Plate 3.7, the Shimadzu QP-2010 system, which includes an Agilent DB-2887 column and a flame ionization detector, was utilized in the present study. A temperature of 350°C was maintained for the injector, via which liquid samples containing 0.1 μL are introduced. A temperature controller that could be programmed to precisely control the oven's temperature at a pace of 15°C per minute was used. The ASTM D-6584 test standard was followed during the testing process.

3.7 Experiment to determine the average particle size of the fuel spray

Fuel atomization is a key factor in determining the combustion process's efficiency. Several elements influence fuel atomization, such as fuel injection pressure, fuel density, and fuel viscosity. When the fuel has low viscosity and high injection pressure, it results in the formation of very fine fuel particles during the spray. These fine particles lead to a decreased ignition delay, thanks to their expanded surface area. However, their penetration capability is restricted due to their lower momentum and reduced velocity relative to the surrounding air. As a result, the combustion process utilizes less air. Consequently, during the diffusion combustion phase, more heat release occurs, leading to increased pressure rise.

Conversely, larger droplet sizes and lower injection pressures cause the second stage of combustion pressure rise to be lessened, which smoothens out the engine's operation. Therefore, achieving an optimal particle size is crucial for efficient engine performance . To measure the average particle size of the fuel spray, used a laser diffraction system. In particular, the Sauter mean diameter (SMD) of the fuel spray in Appendix I was determined in this work using the Malvern Spray tec instrument.

The laser Doppler effect is the foundation upon which the Spraytec technology operates. The analysis is centred on the beam scattering that particle cause as they move across a laser beam. One end of the analyser projects a laser beam, while the other end captures it. An illustration of the Malvern Spraytec device can be found on Plate 3.8. Based on the scattering pattern, one may determine the size and concentration of the particles, which are thought to have a spherical form. An injection mechanism was created to introduce fuel into the laser beam path, as seen in Plate 3.9. The test fuel sample is stored in a fuel tank that is part of this system. Plate 3.10 shows the process of moving gasoline from a tank to a common rail using a high-pressure fuel pump that is driven by a motor that is connected to the pump via a v-belt. The common rail is connected to the fuel injector with a high-pressure fuel line.

The gasoline injector used in this investigation had three holes, two of which are blocked to let fuel to spray through just one hole. The injector that was used in the trials is shown in Plate 3.11. The opening pressure of the fuel injector was consistently maintained at 200 bar, aligning with the injection pressure applied in the engine tests. The experimental setup's schematic is displayed in Fig. 3.3. Every test fuel experiment was run five times, and the mean outcome was calculated. The distance of 60mm was maintained throughout the studies between the injector and the laser beam. Even though a single injection took about 12ms, data gathering took place across 200ms. In order to give a clear comparison of droplet distribution in the middle of the injection procedure, the data given in this work is recorded at 6ms. All testing was carried out in low light in atmospheric circumstances to reduce the impact of sunshine. These conditions are chosen to focus primarily on the study of primary fuel atomization, which is mainly influenced by fuel properties, injector geometry, and the air density during fuel spraying. Fuel quality play a major role in determining the features of the fuel spray because in this study the injector geometry and air density were kept constant.

Plate 3.8 Experimental setup for the Malvern Spraytec

Plate 3.9 Experimental setup for the fuel tank

Plate 3.10 Fuel pump experimental setup with motor connection

Plate 3.11 Experimental setup Injector used in the setup

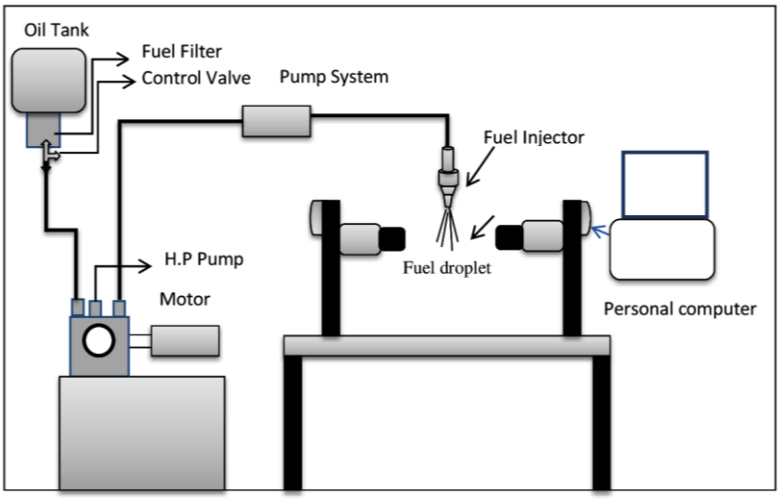


Fig. 3.3 Schematic of the spray analysis setup

3.8 Selection of the test engine

A Kirloskar diesel engine fitted with a water-cooled eddy current dynamometer was used to put up an experimental setup. According to the operational and technical specifications provided by the manufacturer, the engine was operated at a constant speed of 1500 rpm, delivering 6 kW of power. The engine used a mechanical fuel injection system; particular engine details are shown in Table II in Appendix II. A burette and stopwatch are used to measure fuel consumption rates in triplicate; the average was then utilized to make further calculations.

To minimize engine pulsations and maintain a consistent airflow, an injection tank was employed. To maintain a constant airflow via the orifice meter, a constant suction pressure was employed. An encoder was used to measure the crank angle position, and a piezoelectric pressure sensor was used to track the in-cylinder pressure. The signal was first boosted by a charge amplifier and then sent to an A/D converter, which then sent it to a personal computer. A specialized combustion analysis tool called "Enginesoft" generated data on pressure versus crank angle and heat release after processing the signals.

Using an AVL Digas 1000 gas analyzer, specifications given in Table III in Appendix III, the engine exhaust's NOx, HC, CO, and smoke opacity were analyzed. After the engine stabilized, the fuel was changed to biofuels mixtures and methanol fumigation. The engine was initially poured with clean diesel. The engine was loaded 20% at a time, starting at zero and working its way up to maximum load. Every reading was taken three times, then averaged. Following the biofuels experiments, the engine was operated again on diesel and run for fifteen minutes before being shut down.

The engine's performance was assessed using the exhaust gas temperature, brake-specific energy consumption, brake thermal efficiency, NOx, CO, and HC emissions, smoke opacity, heat release rate, and in-cylinder pressure. Diesel served as the baseline fuel and several biofuel blends were employed for the measurements.

Furthermore, methanol fumigation was studied on the same engine by mimicking the method used for droplet measurement in the manifold with an injection of methanol using a Keihin C.V. 135cc bike carburetor. The quantity of injected methanol was manually measured, and different flow rates were optimized to ensure that the auto-ignition of the P10E10D80 blends remained unaffected.

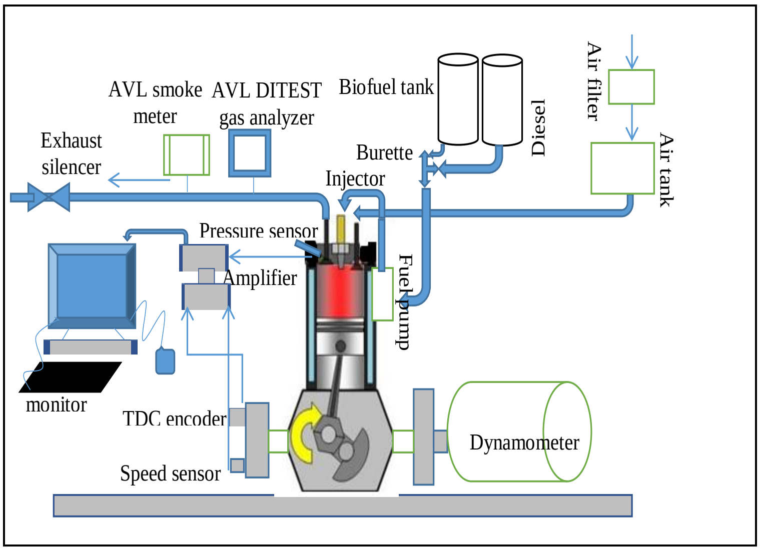


Fig. 3.4 Diagram depicting the setup of the engine experiment

3.9 Selection of the engine test parameters

The careful selection of the right set of test parameters plays a pivotal role in engine testing, and in this study, these parameters are meticulously chosen. The engine testing was carried out in accordance with the directives specified in IS:10000. The following lists provide details of both the observed and calculated parameters:

The observed parameters encompass:

1) Engine load
2) Engine speed
3) Air flow rate
4) Fuel consumption rate
5) Temperature
6) In-cylinder pressure
7) Exhaust emissions of HC, CO, CO_2, and NOx
8) Smoke Opacity

Here are the calculated parameters:

1) Brake thermal efficiency (BTE)
2) Heat release rate
3) Pressure rise rate
4) Brake specific energy consumption (BSEC)
5) Ignition delay
6) Combustion duration

3.9.1 Brake power measurement

When evaluating an internal combustion engine, the brake power (BP) is the most crucial factor. A flexible coupling was used to link the engine to an eddy current dynamometer. The engine's brake power was evaluated using a water-cooled eddy current dynamometer under varied loads and conditions.

The two major parts of an eddy current dynamometer are the rotor and the stator. Permanent electromagnets are incorporated into the stator, while a copper or steel disk makes up the rotor. A connection connects the rotor to the engine shaft. An electric current is run via the electromagnets to activate them when the engine is loaded. The electromagnets induce eddy currents when the rotor revolves. By creating a magnetic field that resists the rotor's rotation, these eddy currents load the engine. Water is circulated through the dynamometer to dissipate the heat produced during this operation. As shown in Plate 3.12, a moment arm exerts force on a load cell to measure the load imparted to the engine. The engine's power output is used to calculate the brake power, and the load is expressed in kilograms calculated according to the equation 3.8. The dynamometer's arm had a length of 0.185 meters.

$$\text{BP (kW)} = \frac{2*\pi*N(\text{rpm})*\text{Load(kg)}*9.81*\text{Dynamometer arm length (m)}}{60*1000} \qquad (3.8)$$

Plate 3.12 Load cell placed on the dynamometer

3.9.2 Engine speed measurement

A toothed disc was attached to the engine shaft at the dynamometer end. In close proximity to this disc, depicted in Plate 3.13, a magnetic pickup-type RPM sensor was installed. This sensor is made up of a coil, a yoke, and a permanent magnet. A pulse was created in the coil each time a tooth on the disk passed by the sensor. The sensor produced more pulses as the engine speed rose. The digital acquisition system received these pulses and used them to determine the engine's revolutions per minute (RPM). Then, the software interface that was loaded on the personal computer and the control panel both well the engine's RPM data.

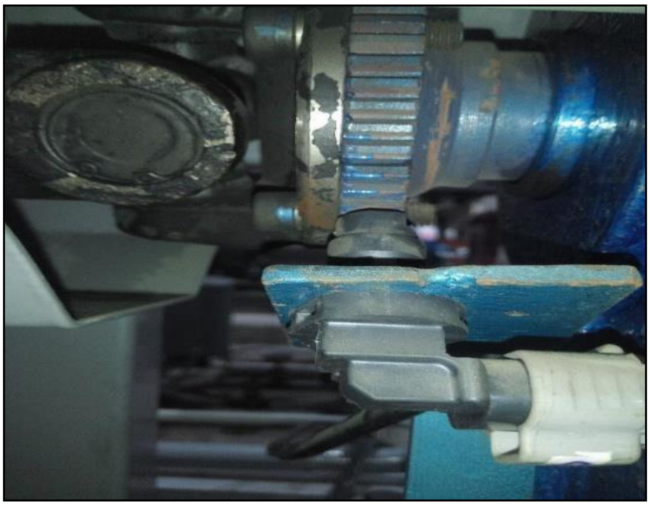


Plate 3.13 Instrument for measuring engine speed

3.9.3 Measurement of fuel flow

In order to measure the engine's fuel consumption under different load conditions, employed a differential pressure transmitter produced by Yokogawa in Japan, applying the principle of hydrostatic head. This transmitter was installed at the end of a burette, as Plate 3.14 illustrates, and it generated an output voltage that was exactly proportionate to the weight of the fuel inside the column. By keeping an eye on changes in the transmitter output at predetermined intervals and multiplying the result by the fuel factor, the fuel flow rate was calculated. The fuel factor is computed using equation 3.9, with ‘ρ’ representing density, ‘g’ standing for gravity, ‘h’ indicating the height of the fuel column, and ‘A’ representing the cross-sectional area.

$$\text{Fuel Factor} = \rho \times g \times h \times A \tag{3.9}$$

The sensor transmitted real-time data to the data acquisition system, and this data was cross-verified by simultaneously manually recording readings from the burette. By analyzing the fuel flow rate, the engine's brake-specific energy consumption (BSEC) at a given load as well as the brake thermal efficiency was calculated. The effectiveness of transferring fuel heat into usable work output at the brakes is known as brake thermal efficiency. Equation 3.10 contains the formula for determining brake thermal efficiency. Similar to this, BSEC uses the formulae in equation 3.11 to quantify the amount of energy in the fuel used for a specific output.

$$BTE(\%) = \frac{\mathrm{BP(kW)} * 100}{\mathrm{Mf}\left(\frac{\mathrm{kg}}{\mathrm{s}}\right) * \mathrm{CV}\left(\frac{\mathrm{kJ}}{\mathrm{kg}}\right)} \tag{3.10}$$

$$\mathrm{BSEC}\left(\frac{\mathrm{MJ}}{\mathrm{kWh}}\right) = \frac{\left(\mathrm{Mf}\left(\frac{\mathrm{kg}}{\mathrm{s}}\right) * \mathrm{CV}\left(\frac{\mathrm{kJ}}{\mathrm{kg}}\right)\right) * 3600}{\mathrm{BP(kW)}} \tag{3.11}$$

Plate 3.14 Fuel flow sensor illustration

3.9.4 Measurement of air flow

The pulsing nature of the engine's air demand makes it difficult to measure the exact amount of air flow in the engine. Since, air is compressible and is sucked in once every two crankshaft revolutions, this pulse happens. As a result, it is impractical to rely only on an induction pipe orifice since this would not yield accurate data. In order to solve this problem, an air box that was the right size and had a circular opening with a sharp edge for measuring air flow was used. A U-tube manometer was used to measure the pressure differential between the air box and the atmosphere. Equation 3.12 was used to compute the mass flow rate of air.

The formula used for calculating the mass flow rate of air is as follows:

$$\text{Mass Flow Rate} = C_d \times A \times \sqrt{(2 \times g \times h_w \times (\rho_w / \rho_a))} \tag{3.12}$$

In the equation, C_d represents the coefficient of discharge (with a value of 0.6). The orifice area is represented by 'A', the water column height is represented by 'h_w', the acceleration caused by gravity is shown by 'g', and the ratio of water density to air density is indicated by ρ_w/ρ_a.

3.9.5 Temperature Measurement

In this experiment, a six-channel digital panel meter was linked to K-type thermocouples composed of chrome-alumel. The engine's intake air, exhaust gas, coolant water (both inlet and outlet), and calorimeter water (both inlet and outlet temperatures) are all measured in large part thanks to these thermocouples. To ensure accuracy, a milli volt source covering a temperature range of up to 800°C was used in a laborious calibration process.

3.9.6 Measurement of in-cylinder pressure

The in-cylinder pressure in this investigation was measured using a "Kubeler" piezoelectric transducer. The signals that the transducer produced are sent to a charge amplifier. This amplifier not only amplified the signals but also mitigated signal noise.

Subsequently, these processed signals forwarded to the data acquisition system. Alongside signals obtained from the crank angle encoder, the data acquisition system facilitated the creation of pressure versus crank angle graphs through specialized software installed on a personal computer. For each degree of crank angle rotation, pressure data was collected. The current study generated the in-cylinder pressure against crank angle data by averaging fifty consecutive cycles, which improved precision. A picture of the pressure sensor located on the engine head may be found in Plate 3.15.

Plate 3.15 Pressure sensor installed on the engine head

3.9.7 Measurement of exhaust emissions

Diesel engine exhaust emissions primarily consist of pollutants like unburnt hydrocarbons, carbon monoxide, carbon dioxide, nitrogen oxides, and smoke. In this study, the smoke emissions were indirectly assessed by quantifying the opacity of the exhaust gas. Higher smoke emissions result in greater smoke opacity, and vice versa. To achieve this, an AVL smoke meter (as seen in Plate 3.16) was utilized. The instrument projects light beams from sources, which is then received by a sensor. Smoke particles in the exhaust gas scatter or absorb some of the light as it travels across the light beam. The sensor, in this case, is a photocell that generates a photoelectric current when exposed to light, providing a measure of the smoke opacity. In addition, the emissions of nitrogen oxides, carbon monoxide, and unburned hydrocarbons were measured using an AVL di-gas analyzer (displayed in Plate 3.17). The specifications are provided in Appendix III and Appendix IV for thorough details regarding the smoke meter's and the di-gas analyzer, respectively. The emission data for carbon monoxide, unburnt hydrocarbons, and nitrogen oxides are subsequently converted to brake-specific emission values during offline data analysis.

Plate 3.16 Smoke meter   Plate 3.17 Five gas analyzer

3.9.8 Calculation of heat release rate

The precise assessment of heat release during the combustion process in a compression ignition engine is critical, as it has a direct bearing on engine efficiency, power generation, and emission levels. Researchers have put forth several models to estimate the rate of heat release. The method described by Heywood [209] was used to estimate the heat release. Based on the fundamental rule of thermodynamics, this approach uses a single-zone model to simplify the combustion process inside the cylinder by treating it as a single unit without distinguishing between portions that have burnt and those that have not. Although there are more complex multi-zone models available, heat transfer estimates for them require tracking both burned and unburned zones. Nevertheless, it has been noted that applying a multi-zone model doesn't produce many advantages, mostly because describing the evolution of these zones is imprecise [211]. This approach is used to calculate the heat emission rate.

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} \times P \times \frac{dV}{d\theta} + \frac{1}{\gamma - 1} \times V \times \frac{dP}{d\theta} - \frac{dQw}{d\theta} \tag{3.13}$$

Where:

dQ/dθ represents the rate of net heat release inside the engine cylinder (measured in J/°CA).

dQw/dθ signifies the rate of heat transfer from the cylinder walls (measured in J/°CA).

γ stands for the ratio of specific heats.

P represents the cylinder pressure (in bar).

V indicates the gas volume (measured in m^3).

θ corresponds to the crank angle (in degrees).

This model has found extensive use among a multitude of researchers [51,167]. Additionally, it was used to determine the ignition delay. This was determined by taking the time, measured in crank angle degrees, between the start of fuel injection to the start of a sudden increase in heat release. The duration of combustion is the amount of time that passes between the beginning and the conclusion of heat release.

3.9.9 Engine trial procedure

The engine testing procedure began with starting the engine under a no-load condition, using pure diesel as the initial fuel. After achieving a steady-state operating condition, the fuel supply was transitioned to the test fuels. To ensure reliable results, the engine was allowed to stabilize for approximately thirty minutes with these test fuels. All measurements were taken at each load only when the engine had reached a stable state. Regular calibrations of all instruments were diligently conducted throughout the experimentation. It is noteworthy that during the testing, the injector opening pressure was constant at its rated value.

No load, 20% of maximum load, 40% of maximum load, 60% of maximum load, 80%

of maximum load, and full load are the conditions at which the engine was operated. For each of these load levels, the exhaust gas temperature, air flow rate, fuel flow rate, CO, HC, NOx, and smoke opacity emissions were meticulously recorded. Additionally, a personal computer was used and the data collection device to capture pressure-crank angle data for 50 consecutive cycles at each load level. This data was then processed to get the average variation in the pressure-crank angle.

3.10 Experiments

- The tests encompassed a range of fuel variations, which included pure diesel as well as a variety of blends combining pine oil and eucalyptus oil in different proportions.
- The study included a thorough assessment of the various fuel mixes' and their corresponding base fuels' engine performance and emission characteristics.
- After determining which diesel blend of eucalyptus and pine oils had the highest engine performance in terms of emissions and brake thermal efficiency, conducted additional experiments. These experiments involved the introduction of different volume percentages of pine oil and eucalyptus oil biofuels with the aim of improving engine performance and reducing emissions.

3.11 Uncertainty and Accuracy Measurement Instruments

The accuracy and range of the instruments used in this investigation is given in Table 3.3. Notably, each device demonstrated exceptionally high precision.

Table 3.3 Accuracy of instruments

Measurement	**Measuring Principle**	**Accuracy**	**Range**
Pressure	Piezoelectric	±1bar	0-200bar
Crank angle encoder	Optical	±0.2°CA	0-720°CA
Exhaust Temperature	K-type thermocouple	±1°C	0-1000°C
Time	Stop-watch	±0.5%	-
Engine Speed	Magnetic pick-up type	±20rpm	0-2000rpm
Engine Load	Strain gauge type load cell	±0.1kg	0-25kg

In every measurement process, deviations from the true values are inevitable, even when experiments are meticulously executed. These errors can be classified into two categories: systematic and random. Systematic errors can be addressed through the application of appropriate correction factors, while random errors are unpredictable and can only be statistically estimated. It is necessary to perform many measurements of the same quantity under the same conditions and with consistent attention in order to detect random mistakes. The Gaussian distribution approach may be used to determine the uncertainty surrounding these measurements, and 97.728% of the measured data falls within these bounds at a confidence level of $\pm 2\tau$. Equation 3.14 provides a means to calculate uncertainty, expressing it as a percentage of the measured parameter itself.

The uncertainty of any measured parameter $(\Delta Z) = (2\sigma i / \bar{Y}) \times 100$ (3.14)

In an attempt to compute uncertainty, studied the observed parameter's mean ($\bar{Y}$) and standard deviation (τ_i). The uncertainty computation technique used complies with the recommendations made by Kline and McClintock.

Let S be a computed number, as defined by Equation 3.15, based on 'n' independently measured parameters, which are represented as Z_1, Z_2, Z_3, Z_4... Z_n.

$$Z_1, Z_2, Z_3, Z_4, \ldots Z_n = S \quad (3.15)$$

For every measured parameter, the following are the uncertainty limits: $Z_1 \pm \Delta Z_1$, $Z_2 \pm \Delta Z_2$, $Z_3 \pm \Delta Z_3$, $Z_4 \pm \Delta Z_4$, ... $Z_n \pm \Delta Z_n$.

For the computed parameter, the uncertainty limit is defined as $S \pm \Delta S$. Its use the root sum-squared approach to find reasonable error bounds for a computed quantity that is generated from several measured values. Equation 3.16 expresses the error's magnitude.

$$\Delta S \text{ is equal to } \sqrt{((\partial S/\partial Z_1 \times \Delta Z_1)^2 + (\partial S/\partial Z_2 \times \Delta Z_2)^2 + (\partial S/\partial Z_n \times \Delta Z_n)^2)} \quad (3.16)$$

The experiments' cumulative uncertainty was determined to be ±2.272%.

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Solution and approach

Building upon an extensive review of the literature regarding the potential utilization of less viscous fuels in diesel engines, the present study focuses on the application of pine oil and eucalyptus oil in three distinct modes: as a blend, a distilled blend, and in dual fuel mode with methanol fumigation.

In this chapter, the investigation primarily centres on the blend fuel mode, aimed at understanding the behavioural characteristics of pine oil and eucalyptus oil in a diesel engine. The objective is to establish a foundational understanding of how these fuels impact engine performance, emissions, and combustion. The pine oil and eucalyptus oil used in this study are commercially sourced and employed in their original state. Various blends of these oils with diesel, such as 5% pine oil and 5% eucalyptus oil, 10% pine oil and 10% eucalyptus oil, 15% pine oil and 15% eucalyptus oil, and 50% pine oil and 50% eucalyptus oil, are meticulously prepared using an ultrasonic agitator to ensure blend uniformity. Subsequently, these prepared blends are subjected to experimental analysis in a single-cylinder diesel engine, both with and without engine modifications. Key performance, combustion, and emission parameters, including Brake Specific Energy Consumption (BSEC), Brake Thermal Efficiency (BTE), Exhaust Gas Temperature (EGT), in-cylinder pressure, heat release rate, Carbon Monoxide (CO), Hydrocarbons (HC), smoke opacity, and Nitrogen Oxides (NOx), are rigorously evaluated under consistent engine speed conditions for varying engine power outputs. Before commencing experimentation with the pine oil and eucalyptus oil blends, the engine was run on diesel for 30 minutes to reach a steady state and establish normal operating temperatures. After this initial run, the base diesel fuel was thoroughly purged from the tank, fuel pump, and lines, and replaced with the test fuels. Additionally, the lubrication oil and cooling air temperatures are recorded to confirm that the engine had reached an adequate warm-up state.

The engine was incrementally loaded from 20% to 100% in 20% increments, achieved by adjusting the current supplied to the eddy current dynamometer. As the engine operated at a constant speed of 1500 rpm, the fuel pump was fine-tuned to maintain this consistent speed across varying load conditions. All data related to the engine's performance and characteristics are documented under ambient conditions, with readings taken when the engine had stabilized and achieved a steady state. The experiments were conducted in triplicate, and the resulting averages were employed for calculations to enhance the precision and reliability of the obtained results.

4.2 GC-MS analysis

The GC-MS profiles for pine oil and eucalyptus oil are visually presented in Figs. 4.1-4.2. Notably, regardless of the specific oil blend, key compounds such as Hept-3-ene (Carene), 1,4-di-methyl-5-(1-methyl-ethyl)-cyclo-pentene, cyclo-hexene, 3,7-di-methyl-octa-1, 6-di-ene, and 2-oxa-bi-cyclo [2.2.2] octane are consistently found in substantial concentrations. An intriguing observation is the absence of certain compounds in one oil type while present in the other. For instance, 3,7,7-tri-methyl-bi-cyclo [4,1,0] and bi-cyclo [2.2.1] heptane, present in eucalyptus oil, are absent in pine oil, while alpha-terpineol-p-menth-1-en-8-ol, found in pine oil, was missing in eucalyptus oil. The major constituents present in pine oil and eucalyptus oil is tabulated in Table 4.1.

4.2.1 Major constituents present in pine oil

In Fig. 4.1, the analysis of a pine oil biofuels sample was conducted using the GC-MS method. This analysis aimed to identify and assess the major constituents of pine oil, focusing on five key components. The concentrations and peak area percentages of these constituents are determined as follows: carene at 46.53%, cyclopentene at 0.06%, limonene at 9.62%, bi-cyclo at 0%, citro-nellene at 0%, alpha-terpineol at 14.79%, alpha pinene at 0%, and 2-oxy-bicyclo [2.2.2] at 0.83%. Remarkably, bi-cyclo, citro-nellene, and alpha pinene are found to be absent (zero concentration) in the test sample. The analysis involved mass spectra within the m/z ratio range of 40 to 650, and the peaks are identified by comparing them with known library constituents. The majority of these constituents adhered to the molecular formula $C_{10}H_{18}$, with the exception of 3-carene, which had a molecular formula of $C_{10}H_{16}$, and alpha-terpineol and 2-oxy-bi-cyclo, which followed the $C_{10}H_{18}O$ molecular formula.

Among these constituents, 3-carene exhibited the highest concentration, with a prominent peak at a retention time of 11.030, as measured using the FID technique. This concentration was over three times higher than that of alpha-terpineol, which had the second-highest peak with a concentration at a retention time of 19.372. Subsequently, limonene, 2-oxy-bi-cyclo, and cyclo-pentene are observed in decreasing order of concentration, with retention times of 11.696, 11.720, and 9.092, respectively. Cyclopentene had the lowest concentration, with a retention time of 9.092, followed by 2-oxy-bi-cyclo, which represented the second-lowest concentration in the pine oil sample.

The primary aim of this study was to enhance the combustion rate of the biofuel blend sample and its constituents. Notably, alpha-terpineol and 2-Oxybicyclo constituted 10.40% within the samples. These constituents are found to notably enhance combustion quality in comparison to other constituents. While many researchers typically isolate such compositions for specific applications, the central focus of this study is not on the separation of these constituents but rather on their collective utilization. Additional insights into this topic will be provided in the subsequent quantitative analysis. The expectation is that the identified constituents can serve as the main fuel source, thereby increasing calorific value and ensuring economic viability. The chemical structure and composition of the blend reveal the presence of several constituents with a high ratio of carbon and hydrogen, rendering them suitable for use in the blend without isolation. This characteristic enables them to serve as performance enhancers, improving atomization through reduced viscosity and increased calorific value.

4.2.2 Major constituents present in eucalyptus oil

The calculated analysis of all constituents present in the eucalyptus oil is depicted in Fig. 4.2. In this investigation, the sample of eucalyptus oil well analysis using the GCMS-QP2010 method, leading to the identification of six major constituents based on their concentrations and peak area percentages. These constituents are as follows: carene at 15.69%, cyclo-pentene at 25.28%, limonene at 6.68%, bi-cyclo at 3.68%, citro-nellene at 2.24%, alpha-terpineol at 0%, pinene at 0%, and 2-oxy-bi-cyclo [2.2.2] at 1.37%. The analysis adhered to mass spectra within the m/z ratio range of 40 to 650, and the peaks are identified through comparison with library constituents. Most of these constituents conformed to the molecular formula $C_{10}H_{18}$, with the exceptions being 3-carene, which had a molecular formula of $C_{10}H_{16}$, and 2-oxybicyclo, which followed the $C_{10}H_{18}O$ molecular formula. Among these constituents, 3-Carene was found to have the second-highest concentration and the second-highest peak, with a retention time of 10.850 as determined using the flame ionization detector (FID) technique. Notably, this concentration was nearly half that of cyclo-pentene, which had the highest peak and concentration, with a retention time of 9.415. Subsequently, limonene, bi-cyclo, citro-nellene, and 2-oxy-bi-cyclo are observed in decreasing order of concentration, with retention times of 11.666, 9.560, 8.058, and 11.720, respectively. Of significance, 2-oxy-bi-cyclo exhibited the lowest concentration at a retention time of 11.720, followed by citro-nellene, which represented the second-lowest concentration in the eucalyptus oil blend sample. The primary aim of this study was to enhance the combustion rate of the biofuel blend sample and its constituents. Notably, 2-Oxybicyclo constituents contributed 10.40%.

In the case of eucalyptus biofuels, it was observed that 2-oxy-bi-cyclo contributed 10.40% additional air in its composition, thereby providing significant support for combustion processes.

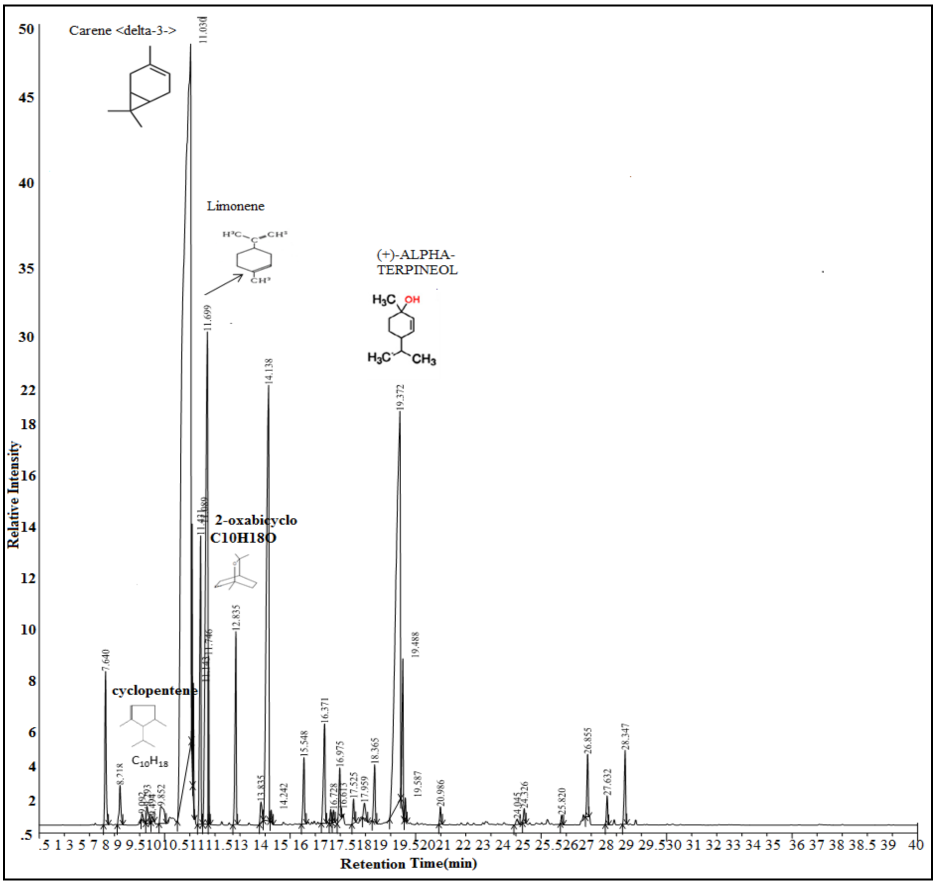


Fig. 4.1 Relative Intensity (area %) with R.T (minutes) of pine oil

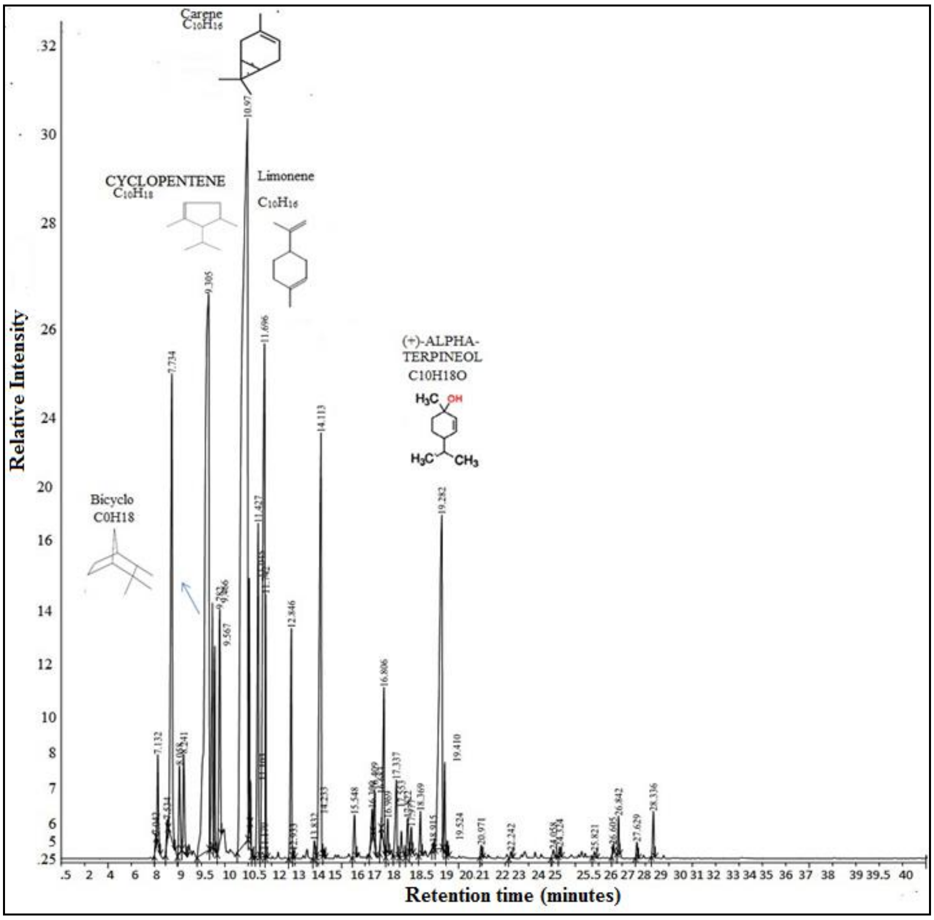


Fig. 4.2 Relative Intensity (area %) with R.T (minutes) of eucalyptus oil

Table 4.1: Major constituents present in pine and eucalyptus oil

Pine oil		Eucalyptus oil		NAME/IUPAC NAME
RT	Concentration (Area %)	RT	Concentration (Area %.)	
-	-	8.087	2.24%	3,7-dimethylocta-1,6-diene (Citronellene)
9.090	0.06%	9.415	25.28%	1,4-dimethyl-5-(1-methylethyl)-cyclopentene
-	-	9.560	3.80%	Bicyclo [2.2.1] heptane,
11.030	46.53%	10.850	15.69%	3,7,7-trimethyl bicyclo [4,1,0] hept-3-ene (Carene)
11.699	9.62%	11.666	6.68%	Cyclohexene (Limonene)
11.746	0.83%	11.720	1.37%	2-oxabicyclo [2.2.2] octane
19.372	14.79%	-	-	(+)-alpha-terpineol p-menth-1-en-8-ol

4.3 Physicochemical properties

Various properties such as density, calorific value, surface tension, flash point and cetane using the instruments described in the previous chapter and the corresponding ASTM standards were measured for the test samples. This section compares the different properties of the test samples. Tables 4.2 and 4.3 shows the different properties of the test samples.

Table 4.2 Biofuels blend sample

Blend Name	Density (kg/m^3)	Kinematic viscosity (m^2/s)	Calorific value (MJ/Kg)	Surface tension (mN/m)	Flash point (^0C)	Cetane index
Diesel	830	$3.5 * 10^{-6}$	43.120	28.39	71	52
Neat pine	874	$1.3 * 10^{-6}$	42.800	27	52	11
Neat Eucalyptus	891	$2.1 * 10^{-6}$	43.270	29.02	54	53
P5E5D90	835	$2.83 * 10^{-6}$	42.735	27.60	69	47
P10E10D80	842	$2.77 * 10^{-6}$	42.793	28.30	67	49
P15E15D80	853	$3.30 * 10^{-6}$	42.505	27.20	61	46
P50E50	882	$1.95 * 10^{-6}$	42.774	26.60	53	31
E20D80	842	$3.22 * 10^{-6}$	43.862	28.90	64	50
P20D80	871	$2.80 * 10^{-6}$	41.321	26.34	56	43
Methanol	780	$0.59 * 10^{-6}$	19.700	-	11	5

Table 4.3: Distilled biofuels blend sample

Blend Name	Density (kg/m^3)	Kinematic viscosity (m^2/s)	Calorific value (MJ/Kg)	Surface tension (mN/m)	Flash point (^0C)	Cetane index
PD10ED10D80	825	$2.65 * 10^{-6}$	43.011	28.30	63	49.4
ED20D80	835	$2.60 * 10^{-6}$	43.320	28.90	64	54
PD20D80	831	$2.67 * 10^{-6}$	43.010	26.30	56	43.3

PD-Distilled Pine oil, ED- Distilled Eucalyptus oil, PDED- Distilled Pine and Eucalyptus oil

4.3.1 Density

Density stands as a crucial factor when assessing the appropriateness of a fuel for use in a compression ignition engine. Critical fuel properties, such as cetane rating and heating value, are closely tied to fuel density. Moreover, fuel density significantly influences the engine's power output [212]. Recognized standards like ASTM D1298 and EN 590 for diesel fuel specify acceptable density ranges of 820-891 kg/m³ and 823-865 kg/m³, respectively. As previously mentioned, the density of the test samples was measured using an Anton Parr oscillating U-tube density meter.

Fig. 4.3 visually illustrates the density variations among various blends, encompassing pine oil, eucalyptus oil, and their combinations with diesel. Notably, pine oil and eucalyptus oil exhibit the highest density, while pure diesel registers the lowest density. As the proportion of diesel in the pine oil and eucalyptus oil blend increases, there is a gradual decrease in density. The recorded density values are as follows: 830 kg/m³ for D100, 874 kg/m³ for P100, 891 kg/m³ for E100, 835 kg/m³ for P5E5D90, 842 kg/m³ for P10E10D80, 853 kg/m³ for P15E15D70, 882 kg/m³ for P50E50, 846 kg/m³ for E20D80, and 840 kg/m³ for P20D80.

In Fig. 4.4, the density of distilled pine oil, eucalyptus oil, and their respective blends with diesel are illustrated. Similar to the earlier findings, pine oil and eucalyptus oil exhibit the highest density, while the blends containing biofuels display lower density values. As the proportion of biofuels in the diesel blend increases, the density decreases. The density values for distilled PD10ED10D80, ED20D80, and PD20D80 are 825 kg/m³, 835 kg/m³, and 831 kg/m³, respectively.

4.3.2 Kinematic Viscosity

Kinematic viscosity plays a significant role in determining how easily a fluid flows, affecting both the functionality of the fuel injection system and the quality of spray atomization. This influence becomes more prominent in colder conditions when fluid viscosity increases, consequently impacting fuel combustion. Moreover, inadequate fuel atomization can result in heightened soot formation and engine deposits [213]. As per the specifications outlined by ASTM D445 and EN 590 for the use of diesel-like fuels in compression ignition engines, the kinematic viscosity should fall within the range of 1.3 to 3.5 m²/s.

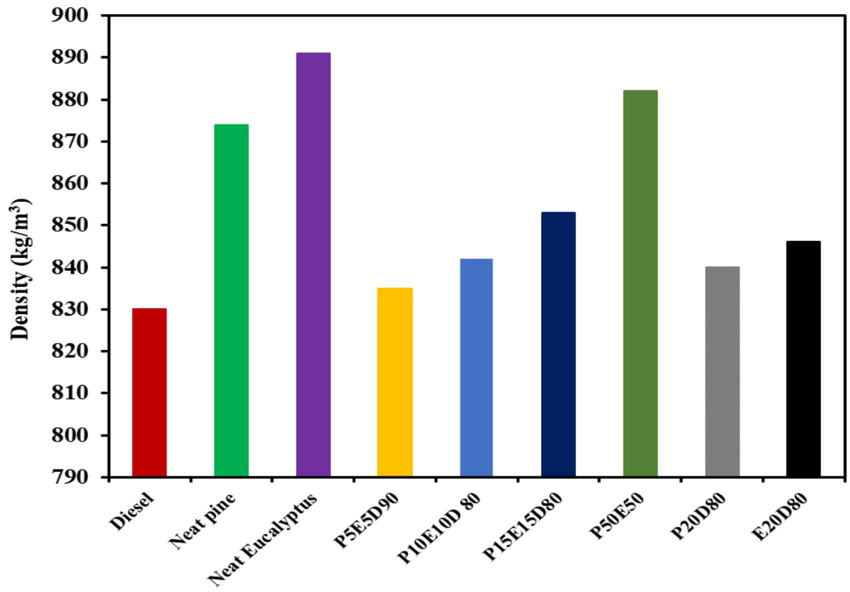


Fig. 4.3 Variation in density for blends of pine oil, eucalyptus oil and diesel

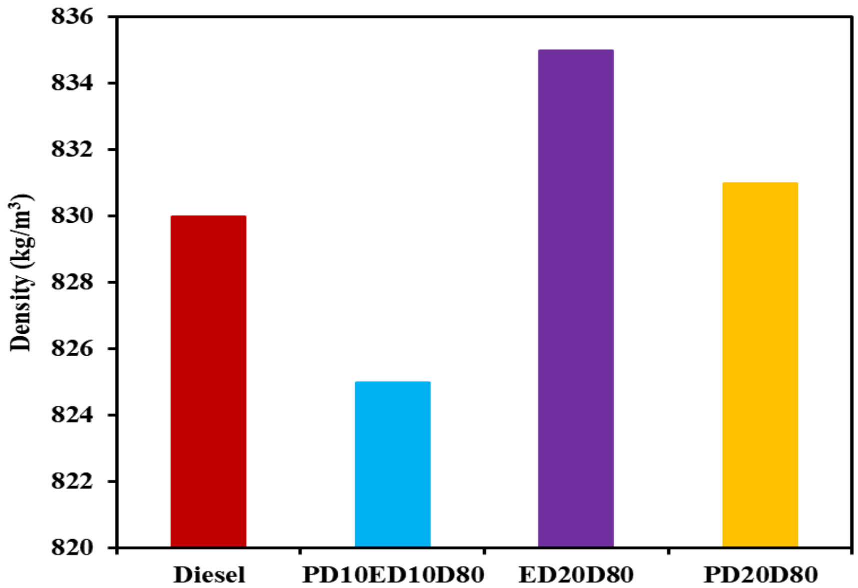


Fig. 4.4 Variation in density for blends of distilled pine oil, eucalyptus oil and diesel

Figs. 4.5 and 4.6 depict the kinematic viscosity of pine oil and eucalyptus oil, as well as a distilled blend of pine oil and eucalyptus oil fuel samples. The observed kinematic viscosity values for D100, P100, E100, P5E5D90, P10E10D80, P15E15D70, P50E50, E20D80, P20D80, PD10ED10D80, ED20D80, and PD20D80 are 3.5, 1.3, 2.1, 2.83, 2.77, 3.30, 1.95, 2.80, 3.22, 2.65, 2.67, and 2.60 m²/s, respectively.

It's noteworthy that the kinematic viscosity of pure pine oil and eucalyptus oil is lower compared to pure diesel, primarily due to the presence of oxygen in the fuel. In a study conducted by [214], it was observed that the presence of oxygen in the fuel has a minor influence on its kinematic viscosity, meaning that higher oxygen content is associated with higher viscosity.

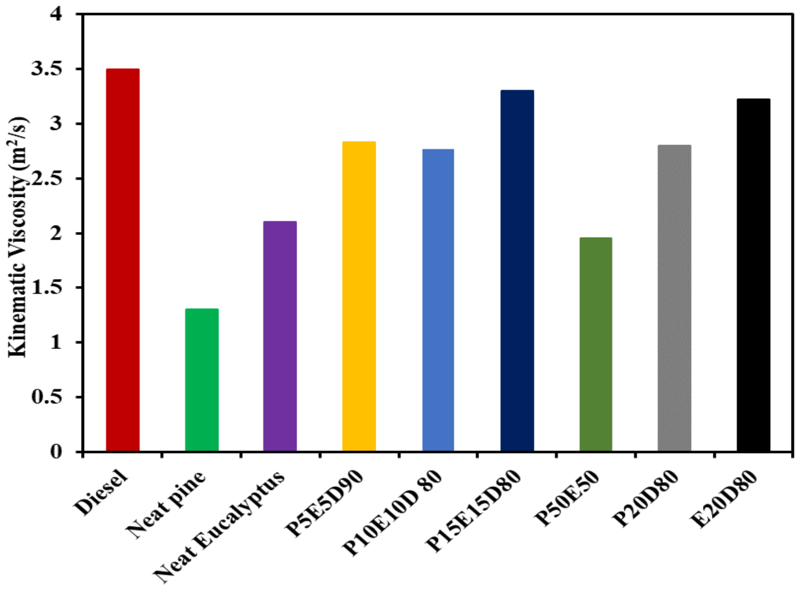


Fig. 4.5 Variation in kinematic viscosity for blends of pine oil, eucalyptus oil and diesel

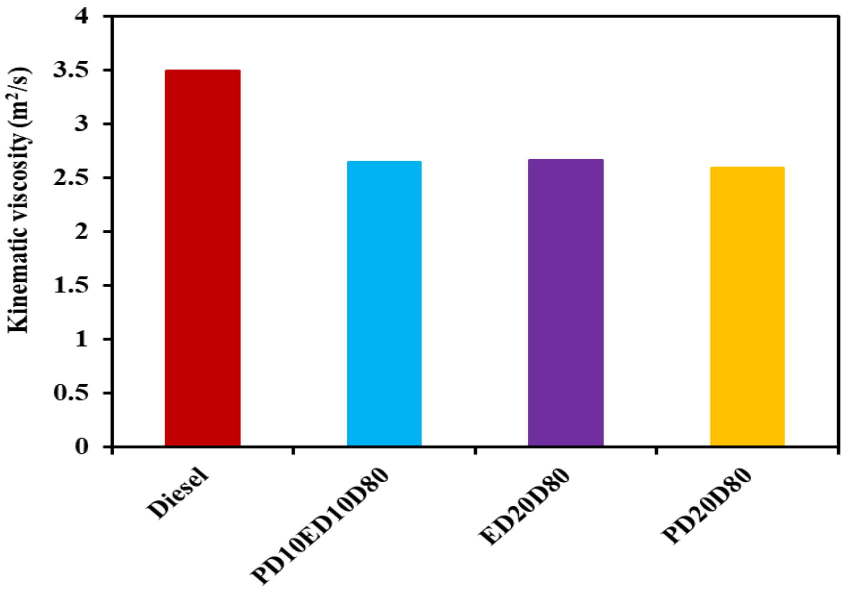


Fig. 4.6 Variation in kinematic viscosity for blends of distilled pine oil, eucalyptus oil and diesel

4.3.3 Calorific Value

Calorific value represents the energy released when a specific mass of fuel undergoes combustion, generating water and CO_2 as by-products. This property has a direct and crucial influence on an engine's power output. As previously mentioned, a bomb calorimeter was used to measure the energy content of the test samples. Fig. 4.7 illustrates the calorific values, wherein diesel exhibits the highest value at 43.12 MJ/kg, which is slightly lower than the calorific value of neat pine oil, standing at 43.27 MJ/kg, but surpasses the calorific value of eucalyptus oil at 43.27 MJ/kg. Neat pine oil records the lowest calorific value at 42.8 MJ/kg.

Fig. 4.8 illustrates the calorific values of blends comprising pine oil and eucalyptus oil, as well as distilled pine oil and eucalyptus oil biofuels when mixed with diesel. The oxygen content of a fuel has a limited effect on its calorific value. This is evident in the comparison of pure pine oil, which contains some oxygen, displaying a lower calorific value than diesel. Conversely, eucalyptus oil, with its higher oxygen content, exhibits a higher calorific value than diesel. In terms of the blends, an increase in the proportion of pine oil and eucalyptus oil within the mixture corresponds to a rise in the calorific value. Notably, for PD10ED10D80, ED20D80, and PD20D80, the calorific values are 43.011 MJ/kg, 43.320 MJ/kg, and 43.01 MJ/kg, respectively.

In the case of pine oil and eucalyptus oil blends, an increase in the percentage of biofuels in the mixture results in a decrease in calorific value. Specifically, the calorific values for D100, P100, E100, P5E5D90, P10E10D80, P15E15D70, P50E50, E20D80, and P20D80 are 43.120 MJ/kg, 42.800 MJ/kg, 43.270 MJ/kg, 42.735 MJ/kg, 42.793 MJ/kg, 42.505 MJ/kg, 43.035 MJ/kg, 43.140 MJ/kg, and 42.072 MJ/kg, respectively.

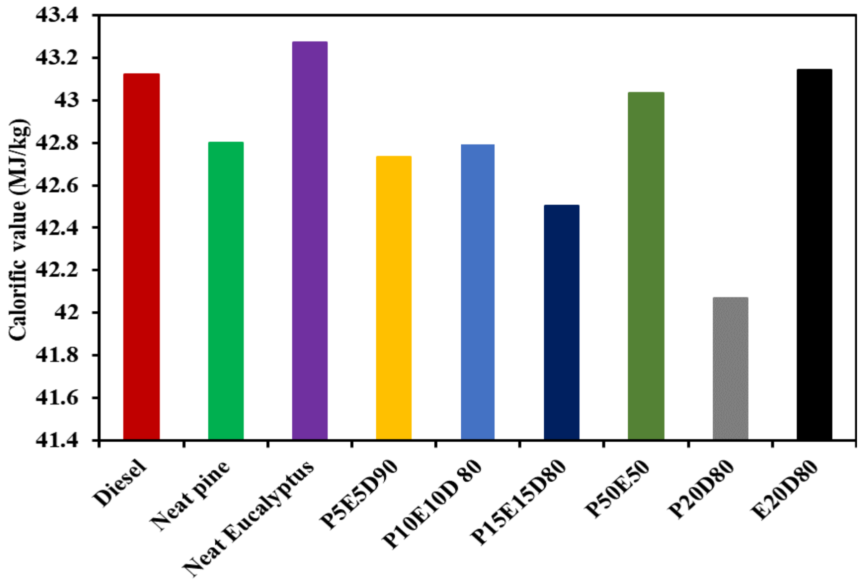


Fig. 4.7 Variation in calorific value for blends of pine oil, eucalyptus oil and diesel

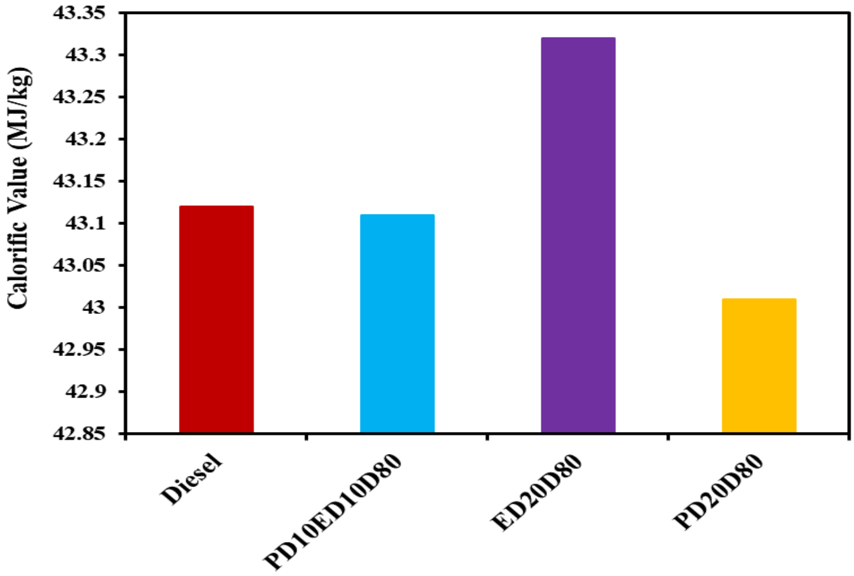


Fig. 4.8 Variation in calorific value for blends of distilled pine oil, eucalyptus oil and diesel

4.3.4 Surface tension

Surface tension plays a pivotal role in comprehending the suitability of a fuel for use in a compression ignition engine. It is closely interconnected with essential fuel properties such as kinematic viscosity, calorific value, and cetane rating, all of which are critical for ensuring proper atomization and combustion in the engine. Notably, the power output of the engine remains unaffected by the surface tension of the fuel [215]. According to the Tensiometer-3 standard specified by LAUDA for diesel fuel; the acceptable range for surface tension is 26.0mN/m to 29.5mN/m. Fig. 4.9 illustrates the surface tension of various blends, including pine oil, Eucalyptus oil, and a blend of distilled pine oil and Eucalyptus oil mixed with diesel. Notably, diesel and Eucalyptus oil exhibit the highest surface tension, while neat pine oil records the lowest surface tension. As the percentage of diesel in the pine oil and Eucalyptus oil blend increases, the surface tension decreases. The observed surface tension values are as follows: 28.39mN/m for D100, 27mN/m for P100, 29.02mN/m for E100, 27.60mN/m for P5E5D90, 28.30mN/m for P10E10D80, 27.20mN/m for P15E15D70, 26.60mN/m for P50E50, 28.90mN/m for E20D80, and 26.34mN/m for P20D80. In Fig. 4.10, the surface tension of pine oil, Eucalyptus oil, and their blends mixed with diesel is depicted. The surface tension of pine oil and Eucalyptus oil is the highest, while the blends containing biofuels display lower surface tension values. Notably, as the ratio of biofuels in the diesel blend boosts, the surface tension diminishes. Specifically, for distilled PD10ED10D80, ED20D80, and PD20D80, the surface tension values are 28.3mN/m, 28.90mN/m, and 26.30mN/m, respectively.

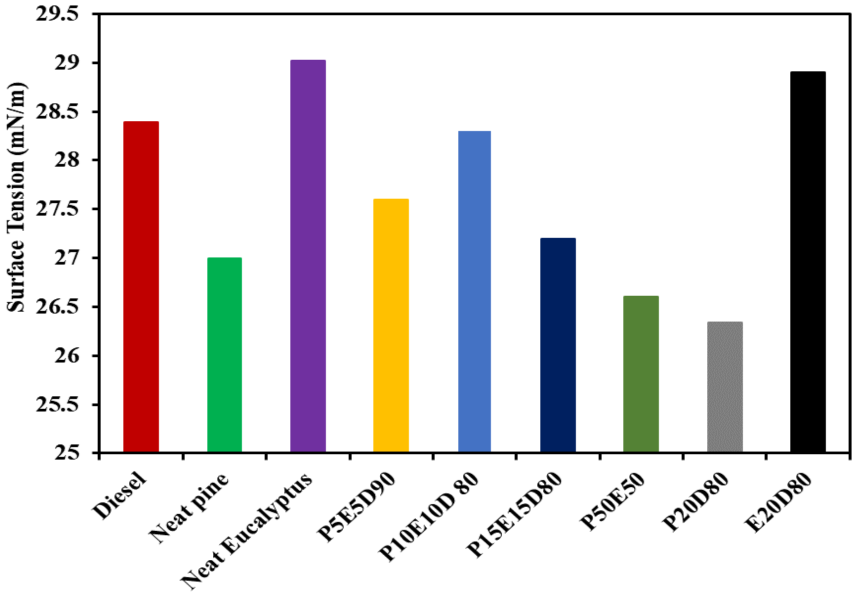


Fig. 4.9 Variation in surface tension for blends of pine oil, eucalyptus oil and diesel

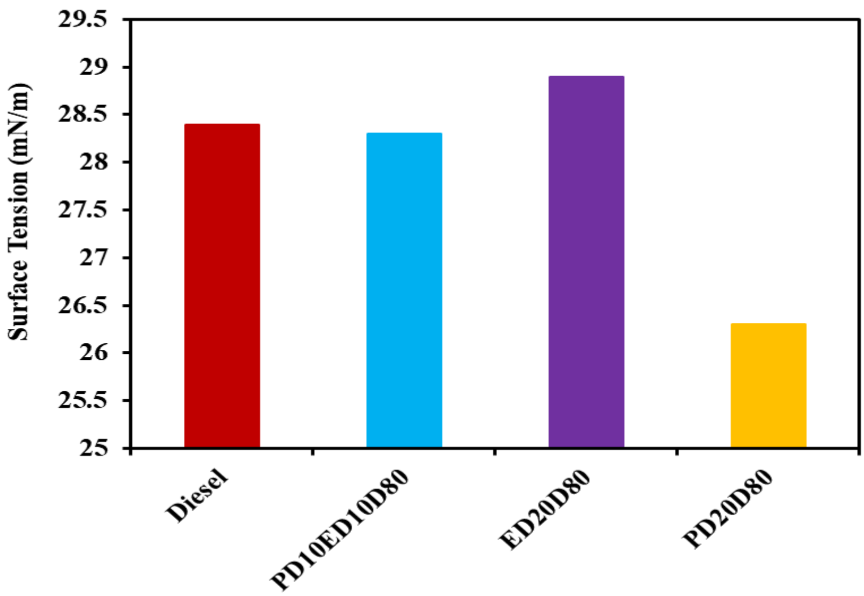


Fig. 4.10 Variation in surface tension for blends of distilled pine oil, eucalyptus oil and diesel

4.3.5. Cetane Index

The cetane number serves as a critical parameter for understanding a fuel's ability to self-ignite once it is injected into the combustion chamber. Due to the challenges associated with cetane number estimation, the cetane index comes to the forefront as an alternative indicator, aiding in grasping a fuel's ignition characteristics. The cetane index is determined through an assessment of fuel distillation and density, as elaborated in detail in the previous chapter. Figs. 4.11 and 4.12 present the cetane numbers of the test fuels. It's evident that eucalyptus oil exhibits a higher cetane index compared to pine oil and diesel. When the percentage of pine oil and Eucalyptus oil blended with diesel increases, the cetane index decreases. Fig. 4.12 highlights the ignition probability of eucalyptus oil, which surpasses that of pine oil and diesel across a range of temperatures, reaching 100% probability at lower temperatures than other test samples. This underscores the significance of the higher cetane index in pine oil and eucalyptus oil blends, facilitating earlier ignition compared to other biofuels blends. Diesel, on the other hand, boasts a higher cetane index than pine oil but a lower cetane index than eucalyptus oil. The distilled blends of pine oil and eucalyptus oil mixed with diesel demonstrate an increase in cetane index as biofuels replaces diesel. Additionally, as the proportion of biofuels in the blend rises, the cetane index decreases. Specifically, the observed cetane index values are as follows: 52 for D100, 11 for P100, 53 for E100, 47 for P5E5D90, 49 for P10E10D80, 46 for P15E15D70, 31 for P50E50, 50 for E20D80, 43 for P20D80, 49.4 for PD10ED10D80, 54 for ED20D80, and 43.3 for PD20D80.

4.3.6 Flash Point

The flash point of a fuel represents the lowest temperature at which the vaporized fuel, upon heating, becomes flammable. At this temperature, when an ignition source is brought near the fuel's surface, the vapours mix with the surrounding oxygen in the air, leading to a momentary flash of combustion. As depicted in Fig. 4.13, pine oil and eucalyptus oil exhibit flash points at 52°C and 54°C, respectively, whereas diesel displays a flash point of 71°C. The blending of pine oil and eucalyptus oil with diesel, as well as the distilled mixtures of these biofuels, results in an increase in flash point as the proportion of the biofuels blends rises. Specifically, the flash points for D100, P100, E100, P5E5D90, P10E10D80, P15E15D70, P50E50, E20D80, and P20D80 are observed to be 71°C, 52°C, 54°C, 67°C, 63°C, 61°C, 53°C, 64°C, and 58°C, respectively. In Fig. 4.14, it is evident that the distilled blend of pine oil and eucalyptus oil possesses the highest flash point at 60°C. The introduction of biofuels blends derived from pine oil and eucalyptus oil results in an elevation of the flash point, and as the percentage of biofuels in the blend increases, the flash point also increases. Specifically, the flash points for PD10ED10D80, ED20D80, and PD20D80 are 63°C, 64°C, and 56°C, respectively.

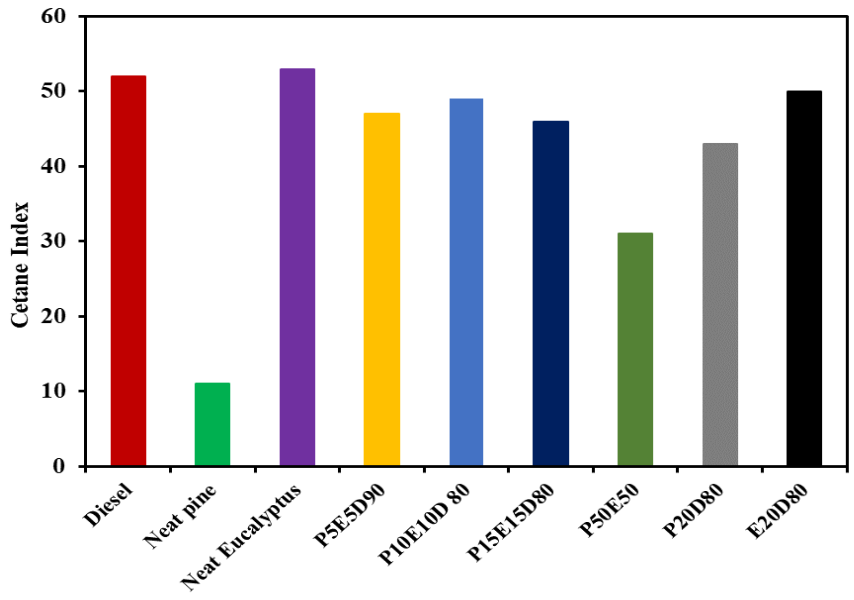


Fig. 4.11 Variation in cetane index for blends of pine oil, eucalyptus oil and diesel

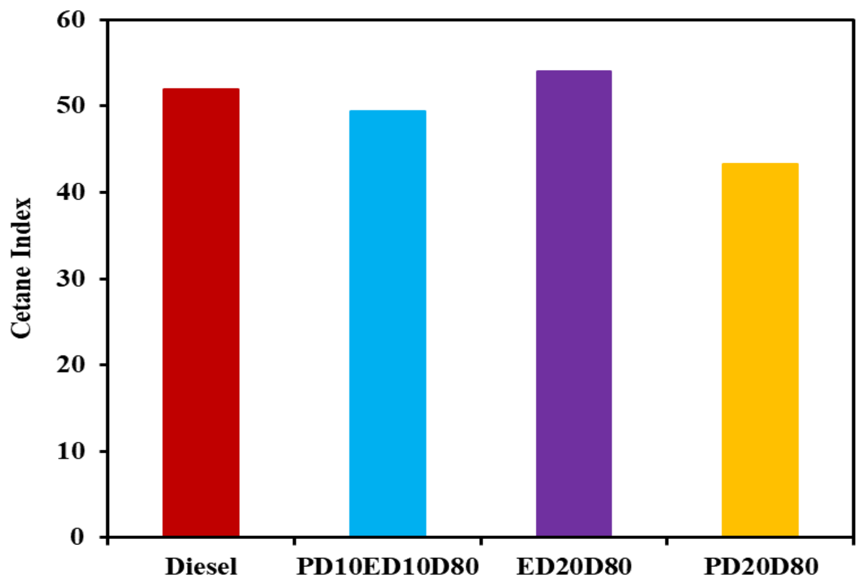


Fig. 4.12 Variation in cetane index for blends of distilled pine oil, eucalyptus oil and diesel

4.3.7 Fuel droplets size

The particle size distribution of the fuel droplets was determined using the Malvern Spraytec, employing a laser diffraction technique. Spray characterization was conducted using a solenoid-controlled, integrally built injector with a single hole, designed with dimensions identical to the original engine injector. The injector nozzle was positioned at a distance of 25 mm from the laser source, and the injection was performed at a constant pump pressure of 200 bar. Although the injection process itself lasted approximately 12 milliseconds, the study spanned duration of 200 milliseconds. For each type of test fuel, a total of five datasets recollected, and the values were subsequently calculated based on their average.

The size of fuel particles plays a critical role in combustion dynamics. Smaller-sized fuel molecules tend to vaporize more efficiently, making ignition easier. The results indicated that the Sauter mean diameter (SMD) of the oil blend was the smallest, followed by methanol. The ranking of SMD values for all the tested fuels was as follows: P10E10D80 < methanol < diesel. These findings suggest that the oil blend, due to its lower viscosity, exhibits the shortest delay period and the fastest vaporization among all the tested fuels. Additionally, fossil diesel fuel exhibited a higher SMD due to its higher viscosity. It was further observed that as the proportion of P10E10D80 in the mixture decreases, the SMD also decreases, indicating that P10E10D80 has the lowest SMD.

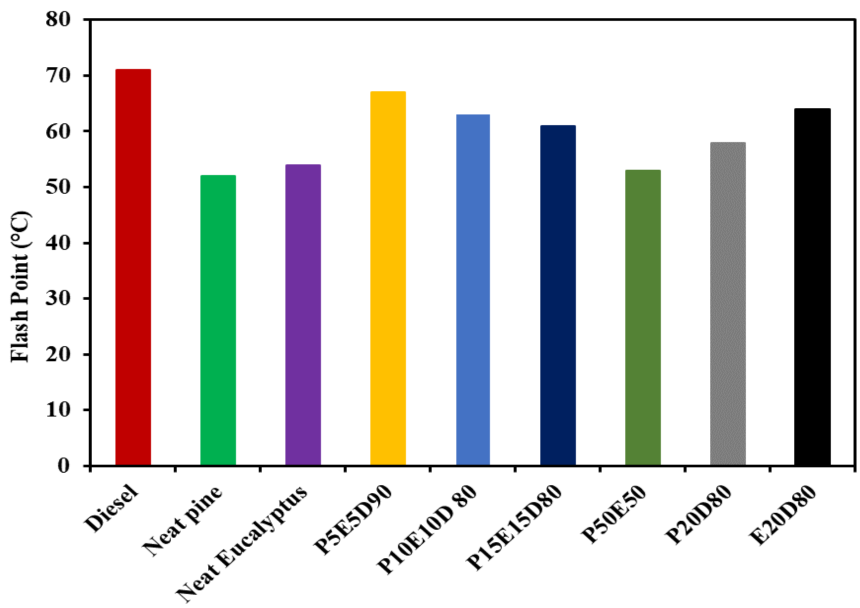


Fig. 4.13 Variation in flash point for blends of pine oil, eucalyptus oil and diesel

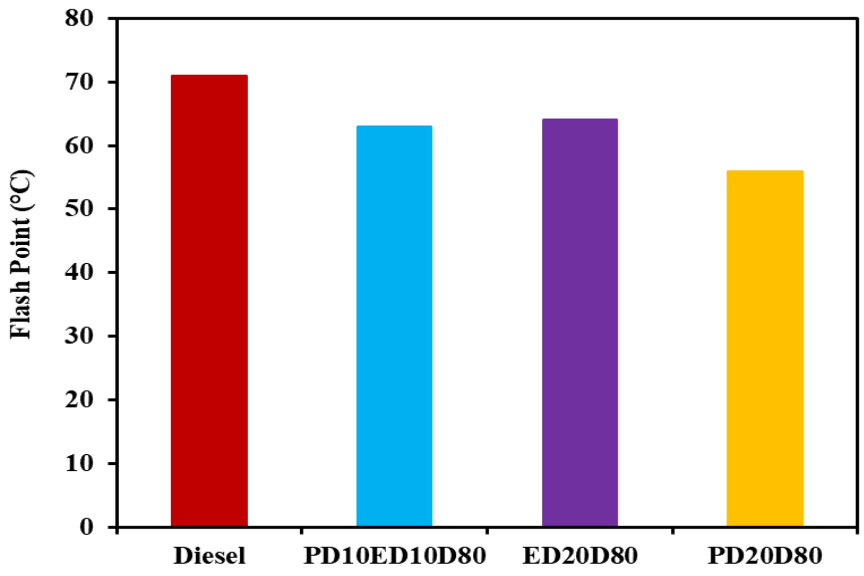


Fig. 4.14 Variation in flash point for blends of distilled pine oil, eucalyptus oil and diesel

4.4 Fuel property changes during storage

In the course of this study, monthly assessments were conducted of three vital fuel properties: density, kinematic viscosity, and calorific value, with the aim of understanding how these properties evolve over a one-year storage period. The fuel samples were meticulously stored in hermetically sealed glass containers to shield them from direct sunlight. Throughout the twelve-month storage period, the density of the test fuels exhibited minor fluctuations. Interestingly, the blends of pine oil and eucalyptus oil experienced a greater rate of density increase compared to the individual components. This change could likely be attributed to the presence of oxygen, which may have contributed to the formation of oxygenated by-products. As the blends contain a higher proportion of distilled pine oil and eucalyptus oil, the oxygen content is proportionately elevated, thus amplifying the changes in the blend's properties. When compared to freshly prepared fuel, the stored fuel's density increased by 0.15% to 0.31%, while distilled blends experiencing a decrease of 0.08% to 0.09%. It was also observed that the kinematic viscosity of the test fuels changes after being stored for twelve months. Fuel viscosity plays a critical role in the performance of the fuel injection system in a diesel engine. An increase in viscosity can compromise the injector's ability to atomize the fuel, resulting in poor engine performance marked by elevated fuel consumption and increased emissions. Additionally, high fuel viscosity can lead to clogging of the fuel filters. As the storage duration increased, there was a noticeable upward trend in viscosity. For freshly prepared pine oil and eucalyptus oil, the initial viscosity of 1.2 m^2/s and 2.1 m^2/s increased to 1.33 m^2/s and 2.5 m^2/s, respectively, over the course of one year of storage. It's noteworthy that the kinematic viscosity of the blends also well an increase, although they remained lower than that of diesel.

It also observed that the calorific value of the various fuel samples changes over a twelve-month storage period. As the storage duration increased, there was a slight decrease in calorific value across all the test samples. Notably, the distilled blend of pine oil and eucalyptus oil demonstrated a relatively higher rate of decrease in calorific value compared to the other fuel samples. For freshly prepared fuel with an initial calorific value of 43.862 MJ/kg, the value decreased to 42.102 MJ/kg after twelve months, indicating a reduction of 0.51% to 0.65%.

Collectively, these findings indicate that fuel properties are indeed affected during a twelve-month storage period, albeit not significantly. Furthermore, the values remain within the specified standards for diesel. This suggests that the stored fuel remains suitable for use in a compression ignition engine without any notable difficulties, even after a year of storage.

4.5 Engine characteristics with blends of pine oil, eucalyptus oil and diesel

Renewable diesel holds promise as a potential alternative fuel for compression ignition engines. As previously mentioned, the physical and chemical properties of this fuel closely resemble those of conventional diesel, making it compatible with existing diesel infrastructure for storage and transportation.

This section presents the experimental investigation of the engine performance of pine oil and eucalyptus oil blends. The research was conducted using a single-cylinder diesel engine commonly employed in agriculture and as a power backup source. The engine's performance, emission characteristics, and combustion behaviour are evaluated across different load conditions. The results with distilled pine and eucalyptus oil are also presented in this section.

4.5.1 Combustion Characteristics

The combustion process can be divided into four distinct stages, and the primary findings related to the first two stages are illustrated in Fig. 4.15. It is evident that the ignition delay (ID) for various fuels, including diesel, P10E10D80, P15E15D70, P5E5D90, P50E50, E20D80, P20D80, was as follows: 8.5°C/A, 4.5°C/A, 7°C/A, 7.5°C/A, 9°C/A, 6°C/A, 9.5°C/A, respectively, with reference to the maximum in-cylinder pressure occurring before the top dead center (TDC). In the second stage of analysis, P10E10D80 exhibited a higher heat release rate (HRR) at an in-cylinder pressure of 77.31 bar and 4.5° crank angle before TDC. Additionally, the HRR of all diesel blends was investigated. The results demonstrate that the lowest HRR is contingent on in-cylinder pressure and ignition delay. The HRR (57.31 J/°CA) at a lower in-cylinder pressure (68.76 bar) was observed in the case of the P20D80 blend, followed by P50E50 (60.99 J/°CA at 70.61 bar). The highest cylinder pressure was recorded when the engine operated with a mixture of P10E10D80 (77.31 bar) and exhibited the maximum HRR of 69.17 J/°CA, occurring at an ignition delay of 4.5°CA before TDC. Further addition of pine and eucalyptus oil to diesel resulted in lower heat release rates. For instance, using P15E15D70, a heat release rate of 64.49 J/°CA, in conjunction with an in-cylinder pressure of 74.30 bar and an ID of 7°C/A before TDC, was observed. The overall analysis of P10E10D80 indicated the maximum HRR at the highest in-cylinder pressure and the shortest ignition delay. In the third and fourth stages of the analysis, the maximum and minimum HRR values are observed during the expansion stroke. The presence of small-sized particles may not undergo complete combustion during the expansion stroke, resulting in the formation of soot and the potential for maximum energy release at the exhaust outlet.

Likewise, the study reveals variations among different fuel blends, including diesel, P10E10D80, P15E15D70, P5E5D90, P50E50, E20D80, P20D80, and the distilled blend samples PD10ED10D80. PD10ED10D80 exhibited the shortest ignition delay due to its lower viscosity, resulting in a higher heat release rate at an in-cylinder pressure of 79.63bar, occurring 4.2°CA before top dead center (TDC). Additionally, the presence of oxygen in the sample may contribute to improved combustion, particularly at full load. Conversely, the P20D80 blend produced the lowest in-cylinder pressure, 68.76 bar, and experienced a higher ignition delay, primarily due to its lower cetane number and calorific values. When examining the pattern of ignition delay at maximum in-cylinder pressure before TDC, the sequence for conventional diesel and diesel blends is as follows: PD20D80 (9.2°C/A) > diesel (8.3°C/A) > ED20D80 (5.6°C/A) > PD10DE10D80 (4.2°C/A) (refer to Fig. 4.16).

The results related to heat release rates for all the tested samples are presented in Fig. 4.17. As evidenced, the heat release rate (HRR) is intrinsically linked to both in-cylinder pressure and ignition delay (ID). PD20D80 exhibited the lowest HRR (59.60 J/°CA) due to its higher ignition delay, lower calorific value, and cetane number. In contrast, ED20D80, with higher calorific value and cetane index, displayed an HRR of

69.27 J/°CA at an in-cylinder pressure of 77.69 bar and an ignition delay of 5.6°CA before TDC.

Furthermore, the engine running on PD10ED10D80, in comparison to conventional diesel, achieved a maximum in-cylinder pressure of 79.63 bar which can be attributed to its very low viscosity. This blend exhibited an HRR of 71.94 J/°CA and an ignition delay of 4.2°CA before TDC. These findings underscore that PD10ED10D80 resulted in the highest heat release rate at the maximum in-cylinder pressure and the shortest ignition delay, thanks in part to the presence of oxygen in the biofuels sample, further enhancing combustion performance, particularly at full load (refer to Fig. 4.18).

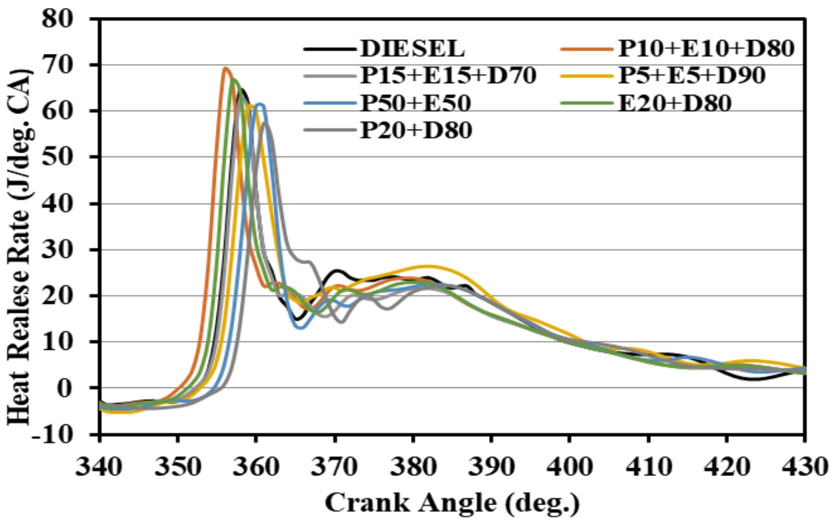


Fig. 4.15 Heat release rate with crank angle for pine oil and eucalyptus oil blend

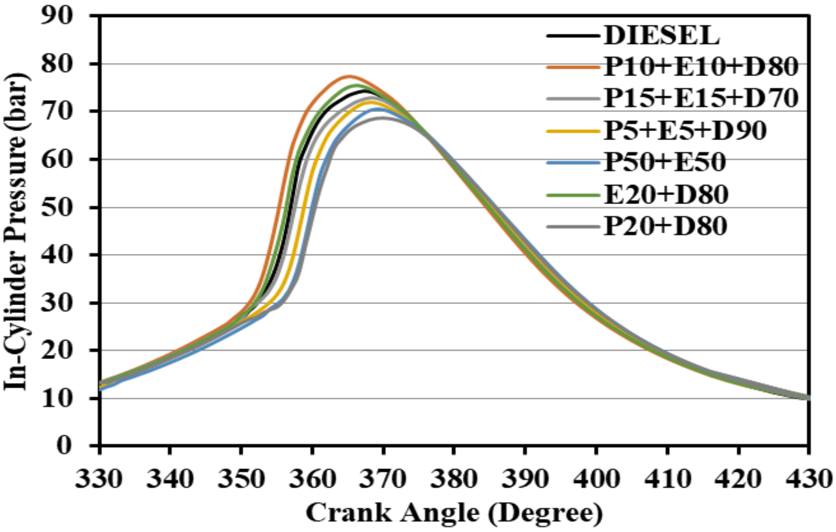


Fig. 4.16 Cylinder pressure with crank angle pine oil and eucalyptus oil blend

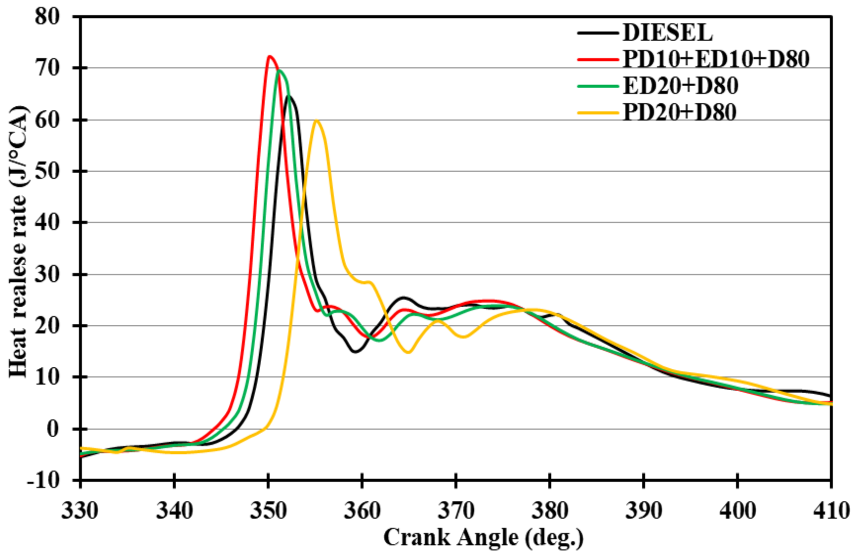


Fig. 4.17 Heat release rate with crank angle for distilled pine oil and eucalyptus oil

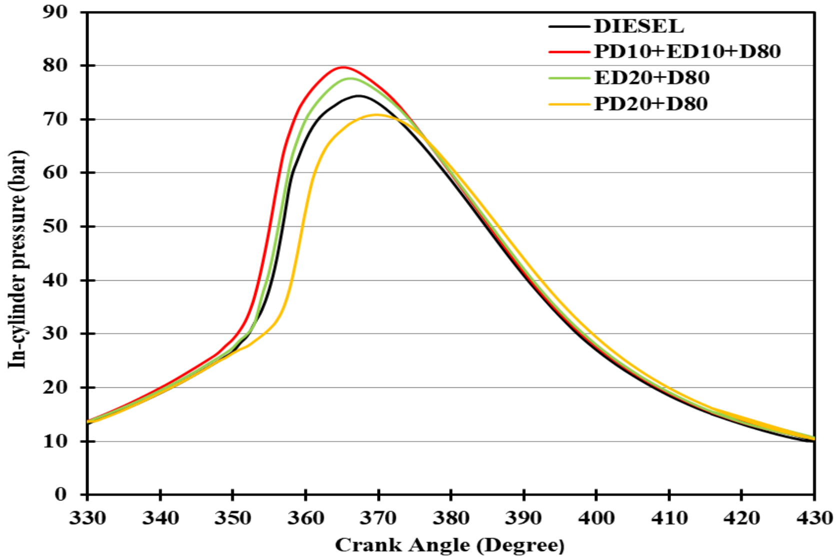


Fig 4.18 Cylinder pressure with crank angle for distilled pine oil and eucalyptus oil

The study assessed the performance of various diesel blends with respect to combustion duration and ignition delay. As shown in Fig. 4.19, all diesel blends, except for P20D80, exhibited favourable ignition delay and combustion duration characteristics. Specifically, P20D80 blend displayed an ignition delay of 9.5°CA and combustion duration of 65°CA. In contrast, E20D80 and P10E10D80 demonstrated ignition delays of 5.6°CA and 4.2°CA, along with combustion durations of 50°CA and 40°CA, respectively. These variations can be attributed to the notably low viscosity, flash point, and boiling point of P10E10D80 and E20D80. Conversely, the inferior ignition delay and combustion duration performance of P20D80 can be attributed to its calorific value and Cetane number.

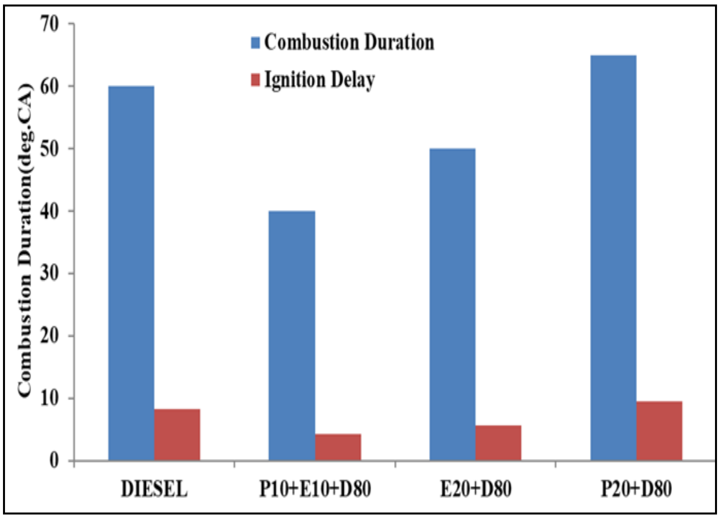


Fig 4.19 Variation between combustion duration and ignition delay

4.5.2 Performance Characteristics

Fig. 4.20 illustrates the variations in brake thermal efficiency with different load conditions for the tested fuel samples. Notably, the P20D80 blend exhibited the lowest thermal efficiency across all load levels, closely followed by P50E50. This can be attributed to the lower cetane number of these samples, which results in increased ignition delay and reduced heat release during premixed combustion due to their lower calorific values, consequently leading to decreased thermal efficiency. In contrast, engine operation with E20D80 yielded higher thermal efficiency compared to diesel. This improved performance can be attributed to the higher cetane index and calorific value of E20D80 relative to diesel, facilitating better combustion. Additionally, the presence of oxygen in the sample further enhances combustion. To delve deeper into engine efficiency, tests are performed by blending equal volumes of pine oil and eucalyptus oil with diesel. Notably, the engine efficiency increased with the P5E5D90 blend, reaching its maximum with a 10% pine oil and 10% eucalyptus oil blend in diesel. Further additions of pine and eucalyptus oil in diesel, such as P15E15D70, resulted in a reduction in thermal efficiency but remained higher than diesel. The inclusion of these biofuels improves the oxygen content and cetane number of the blend, contributing to enhanced combustion.

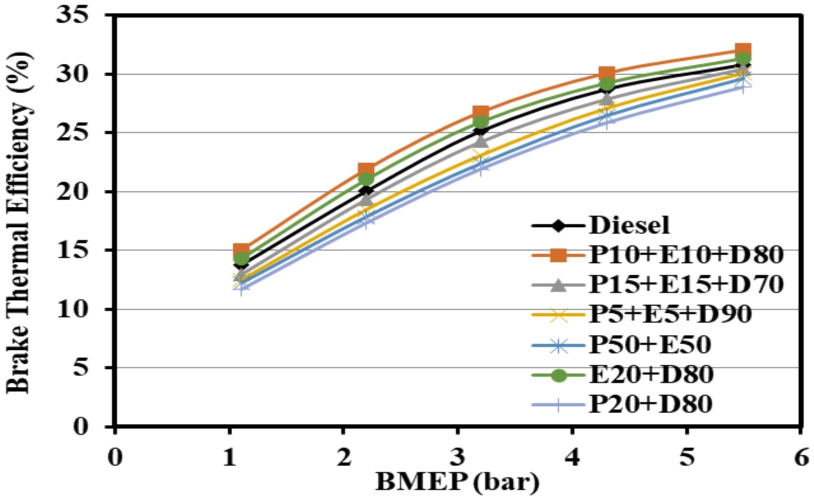


Fig. 4.20 BTE with BMEP for pine and eucalyptus oil blend

The assessment of brake thermal efficiency under various load conditions for different blends of distilled pine and eucalyptus oil blended with diesel revealed distinct performance characteristics (see Fig. 4.21). As compared to conventional diesel, PD10ED10D80 displayed the highest thermal efficiency, closely followed by ED20D80, whereas the efficiency of PD20D80 was even lower than that of the engine operating on conventional diesel alone. The elevated thermal efficiency in PD10ED10D80 and ED20D80 can be attributed to their higher calorific values and cetane index values, which contribute to improved combustion. Conversely, the reduced thermal efficiency in PD20D80 may result from its lower cetane value and reduced calorific values, which increase ignition delay and lead to less heat release during premixed combustion, consequently resulting in lower thermal efficiency.

The variation in brake specific energy consumption with load for diesel, P20D80, E20D80, P50E50, P5E5D90, P10E10D80, and P15E15D70 is presented in Fig. 4.22. The energy consumption trend is inversely related to brake thermal efficiency; higher energy consumption is observed when thermal efficiency is lower. It is evident from the figure, that P20D80 had the highest energy consumption, followed by P50E50. This phenomenon is due to the increased fuel consumption required to produce a specific power output, resulting from the lower thermal efficiency of engines operating with these two fuel samples, stemming from their lower calorific values. Conversely, for the other test samples, energy consumption is lower than diesel, owing to better combustion facilitated by their lower viscosity and higher heat release during combustion compared to diesel. At full load, the energy consumption for diesel, P20D80, E20D80, P50E50, P5E5D90, P10E10D80, and P15E15D70 is 8.7, 10, 6.4, 9.3, 8.01, 5.15, and 7.36 MJ/kWh, respectively. These results align with those reported by previous studies [54,165,176].

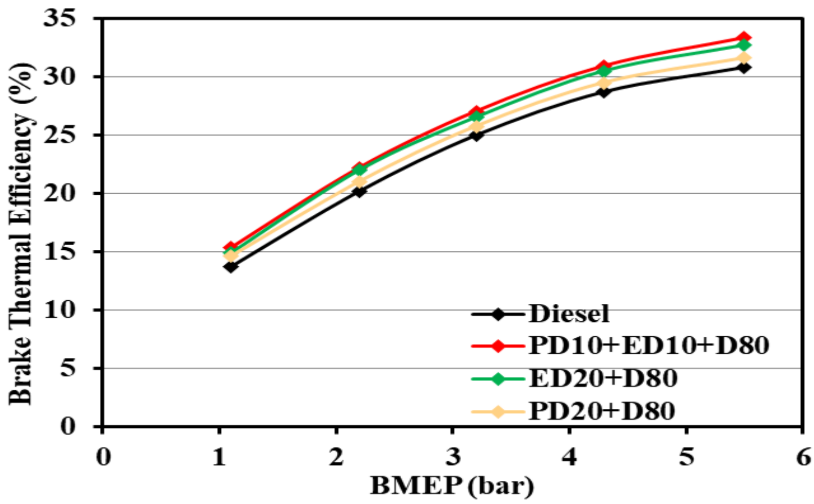


Fig. 4.21 BTE with BMEP for distilled pine and eucalyptus oil blend

Similar performance outcomes are observed when evaluating brake-specific energy consumption with varying loads for conventional diesel, distilled blends of pine oil and eucalyptus oil mixed with diesel (Fig. 4.23). The fuel requirements for generating a specific power output are directly linked to their lower calorific values. As a result, the energy consumption of PD20D80 was the highest, while for the remaining blends characterized by lower viscosity and higher calorific values, the brake-specific energy consumption was lower than that of conventional diesel.

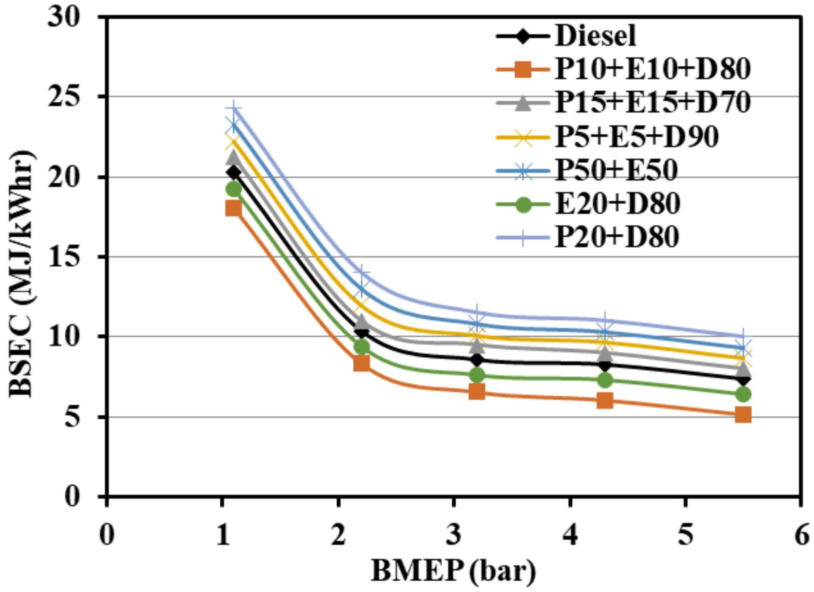


Fig. 4.22 BSEC with BMEP for pine and eucalyptus oil blend

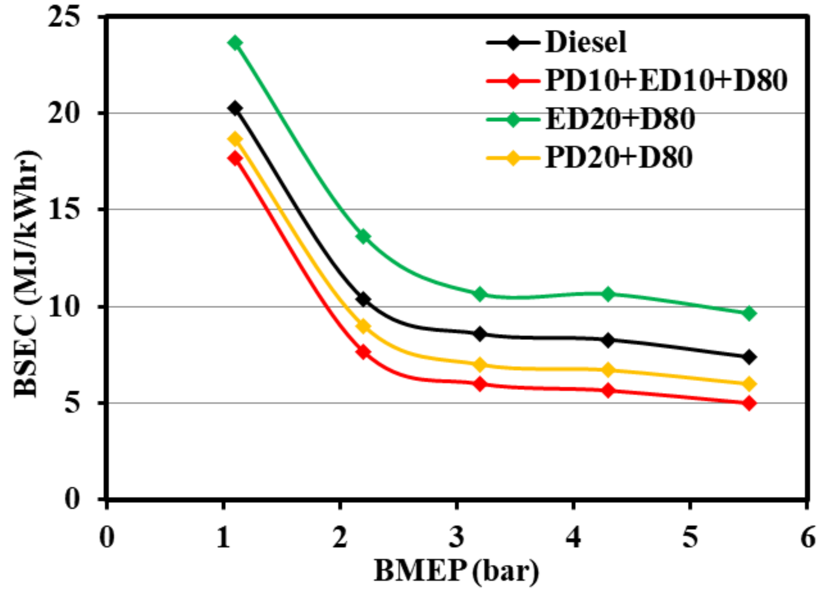


Fig. 4.23 BSEC with BMEP for distilled pine and eucalyptus oil blend

The exhaust gas temperature (EGT) for the various test fuel samples at different load conditions is presented in Fig. 4.24. It was observed that as the load increased, the exhaust gas temperature also increased for all the samples. This phenomenon was attributed to the higher load demanding more fuel injection, which subsequently raised the heat release, combustion temperature, and exhaust gas temperature. Specifically, at full load, the EGT values for diesel, P20D80, E20D80, P50E50, P5E5D90, P10E10D80, and P15E15D70 are 370°C, 398°C, 345°C, 380°C, 360°C, 335°C, and 350°C, respectively. The combustion process of pine oil and eucalyptus oil blends exhibited low heat release in the diffusion combustion phase, resulting in a reduction of the exhaust gas temperature. In contrast, the presence of 20% pine oil in diesel and an equal blend of pine oil and diesel led to higher EGT due to the elevated heat release in the latter part of combustion, which is not effectively utilized for useful work on the piston. These findings are consistent with previous research on pine oil [166-168,172]. At full load, the EGT for the distilled blends mixed with diesel, PD10ED10D80, ED20D80, and PD20D80 is 372°C, 335°C, 350°C, and 397°C, respectively (see Fig. 4.25). The lower EGT in the diesel blends (except for PD20D80) may be attributed to the lower heat released during the diffusion combustion phase.

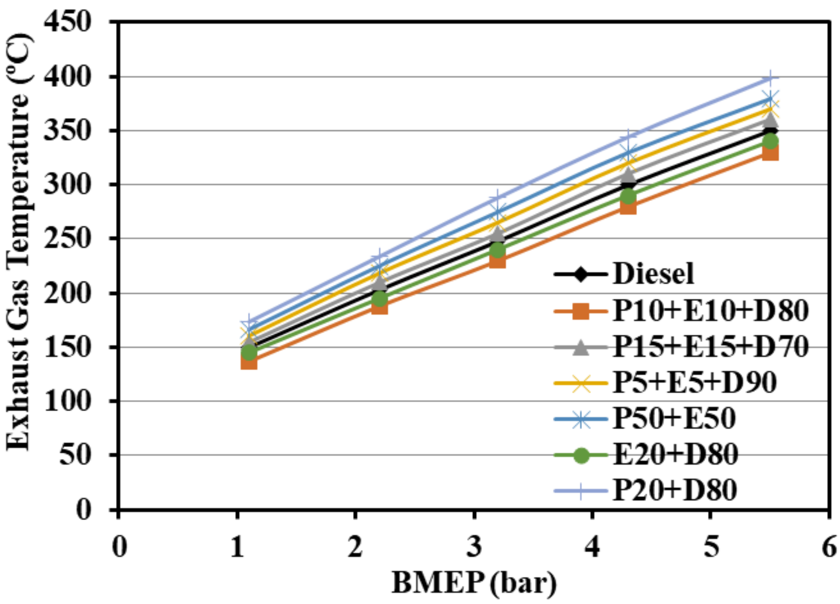


Fig. 4.24 EGT emission with BMEP for pine and eucalyptus oil blend

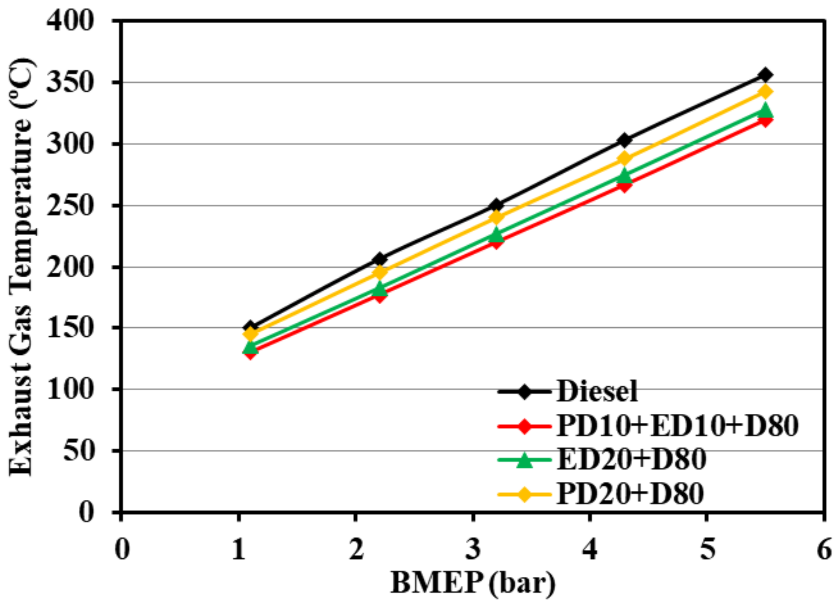


Fig. 4.25 EGT emission with BMEP for distilled pine and eucalyptus oil blend

4.5.3 Emission Characteristics

The hydrocarbon (HC) emissions associated with the blends of pine oil and eucalyptus oil in diesel are found to be lower than those of pure diesel (see Fig. 4.26). At full load, the relative reductions in HC emissions for P5E5D90, P10E10D80, P15E15D70, and E20D80 are 2.9%, 14.7%, 5.8%, and 8.8%, respectively. The improved burning rate of the fuel in the pine oil and eucalyptus oil blends, attributed to their higher calorific value and cetane number, increased the combustion temperature, leading to a decrease in hydrocarbon emissions. Interestingly, engines operating with P20D80 and P50E50 displayed higher HC emissions compared to diesel, with increases of 2.94% and 5.8%, respectively.

A similar pattern was observed in the analysis of hydrocarbon emissions (see Fig. 4.27). At full load, PD10E10D80 and ED20D80 reduced hydrocarbon emissions by 23% and 12.19%, respectively, while the engine running with the diesel blend of PD20D80 resulted in higher hydrocarbon emissions (5.41%). The improved hydrocarbon emissions are a result of higher calorific values, leading to increased combustion temperatures. Conversely, the increased hydrocarbon emissions may be due to low thermal efficiency and a high amount of fuel injected to produce equivalent power, resulting in incomplete fuel combustion.

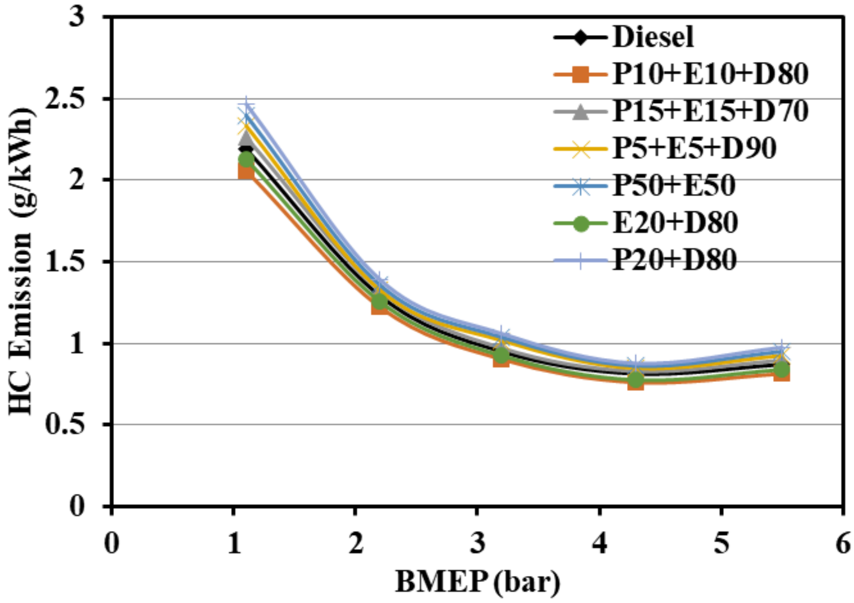


Fig. 4.26 HC emission with BMEP for pine oil and eucalyptus oil blend

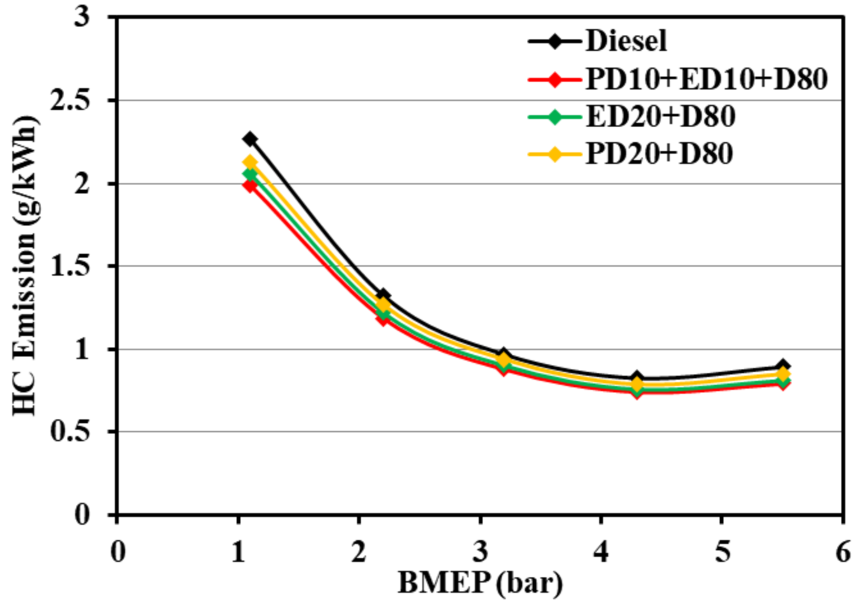


Fig. 4.27 HC emission with BMEP for distilled pine and eucalyptus oil blend

At full load, the carbon monoxide (CO) emissions for diesel, P20D80, E20D80, P50E50, P5E5D90, P10E10D80, and P15E15D70 are 15.1, 9.3, 12.7, 11.7, 8.8, and 10.5g/kWh, respectively (see Fig.4.28). The reduced CO emissions with the pine oil and eucalyptus oil blends can be attributed to the additional availability of oxygen and an increased combustion temperature, which enhanced the reduction of CO. However, despite the higher oxygen content in the fuel samples compared to diesel, CO emissions increased with P50E50 and P20D80. This can be explained by the fact that most of the heat released with these two fuel samples occurred during the second stage of combustion, reducing the time available for the conversion of intermediate carbon monoxide into carbon dioxide and thereby increasing emissions.

The results of carbon monoxide (CO) emissions for all the test samples are shown in Fig. 4.29. The results indicate that at full load, the pattern of CO emissions for the distilled blend is PD20D80 > diesel > ED20D80 > PD10ED10D80. The high oxygen availability and high combustion temperature in ED20D80 and PD10ED10D80 blends help in the oxidation of CO. Interestingly, although the blending substitutes are alcohol derivatives and are expected to have higher oxygen content than diesel, CO emissions are higher in PD20D80 than in diesel. This may be due to the lower combustion efficiency of the PD20E80 blend, resulting in incomplete oxidation of carbon monoxide to carbon dioxide.

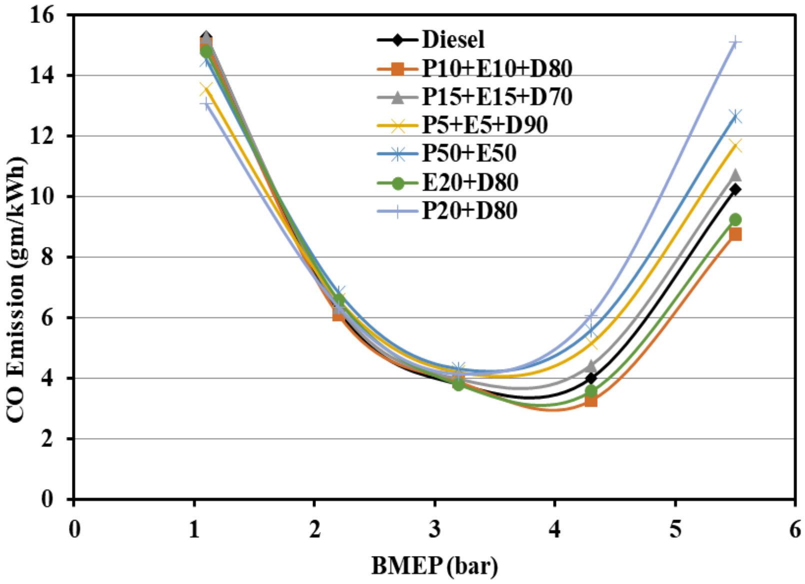


Fig. 4.28 CO emission with BMEP for pine and eucalyptus oil blend

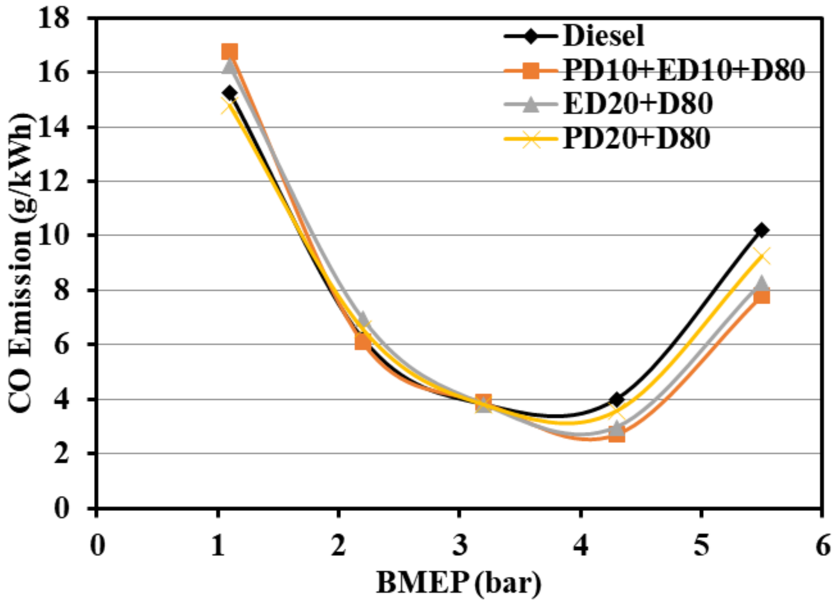


Fig. 4.29 CO emission with BMEP for distilled pine and eucalyptus oil blend

Fig. 4.30 illustrates the variation in nitrogen oxides (NOx) emissions with load for diesel, P20D80, E20D80, P50E50, P5E5D90, P10E10D90, and P15E15D70. In comparison to diesel, NOx emissions at full load increased by 4.76%, 1.4%, 8.59%, and 3.2% with E20D80, P5E5D90, P10E10D90, and P15E15D70, respectively. NOx is formed when nitrogen in the incoming air reacts with excess oxygen, both from the air and the fuel, at high temperatures. The improved combustion characteristics of these fuels resulted in higher combustion temperatures and, subsequently, increased NOx emissions.

As compared to the distilled blend of diesel (Fig. 4.31), NOx emissions at full load with ED20D80 and PD10ED10D90 increased by 4.94% and 8.60%, respectively. This increase may be attributed to the fact that thermal NOx is formed at high combustion temperatures and with high oxygen availability. In contrast, NOx emissions decreased by 3.90% with PD20D80 compared to diesel, which is likely due to the lower combustion temperature as the combustion deteriorated with these two fuels.

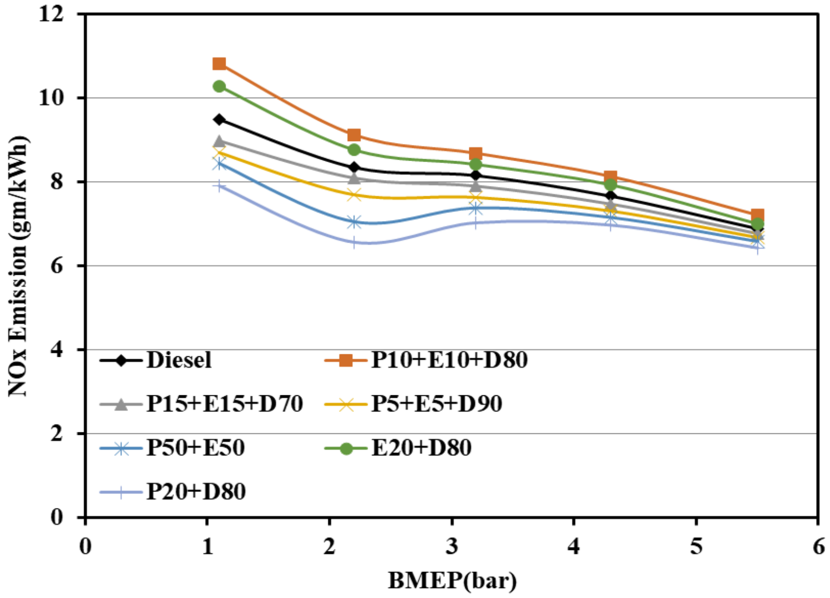


Fig. 4.30 NOx emission with BMEP for pine and eucalyptus oil blend

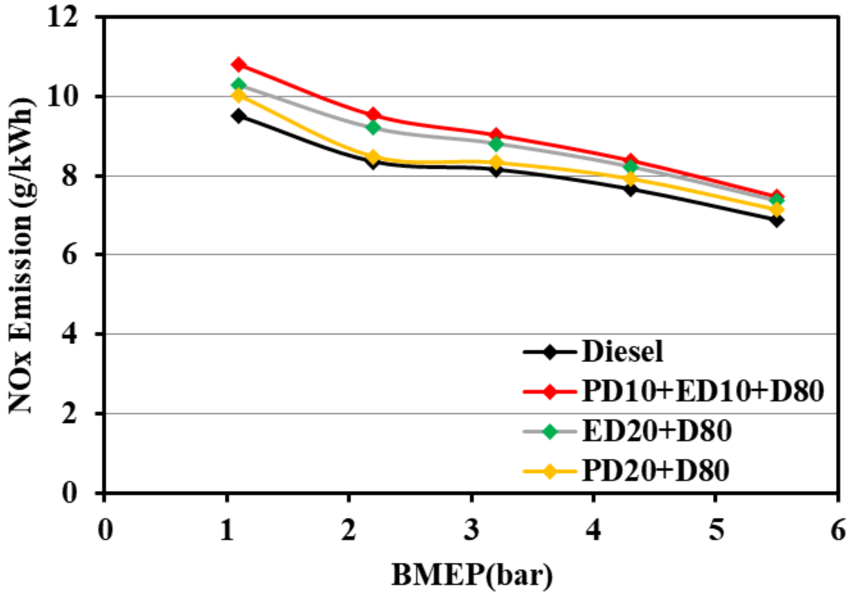


Fig. 4.31 NOx emission with BMEP for distilled pine and eucalyptus oil blend

In the absence of oxygen, larger fuel molecules are paralyzed and form particulate matter, increasing the smoke opacity of the distilled blend of pine oil and eucalyptus oil mixed with diesel. As the engine load increased, smoke opacity also increased, as more fuel was injected. The smoke opacity of the engine exhaust when operated with the test samples is shown in Fig. 4.32. It was observed that the smoke opacity was higher with P20D80 (9%) and P50E50 (3.4%) compared to diesel, whereas the smoke opacity for the other fuel samples, namely P5E5D90 (2.72%), P10E10D80 (20.14%), P15E15D70 (4.53%), and E20D80 (11.8%), was lower than diesel. The higher quantity of injected P20D80 and P50E50, coupled with their higher energy consumption and lower combustion temperature, caused the excess fuel molecules to burn inefficiently, resulting in higher soot emissions. Conversely, with P5E5D90, P10E10D80, P15E15D70, and E20D80, soot emissions decreased due to higher combustion temperatures, lower injected fuel amounts, and increased oxygen availability due to oxygenated fuel. The results of smoke opacity with the distilled blends are illustrated in Fig. 4.33. The figure shows that the smoke opacity with the diesel blend P20D80 (8.31%) is higher than diesel, while the blends, namely PD10ED10D80 (26.79%) and ED20D80 (19.13%), had lower smoke opacity. The reduced smoke opacity may be attributed to the high combustion temperature, lower fuel injection, and higher oxygen availability, resulting in better combustion of fuel molecules and subsequently less soot formation. However, the opposite is the case with PD20D80.

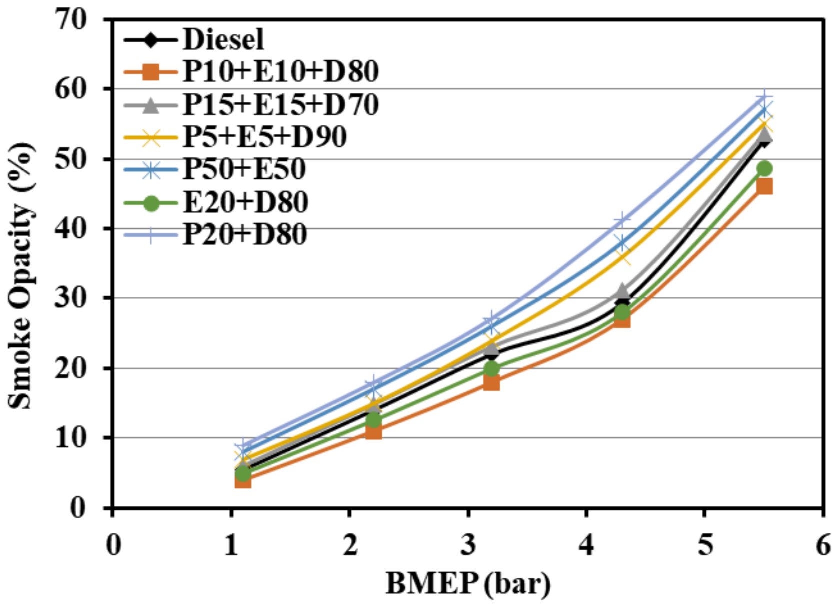


Fig. 4.32 Smoke opacity with BMEP for pine and eucalyptus oil blend

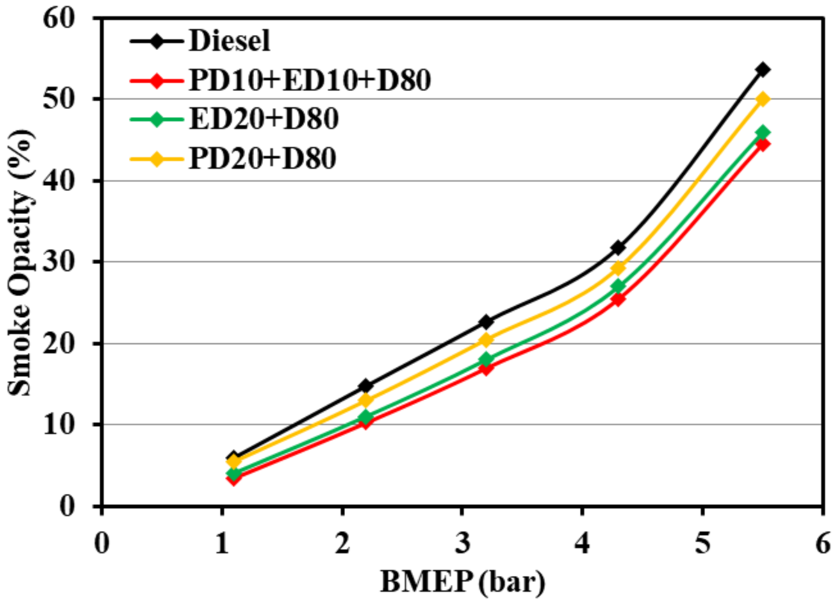


Fig. 4.33 Smoke opacity with BMEP for distilled pine and eucalyptus oil blend

4.6 Engine characteristics in dual fuel mode

In the fumigation studies, the mass flow rate of methanol was gradually increased from 0.0321 ml/min to 0.1411 ml/min. Accordingly, the governor adjusted the supply of P10E10D80 blend to maintain a constant speed The results show that with an increase in load, the percentage of methanol decreased, and vice versa for P10E10D80 (See Fig. 4.34). At the maximum methanol flow rate, approximately 56% of the P10E10D80 was replaced at 20% load, and up to 39% replacement of P10E10D80 was observed at 100% load. These findings align with previous reports. Chauhan et al. [195] also reported lower diesel replacement at higher loads. In another study, Sahin et al. [216] observed engine knock and unstable engine operation when a higher fraction of diesel was replaced with fumigation. Therefore, achieving an optimal balance between fumigation and diesel replacement is essential for ensuring better engine stability.

4.6.1 Performance characteristics

The brake-specific energy consumption (BSEC) at various methanol proportions in the inlet manifold is illustrated in Fig. 4.35. The figure demonstrates a decrease in BSEC as the methanol percentage increases, particularly at full loads. A notable 17% reduction in BSEC was observed when using 39% methanol compared to 17% at 100% load. These findings align with the results reported by Song et al. [217], who observed a decrease in BSEC at higher loads due to the efficient combustion of the fuel mixture. Furthermore, when the fumigation increased from 39% to 56% at 20% load, there was a slight improvement in BSEC, which might be attributed to the lower cetane index of methanol. It can be inferred from the study that higher methanol content at lower loads significantly extends the ignition delay and influences the combustion process. Similar observations are reported by Ajay et al. [218] and Cheng et al. [219].

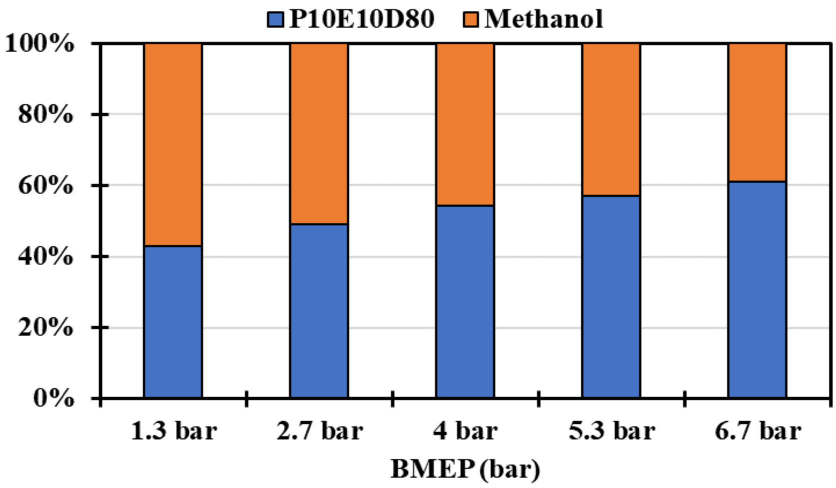


Fig. 4.34 Consumption of methanol/pine and eucalyptus oil

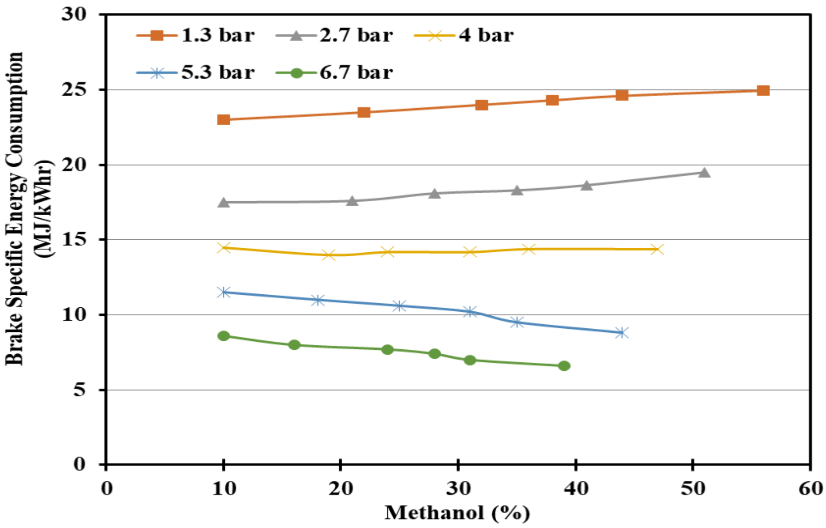


Fig. 4.35 Variation in brake specific energy consumption with varying percentages of methanol

The brake thermal efficiency plot exhibits a decreasing trend at lower loads (20%) and an increasing trend at higher loads (100%) with an increased percentage of methanol fumigation (Fig. 4.36). Similar results also observed by Tsang et al. in their fumigation study using ethanol. This phenomenon can be attributed to the fact that at 100% load, when the ignition is initiated by a blend of pine and eucalyptus oil, faster and more complete combustion of pre-heated fuel is achieved, leading to higher break thermal efficiency at 39% methanol compared to 10% at 100% load. The findings of this study are consistent with previous reports [220,221], which support fumigation as a recommended mode for diesel engine operation at higher loads.

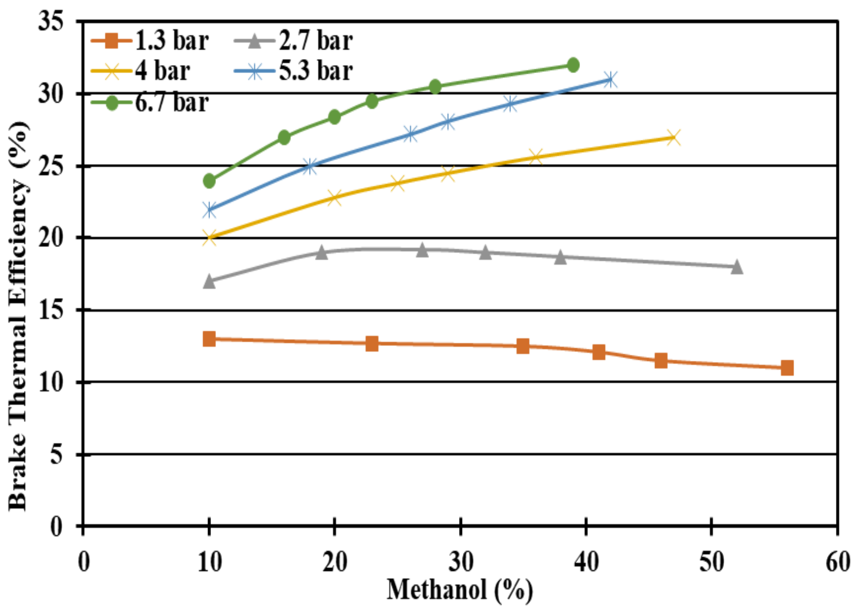


Fig. 4.36 Variation in brake thermal efficiency with varying percentages of methanol

Fig. 4.37 shows the variations in exhaust gas temperature (EGT) during methanol fumigation trials. It is evident that EGT decreases as methanol substitution in the inlet manifold increases. When a 135cc bike carburetor is employed, methanol fumigation leads to lower temperatures, primarily due to enhanced atomization and a higher lower heating value of methanol. Consequently, EGT is reduced as a result of the cooling effect induced by methanol fumigation.

Notably, for 20% and 40% loads, there is a small decrease in temperature. However, there is a noticeable drop in EGT while the engine runs in the 60% to 100% load range. In particular, EGT drops to 255°C with increased methanol substitution from 280°C at full (100%) load. Likewise, with 80% load, EGT starts at 250°C and decreases to 200°C as the concentration of methanol rises. The decrease in heat losses, a low cetane number, and a lower flash point are the main causes of the EGT drop. These findings align well with the concept that increasing methanol fumigation leads to a decrease in EGT.

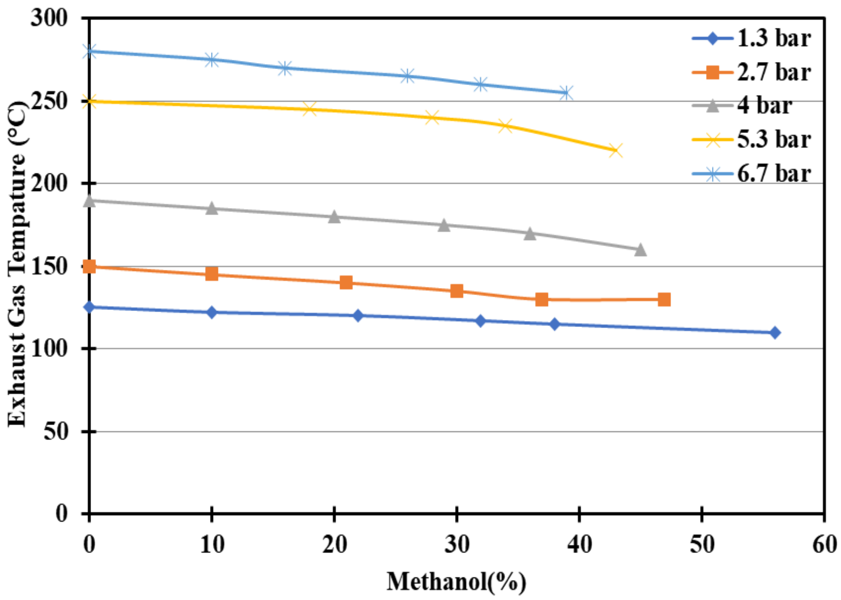


Fig. 4.37 Variation in exhaust gas temperature with varying percentages of methanol

4.6.2 Combustion characteristics

The heat release rate (HRR) for various methanol percentages at 100% load is depicted in Fig. 4.38). The results demonstrate a consistent increase in the HRR with the rising methanol proportion. This phenomenon is likely attributed to the rapid combustion of the methanol/air mixture after a prolonged delay period, resulting in a reduction in the amount of fuel to be burned during diffusion combustion, as a significant portion of the fuel is consumed during the premixed combustion phase. Similar explanations have been reported by Anandavelu, et al. [222] and Cheng et al. [223].

Furthermore, in addition to the premixed combustion phase, the subsequent combustion duration and the diffusion combustion phase are also observed to be altered. This slight reduction in combustion duration with methanol can be attributed to improved characteristics for evaporation and the presence of inherent oxygen. Table 4.4 and Fig. 4.39 provide information about the maximum and minimum mass fraction burn CA10, CA50, CA90, which are 3.391, 44.885, 89.111 at 39% methanol, and a minimum of 0.920 at 10% methanol, 25.303 at 26% methanol, and 64.472 with diesel at 100% load.

Table 4.4 Mass Fractions Burn and Methanol %CA10, CA50, CA90

Angle	MFB	Diesel	P10E10 D80	10%	16%	26%	32%	39%
CA10	0.10	2.232	2.8264	0.920	2.243	1.2446	2.2	3.371
CA50	0.50	31.213	38.001	29.472	35.271	25.303	33.53	44.885
CA90	0.90	64.472	71.941	73.311	75.694	76.769	78.49	89.111

The combustion duration, as shown in Fig. 4.40, 4.41, is determined by calculating the time from the initiation of the heat release rate until 90% of the total heat release rate is achieved. It decreases from 75°CA (crank angle degrees) for 10% methanol injection to 73°CA for 39% methanol injection at 100% load. This minor reduction in combustion duration when using methanol can be attributed to enhanced characteristics related to evaporation and the presence of inherent oxygen.

The variation in maximum in-cylinder (peak) pressure with load for different levels of introduced fumigated methanol into the cylinder is depicted in Fig. 4.42. At 100% load, it was observed that the peak pressure was 11.3% higher for 39% methanol compared to 10%, whereas at lower loads, the higher in-cylinder (peak) pressure was reduced with an increase in methanol proportion. This can be attributed to the extended premixed combustion phase due to a longer ignition delay. The ignition delay for various methanol proportions and the maximum pressure rise rate at 100% load are shown in Fig. 4.43. The results demonstrate a continuous increase in pressure rise rate and ignition delay period with the rise in methanol proportions, aligning with previous findings by Kasiraman et al. [224]. This behaviour is likely attributed to the lower cetane number of methanol, which considerably extends the chemical ignition delay period.

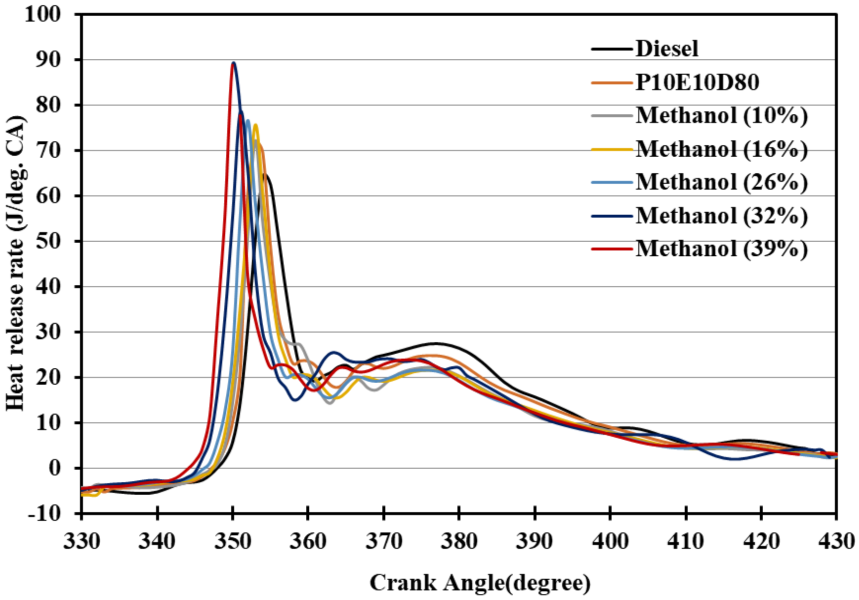


Fig 4.38. Variation in methanol percentage with Heat Release Rate at 100% load.

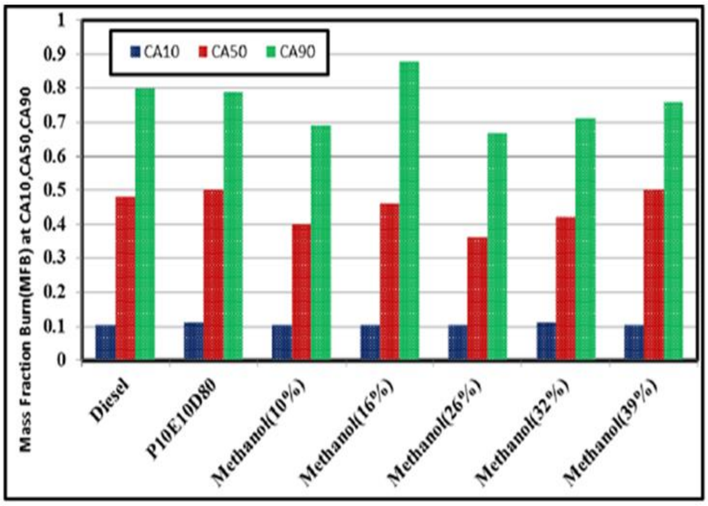


Fig 4.39 Mass fraction burn at CA10, CA50, CA90

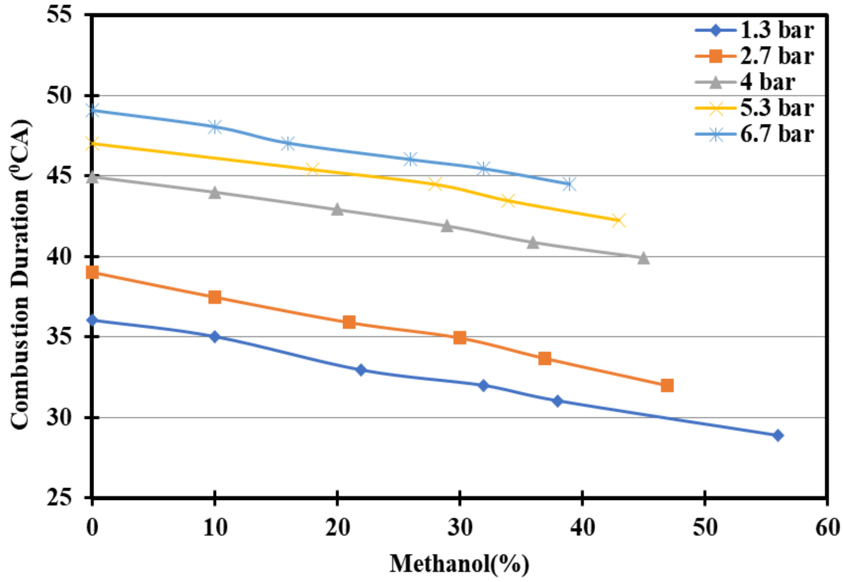


Fig. 4.40. Variation in combustion duration with methanol percentage

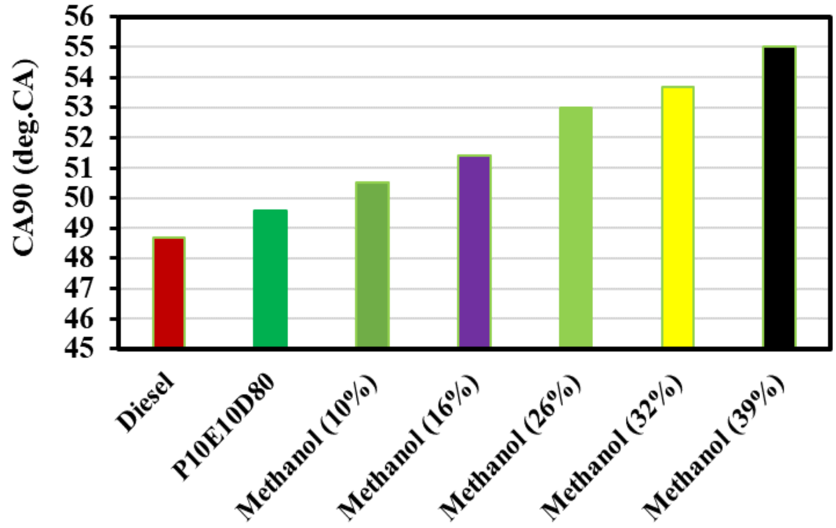


Fig. 4.41Variation in CA90 at 100% load

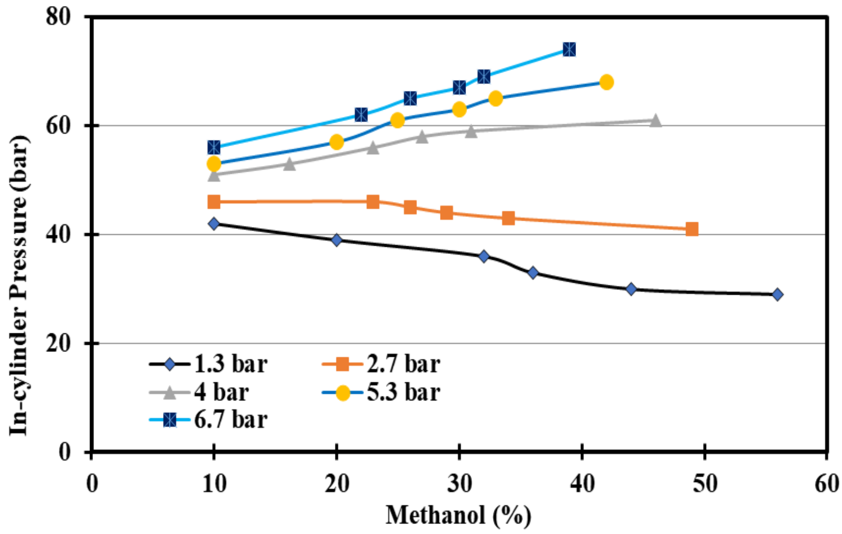


Fig. 4.42 Variation in methanol percentage with in-cylinder pressure

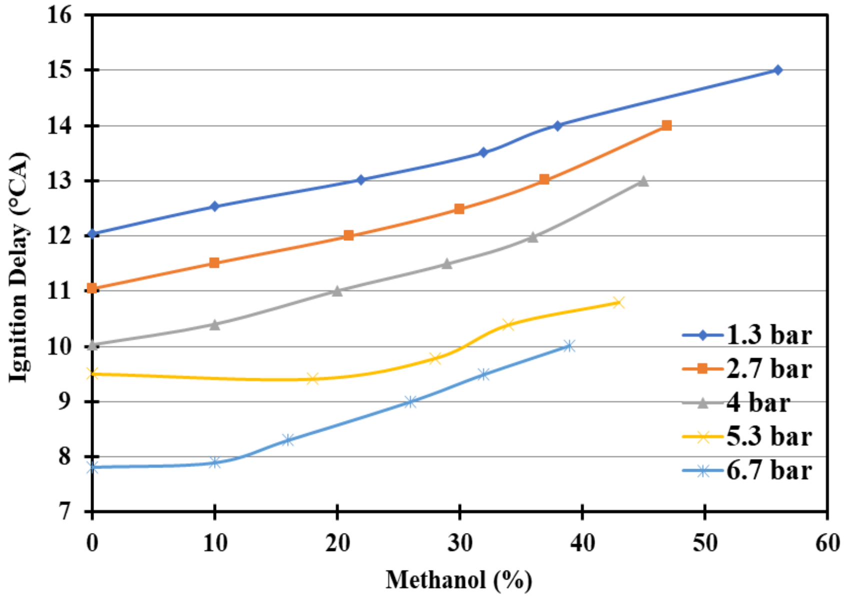


Fig. 4.43 Variation in methanol percentage with ignition delay

4.6.3 Emission characteristics

Figs. 4.44 to 4.48 depict the emission characteristics of methanol, which are examined at full load and expressed as Relative Exhaust Emissions (REE). The findings showed that when the amount of methanol in the manifold increased, the emissions of exhaust gas temperature (EGT), carbon monoxide (CO), smoke opacity, and nitrogen oxides (NOx) decreased. In contrast, Hydrocarbon (HC) emissions displayed a direct relationship with methanol proportion. This observed pattern may be attributed to the improved evaporation of methanol, which enhances combustion and reduces emissions. Methanol's intrinsic oxygen content and reduced latent heat of vaporisation also aid in the better oxidation of CO and soot. It should be mentioned, nevertheless, that the presence of methanol and lower oxygen levels at lower in-cylinder temperatures are what cause increased HC emissions at 20% to 40% loads. Through methanol fumigation, gain insights into the impact of different methanol proportions in the inlet manifold on various emissions, including EGT, HC, NOx, CO, and smoke opacity.

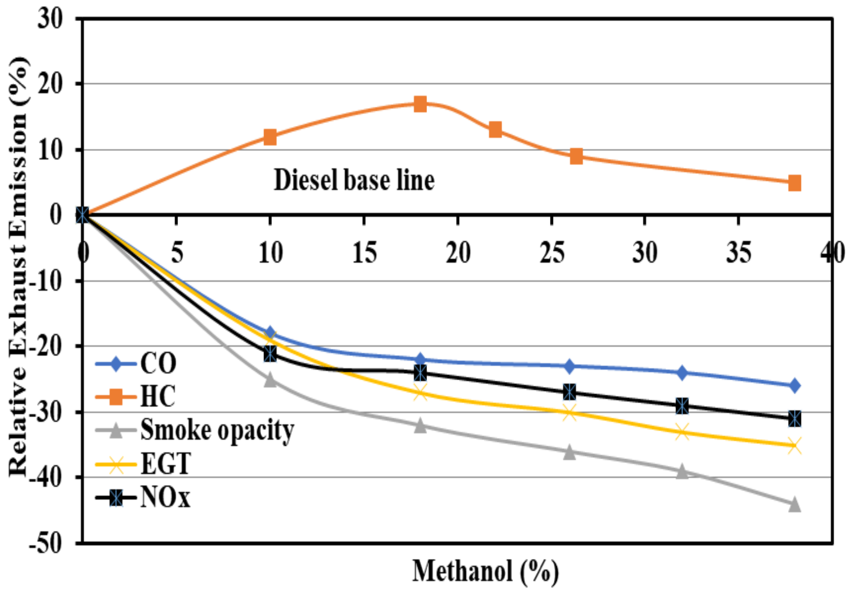


Fig. 4.44 Relative exhaust emission with varying percentages of methanol

Fig. 4.45 illustrates the variations in CO emissions in a conventional diesel engine under different loads with varying methanol proportions. It's important to note that air-fuel ratio and stoichiometric proportions play a significant role in influencing the levels of CO emissions [225]. CO emissions from the P10E10D80 blend gradually increased as the engine load increased while keeping the air volume constant. CO emissions during methanol fumigation may be affected by the addition of more oxygen to the combustion chamber. It's interesting to note that CO emissions go down from 20% to 100% load as methanol substitution increases. However, at 20% load conditions, rich mixtures tend to occur locally, leading to an increase in CO emissions despite an excess supply of air in the engine cylinder. This is attributed to poor combustion characteristics resulting from incomplete combustion. It is noteworthy that the inability to substitute methanol at a 20% load was caused by difficulties in controlling the governor-controlled quantity of P10E10D80. The graph shows that while the engine is run at 20% to 100% load, CO levels can drop by up to 39% during methanol fumigation. Nevertheless, CO levels increase at 20% and 40% loads, respectively, with

16% methanol fumigation. Moreover, it is noted that when the load is increased from 60% to 100%, CO levels drop significantly, up to 17% methanol fumigation.

The changes in unburned hydrocarbon (HC) emissions at varying methanol substitution rates and under varied load situations are shown in Fig. 4.46. When methanol fumigation is applied at 20% and 40% loads, higher levels of unburned hydrocarbons are observed. This could be attributed to the fuel having a reduced tendency to adhere to surfaces. Low exhaust gas temperature (EGT) brought on by extra air in the combustion chamber and inefficient fuel distribution could be contributing factors to the higher rate of UHC emissions. Lean fuel-air mixtures can cause more unburned particles to escape through the exhaust. It's noteworthy that Canakci et al. [226] and Whitten et al. [227], among others, have reported similar findings. UHC levels increase with the higher methanol substitution rate at 20% load. The increased Lower Heating Value (LHV) of methanol, which can cause inefficient combustion and ultimately lower the combustion temperature, may help to explain this. However, because to improved combustion at high loads, UHC emissions progressively drop up to 17% methanol replacement at 80% and 100% loads. In addition, UHC emissions do not change even with a 17% methanol replacement.

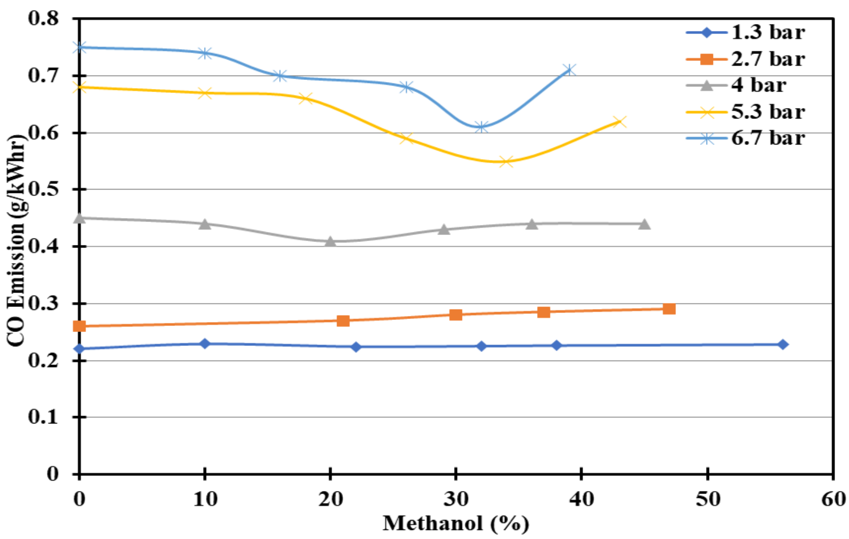


Fig. 4.45 Variation in carbon monoxide with varying percentages of methanol

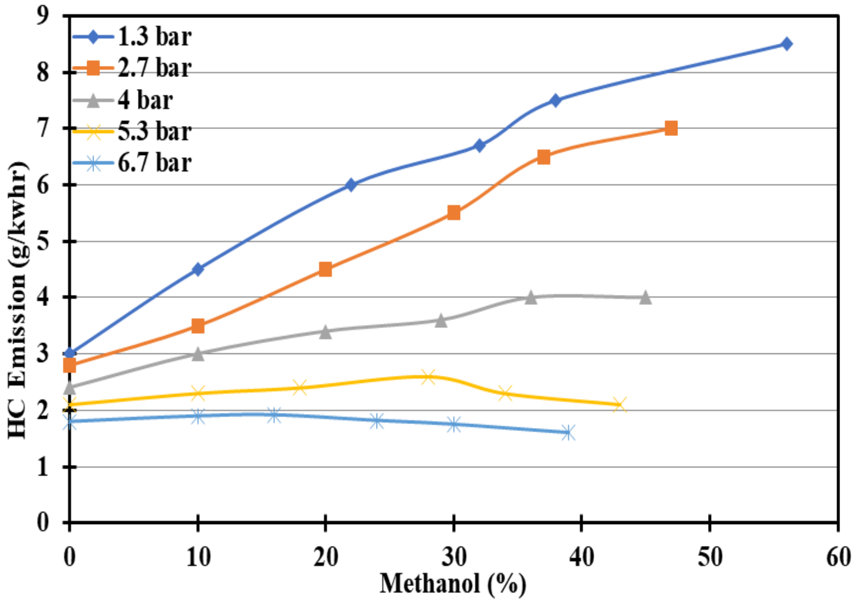


Fig. 4.46 Variation in unburned hydrocarbon with varying percentages of methanol

Fig. 4.47 presents the variations in NOx emissions at different methanol percentages and under various load conditions. The temperatures of ignition within the engine cylinder and the surrounding fuel/air combination have a major impact on the generation of Nitrogen Oxides (NOx). It is observed that NOx levels increase with rising combustion temperature due to differences in engine load, which aligns with findings by Challen et al. [228]. At lower loads, it's noted that NOx continues to increase with higher methanol fumigation percentages. In particular, 10% methanol injected through the inlet manifold reduces NOx at lower loads. Nevertheless, NOx rises with up to 16% methanol substitution at 40% load, at which point NOx levels start to grow. In the range of 60% to 100% load, methanol consumption is reduced to approximately 17%, as higher temperatures and pressures approach their maximum limits. This is because the governor automatically supplies the maximum quantity from the main injector at higher loads. Consequently, NOx formation decreases automatically up to a certain threshold. Furthermore, the use of methanol fumigation through a 135cc bike carburetor reduces combustion temperatures, leading to a reduction in NOx levels. Additionally, NOx decreases as methanol substitution increases.

Fig. 4.48 displays the variations in smoke opacity with different levels of methanol fumigation at various engine loads. Smoke opacity is a measure of how dark the exhaust smoke appears due to light obstruction by carbon particles. Across all load conditions, increasing the percentage of methanol fumigation leads to a reduction in smoke opacity. Smoke opacity is significantly reduced when up to 17% methanol fumigation is introduced at load conditions ranging from 60% to 100%. On the other hand, oxygenated gasoline generates less smoke opacity at 20% load because of the increased methanol content, which encourages extra air. But with P10E10D80 fuel, a shorter combustion time results in a greater combination of methanol and air, which raises smoke levels at higher loads. The findings indicate that at full load (100%), the use of 17% methanol fumigation results in the lowest smoke opacity. Smoke opacity levels can be influenced by factors such as restricted air filters, inaccurate injection timing, and inadequate maintenance.

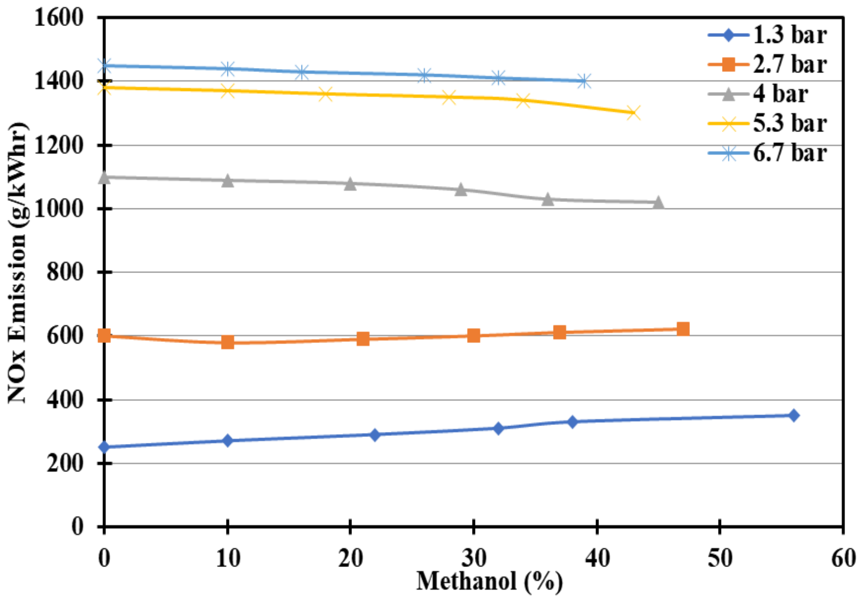


Fig. 4.47 Variation in oxide of nitrogen with varying percentages of methanol

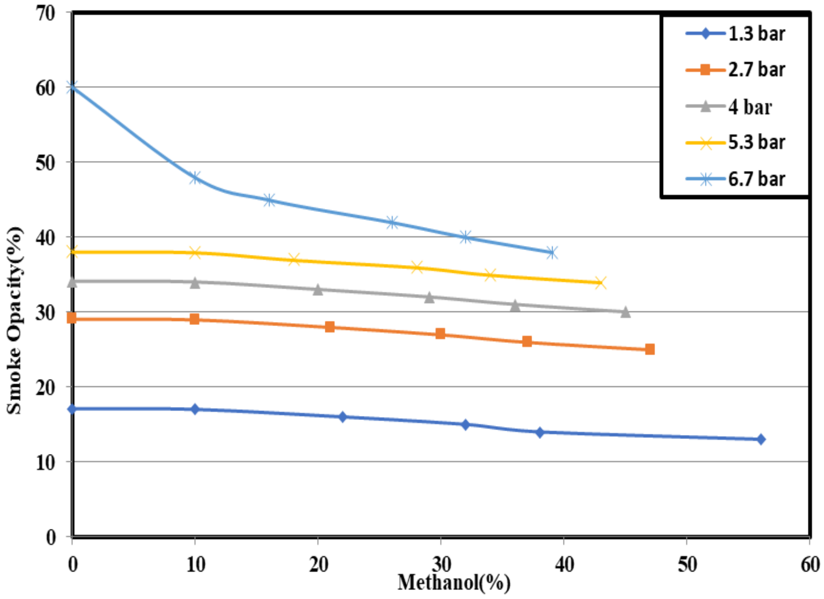


Fig. 4.48 Variation in smoke opacity with varying percentages of methanol

4.7 Economic Analysis

In 2017, the costs of diesel, pine oil, and eucalyptus oil were INR 53.61, INR 100, and INR 105 per litre, respectively. These prices have increased significantly by the current period.

Cost Breakdown for Biofuel Blend Preparation:

For preparing one litre of a specific blend (P10E10D80 - 10% pine oil, 10% eucalyptus oil, 80% diesel), the following costs are involved:

Table 4.5 present Cost analysis

Diesel cost	INR 87 per litre
Pine oil	INR 115 per litre
Eucalyptus oil	INR 120 per litre
Methanol	INR 120 per litre
Electricity cost for preparing one liter of sample	INR 6
Distillation of sample for water removal	INR 5
Cost of preparing one litre of P10E10D80	INR 140
Cost of operating engine with P10E10D80 (61%) and methanol (39%)	INR 99.8

1. There have been significant price increases for diesel, pine oil, and eucalyptus oil since 2017.
2. The preparation cost for the P10E10D80 blend is INR 104 per litre, which is little bit higher than the cost of diesel.
3. Methanol fumigation by carburetor operating an engine with a blend of 61% P10E10D80 and 39% methanol costs INR 99.8 per litre.
4. Despite the higher preparation costs, the blend offers potential cost efficiencies during maximum loading conditions.

CHAPTER 5

CONCLUSIONS

The current study endeavours to explore the novel blend of pine oil and eucalyptus oil as a means to substitute diesel. Blends of pine oil and eucalyptus oil in various compositions P5E5D90 (5% Pine oil and 5% Eucalyptus oil), 10P10ED80 (10% Pine oil and 10% Eucalyptus oil), P15E15D70 (15% Pine oil and 15% Eucalyptus oil), P50E50 (50% Pine oil and 50% Eucalyptus oil), E20D80 (20% Eucalyptus oil) and P20D80 (20%Pine oil) were explored. Further methanol was also fumigated to explore its effect on engine performance and emission.

The following key findings and conclusions have emerged from this research study:

1. The blends exhibit unique physicochemical properties that enable them to substitute 100% of conventional diesel while reducing the transesterification method cost by 15%.
2. Examination of profiles for eucalyptus oil and pine oil reveal that cyclo-hexene, 3,7-dimethyl-octa-1, 2-oxabicyclo [2.2.2] octane, hept-3-ene (Carene), 1,4-dimethyl-5-(1-Methylethyl)-cyclopentene, 6-diene, are found in major concentrations.
3. Physicochemical characteristics of the blends including density, viscosity, flash point, surface tension and calorific value were measured. The results met the established ASTM standards.
4. The viscosity of the neat biofuels and that of the blends were lower than diesel. Similarly, the calorific value and cetane index of the blends was lower than diesel. However, the cetane index and calorific value of eucalyptus oil was higher than diesel.
5. The distilled blends had lower kinematic viscosity, similar calorific value and cetane index as compared to diesel.

6. A study on the particle size distribution was carried out which shows that the diameter of the particle during spray was smaller than that of diesel.
7. One-year stability assessment of stored fuels showed an increasing trend in viscosity and density of the stored fuels. While the calorific value of the stored fuel was found to decrease over a period of one year. However, the measured parameters were still found to be within limits, hence, it can be said that the fuel samples have a long shelf life.
8. The engine experiments revealed that P10E10D80 blend had the highest in-cylinder pressure of 77.31 bar. While the lowest was observed with P20D80.
9. The heat release rate was observed to be the highest with P10E10D80. The start of combustion with the blend was found to be earlier than diesel on account of higher cetane index.
10. The thermal efficiency of the engine with the binary blend of E20D80 was higher than diesel, while the efficiency with P20D80 was lowest. Since, pine oil has a low calorific value. Moreover, the ternary blend of P10E10D80 gave the best thermal efficiency at all the loads.
11. The unburned hydrocarbon, carbon monoxide emissions and smoke opacity were found to be the lowest with P10E10D80 blend, while the emissions were highest with P20D80 blend. The improvement in combustion is due to low viscosity and higher cetane index of the ternary blend.
12. The oxides of nitrogen emission was found to be the highest with P10E10D80 blend. Elevated combustion temperature due to better combustion characteristics was the main cause for the rise in NOx emission.
13. The in-cylinder pressure and heat release rate were further elevated when the distilled blends of pine and eucalyptus oil were used. Among all the tested blends

PD10ED10D80 showed the highest in-cylinder pressure and heat release rate. Due to distillation, the oxygen containing compounds were further reduced resulting in higher calorific value.

14. The thermal efficiency with the distilled blend of PD10ED10D80 was found to be the highest among all the blends. The improvement in efficiency can be attributed to higher calorific value and cetane index value.
15. The HC, CO and smoke opacity was found to be further reduced with PD10ED10D80. As compared to diesel, the HC, CO and smoke emission reduced by 11%, 24% and 17%, respectively with the blend. Improvement in combustion due to lower viscosity and fast burning resulted in emission reduction.
16. The NOx emissions with PD10ED10D80, however, were 8% higher than diesel. The increase in temperature can be attributed to higher combustion temperature caused by higher heat release during the initial stages of combustion.
17. It was observed that the distilled blend PD10ED10D80 showed better engine characteristics than P10E10D80. However, the distillation of the oil increases the cost of the fuel. Therefore, for further analysis P10E10D80 blend was used.
18. To further substitute fossil diesel, methanol was fumigated in the intake manifold and P10E10D800 was the direct injected fuel. The percentages of methanol were varied with load. An improvement in thermal efficiency and decrease in energy consumption was observed at the maximum load condition.
19. Heat released as compared to diesel was high. The prolonged delay period allowed more amount of fuel to be ready to burn resulting in rapid release of heat.
20. HC, CO and smoke opacity decreases with methanol fumigation for all the load conditions and all the methanol percentages. At maximum load condition, the HC, CO and smoke opacity decreased by 11%, 5.3% and 37%.

21. NOx emission was found to be higher than P10E10D80 in single fuel mode at low load conditions. While the situation reverses at high load condition and a decrease in NOx emission was observed. The cooling effect of methanol plays a major role in the trend.

The results of the study show that the equal blend of 10% of pine oil and eucalyptus oil in diesel along with methanol fumigation would result in improvement in the unmodified engine's performance and emission characteristics. The study further shows that refined biofuels represent a promising solution for cleaner and more efficient energy.

Comparison of conventional diesel engines, P10E10D80 and PD10ED10D80 blend

Though all the results reported have used conventional diesel as a reference in figures 4.15 to 4.21and figures 4.24 to 4.25 however for the convenience of reader, a composite table depicting a comparison between diesel and selected fuel (P10E10D80) & (PD10ED10D80) has been added in conclusion.

Parameters	**Diesel and blend composition**		
	Diesel	P10E10D80	PD10ED10D80
HRR (Heat Release Rate)	64.49J/^{0}CA	69.17J/^{0}CA	71.94J/^{0}CA
In-cylinder pressure	74.30bar	77.31 bar	79.63bar
BTE (Break Thermal Efficiency)	30.82%	32.2%	33.1%
EGT (exhaust gas temperature)	370^{0}C	335^{0}C	330^{0}C

ASTM 7647 Stability Test of biofuel:

The stability of biofuel was accessed for 3 main parameters for ASTM 7647, which were Kinematic viscosity, calorific value point and density. The recommendation of ASTM for stability studies for these parameters were as follows:

- **Kinematic Viscosity** should remain between **1.2m^2/s and 2.5 m^2/s**.
- **Calorific Value** must stay blow **43.68 MJ/kg to 42.102MJ/kg**.
- **Density** should be increase**0.15% to31% 842 kg/m^3 to976.12 kg/m^3**

It is important to note that when the selected biofuel P10E10D80 was tested for the same, all the values fell within the acceptable limit. Moreover, there was no significant variation in the values during the stability studies.

The same has been incorporated in the conclusion section

Para	Before	After	limit
Kinematic Viscosity	1.2m^2/s	2.5 m^2/s.	1.2 m^2/s -3.5 m^2/s
Calorific Value	43.68 MJ/kg	42.102MJ/kg	43 MJ/kg -42 MJ/kg
Density	842 kg/m^3	976.12 kg/m^3	842 kg/m^3-976 kg/m^3

5.1 Future Scope

1. Comprehensive lifecycle assessments of biofuels and methanol blends will be performed to evaluate their environmental impact from production to combustion.
2. Long-term durability and reliability testing of engines using biofuel and methanol blends will be undertaken to ensure sustained performance and emission benefits.
3. Advanced combustion strategies, such as dual-fuel injection and homogeneous charge compression ignition (HCCI), will be researched to further enhance emission reductions and engine efficiency.
4. The integration of methanol fumigation and biofuel blends with hybrid and electric propulsion systems will be explored to create more sustainable and efficient energy solutions.
5. Regulatory and policy support will be advocated to promote the adoption of refined biofuels and methanol fumigation in the automotive and industrial sectors.

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Appendix–I

Table I Specification of Malvern Spraytec

Measurement Principle	Laser diffraction
Size range	0.1 – 2000µm
Optical Models	Mie theory and Fraunhofer Approximation including Patented Multiple Scattering correction
Lens ranges	750mm lens:2.0–2000µm (D_{v50}:5–1600 µm)
Working range	100mm at 0.5µm, >1m at 10µm
Concentration range	Minimum acceptable transmission: 5% (dependent on particle size range)
Detection system	35 element log-spaced silicon diode detector array
Light source	632.8nm, 5mW helium–neon laser
Maximum cquisitionrate	Continuous mode: 1 Hz Rapid mode:2.5kHz as standard, 10kHz with additional software feature key
Accuracy	Better than±1% on the D_{v50}
Precisions/repeatability	Better than±1% COV on the D_{v50}

Appendix II

Table II Test Rig Specifications

Component	**Specifications**
Engine type	Single cylinder, 4 stroke, CI engine
Bore*stroke	87.5*110 mm
Cubic capacity	0.661ltr
The compressionratio of	17.5:01
Rated output power	5.9(8) kW(hp)
Rated output rpm	1500rpm
Crankshaft height	203mm
Specific fuel capacity (SFC)	202+5%(gm/hp-hr)
Combustion system	direct injection

Appendix III

Table III Technical Specifications of AVL Smoke Meter

Measurement principle	Extinction measurement	
Operating temperature	+5-45°C subject to measuring accuracy	
	+1 -50°C measurement ready	
Humidity	Max. 90%, non-condensing	
Dimensions	395*285*136 (width*height (depth)	
Weight	3.5kg	
Opacity Chamber		
Measuring chamber heating	100°C	
Effective length	0.215m±0.0002m	
Maximum exhaust temperature	200°C	
Measurement parameters		
	Measurement Range	Resolution
Opacity	0 – 100%	0.1%
Absorption(kvalue)	0 – 99.99 1/m	0.01 1/m

Appendix IV

Table IV Technical Specifications of AVL Di-Gas Analyzer 1000

Parameter	Measurement Range	Resolution	Precision
CO	0 – 15%vol.	0.01%vol.	<0.6%vol.±0.03%vol. ≥0.6%vol. ±5% of reading
CO_2	0 – 20%vol.	0.01%vol.	<10%vol.±0.5%vol. ≥10%vol. ±5% of reading
HC	0 – 30000ppm vol.	1ppm vol.	<200ppm vol. ±10ppmvol. ≥200ppm vol. ±5% of reading ≥10000ppm vol. ±10% of reading
NO	0 – 5000ppm vol.	1ppm vol.	<500ppm vol. ±50ppmvol. ≥500ppm vol. ±10% of reading
O_2	0 – 55%vol.	0.01%vol.	<2%vol.±0.1%vol. ≥2%vol.±5% of reading

Appendix V

Table V Accuracy and uncertainty of engine

Measurement	Range		Accuracy	Uncertainty (%)	Measurement techniques
	Min	Max			
Burette measurement	00	100cc	±1cc	±1cc	Volumetric Quantity
Speed Sensor (RPM)	00	1000	±10 rpm	±1	Magnetic pick-up system
Digital Watch	1sec	65sec	±0.6sec	±0.20	Manual stop watch
Examine BTE	-	-	-	±0.68	NDIR techniques
Examine BSEC	-	-		±0.68	NDIR techniques
Examine BMEP	-	-	-	±0.68	NDIR techniques
Crank angle Encoder (TDC)	-		$\pm 1^{o}$	±0.25	Magnetic pickup system
Pressure Encoder	-		1bar	0.15	Magnetic pickup system
Fuel consumption (total)	0	100%	±1.00%	±1	Smoke meter

BIOGRAPHY

Kirat Singh is working as a Director and Founder of Nextup Robotics PVT LTD Ghaziabad Uttar Pradesh, India. He did his M.Tech in Thermal Engineering from Delhi Technological University, He is pursuing his Ph.D. from Delhi Technological University and has over 19 years of experience in teaching and research working as a Research Scholar at the Centre for Advanced Studies and Research in Automotive Engineering (CASARE), Mechanical Engineering Department. He is involved in exploring alternative fuel engines for vehicles and off-road engines. His research expertise includes alternative fuels with special emphasis on Biodiesel. he has published 3 research papers in SCI, 2 Grant Patents and 3 CONFERENCE research papers in peer reviewed journals/conferences. He is working for more than 12 years as an Assistant Professor in the department of Mechanical Engineering, R.KG.I.T Ghaziabad. He is keen to use a decentralized energy model for rural electrification and is interested in collaborative research and working with leading researchers.

Right Now he dedicatedly works in robotics and Automation, the team of Nextup Robotics Developed three major projects on Arm Robot, Humanoid robots, cleaning Robot, Foldable cooler, and many more projects in the pipeline.

Present Working place

Company Profile: Nextup Robotics Private Limited

Nextup Robotics is a newly developed startup. This is a team of professionals. It Work on New Ideas and Vision and are fully involved in robotics and also do other innovations. So far, we have developed the concept of a humanoid Robot, Arm Robot, and Folder Cooler,

Company Education Model:

For engineering college

Nextup Robotics offers training courses on robotics and automation, including a special feature on artificial intelligence, designed for **B. TECH, M. TECH, Ph.D., DIPLOMA, and ITI** students. The courses are available in various durations, including **one-day, five-day, fifteen-day, one-month, one-and-a-half-month, three-month, six-month, and one-year** skill training programs.

Company Product:

Industrial Arm Robot

Cleaning Robot

Humanoid Robot

Foldable cooler

Automatic Door Controller Design

PUBLICATIONS

1. Study of combustion, performance and emissions characteristics of oxygenated constituents and methanol fumigation in the inlet manifold of a diesel engine, N Kumar, K Singh, Sustainable Energy Technologies and Assessments 49, 101748 (SCIE IF-8)
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1. Kirat Singh, Shubham Singh, Dhirendra Kushwaha, hydraulically powered power Robot, Grant Patent Published 23/07/2021.
2. Kirat Singh, Shubham Singh, Dhirendra Kushwaha, Foldable cooler, Grant PatentPublished-10/09/2021.

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