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Combustion and Emission Behavior of Fusel Oil–Gasoline Blends in a Modern Spark-Ignition Engine

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Abstract
The global pursuit of sustainable energy solutions has intensified interest in alternative fuel sources, particularly bio-derived alcohol fuels such as fusel oil. This study investigates the combustion characteristics and emissions mitigation potential of fusel oil–gasoline blends (F10, F20, F30) in a turbocharged gasoline direct injection (GDI) engine under steady-state conditions at 2000 rpm and 40% throttle load. Employing detailed combustion analysis and emissions measurement, the study highlights that blending fusel oil, a renewable by-product of ethanol production, can significantly enhance combustion completeness due to its inherent oxygen content. Methodologically, each fuel blend was tested systematically, ensuring high data reproducibility by averaging results over 200 combustion cycles per test condition, repeated three times. Results showed fusel oil blends accelerated combustion initiation, as evidenced by an earlier peak in-cylinder pressure and rate of heat release. Specifically, HC emissions decreased notably from 125 ppm (F0) to 90 ppm (F30), signifying improved combustion efficiency. Conversely, CO emissions increased from 0.22 mg/kg (F0) to 0.45 mg/kg (F30), indicating localized incomplete combustion due to fusel oil's high latent heat of vaporization. This research uniquely explores fusel oil's viability in turbocharged GDI engines without hardware modifications, contributing valuable insights toward optimizing biofuel integration and reducing harmful emissions, thus supporting sustainable automotive advancements.

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1. Introduction

The increasing global energy consumption and the pressing issue of environmental degradation have accelerated the pursuit of alternative fuel sources, particularly in the transportation and industrial domains [1–3]. Among the viable options, fusel oil, an ethanol distillation by-product derived from molasses, has emerged as a promising candidate [4–6]. Rich in higher alcohols, fusel oil shows potential for blending with gasoline to improve combustion characteristics and lower harmful emissions, offering a partial solution to the limitations of conventional fossil fuels [7–10]. However, its application still requires more in-depth investigation, especially concerning its performance, combustion behaviour, and emission profile in spark-ignition (SI) engines.

Recent studies have highlighted that biofuels and alcohol-based fuels offer considerable potential for enhancing engine performance and minimizing emissions due to their high octane numbers and oxygen content. Fusel oil, which contains a mixture of higher-chain alcohols, is one such candidate; however, comprehensive studies on its application remain limited. The demand for environmentally friendly fuels compatible with existing engine systems has become increasingly urgent. For instance, various investigations into ethanol-gasoline blends have consistently shown improvements in combustion efficiency and significant reductions in emissions [11–13]. One study evaluated a spark-ignition single-cylinder four-stroke engine using compression ratios of 8:1, 8.5:1, and 9.12:1 [14]. Six fusel oil-gasoline blends (ranging from F0 to F50 in 10% increments) were tested, with the F30 blend at a 9.12:1 compression ratio yielding the best results, an efficiency gain of 6.91% and a 2.35% decrease in Brake Specific Fuel Consumption (BSFC). While CO_2 emissions increased, notable declines in CO, HC, and NOx emissions were observed. In another experiment, the impact of varying diethyl-fuel blend ratios on engine performance was studied under different lambda conditions and engine speeds, with optimal performance found at lambda values between 2.1 and 2.2 for the D40F60 blend [15–17].

Another study showed that blending fusel alcohols can improve combustion efficiency while overcoming common problems in pure ethanol blends, such as increased vapour pressure and low energy density. The blends can improve fuel consumption efficiency comparable to, and even exceed, pure ethanol at higher blending levels. The most optimal blend consisted of 90% iso-butanol and 10% 2-phenyl-ethanol, blended at 45% with gasoline, resulting in a fuel efficiency increase of 4.67%. Fusel oil, a by-product of alcohol fermentation, was studied as an alternative fuel in a direct injection diesel engine in a separate study conducted by [18–20]. Tests were performed on a single-cylinder CI engine at 2600 rpm with varying loads (2.5–12.5 Nm). The results showed that diesel–fusel oil blends increased specific brake energy consumption and reduced exhaust gas temperatures compared to pure diesel. Although CO, NOx, and smoke emissions were reduced, using fusel oil in high concentrations negatively impacted engine performance, including a decrease in maximum pressure in the cylinder. This indicates that although fusel oil has the potential to be a low-emission fuel, the compromise on engine performance remains a challenge that needs to be considered.

Recent research has demonstrated growing interest in oxygenated fuels, especially fusel oil, due to its high octane rating and oxygen content, which enhance combustion and reduce unburned hydrocarbons [4,5,21]. For example, the dual benefit of fusel oil in improving combustion efficiency and reducing NOx when blended with diesel has been highlighted in previous research, and its role in enhancing combustion characteristics in CI engines has also been reported in the literature [5]. Previous studies have also reported improved flame propagation and reduced HC emissions when using alcohol-based blends such as acetone-butanol-ethanol (ABE) in spark-ignition engines [21–23]. Despite these developments, the application of fusel oil in turbocharged Gasoline Direct Injection (GDI) engines remains underexplored. Previous works have focused on naturally aspirated or compression ignition engines, often requiring engine modifications or fusel oil combined with other alcohols such as ethanol or diethyl ether [15,18,24]. Using turbocharged GDI systems introduces unique combustion dynamics due to higher in-cylinder pressures and temperatures, significantly influencing the behaviour of oxygenated fuels. Therefore, the present study fills a critical research gap by systematically evaluating fusel oil–gasoline blends in a turbocharged GDI engine under steady-state conditions. Unlike earlier studies, this research uses a modern engine platform without extensive modifications, offering insights into real-world applications of fusel oil. The novelty lies in investigating combustion phasing, ROHR, ROPR, and emissions concerning fusel oil blends (F10–F30) in a turbocharged environment, an area still scarcely reported in the literature [7,18,25].

This research evaluates the feasibility of operating a turbocharged 1.8L gasoline direct injection (GDI) engine using gasoline blended with fusel oil, a renewable by-product of alcohol production. The use of

a turbocharged GDI platform is intentional, as it reflects current automotive trends in high-efficiency, downsized engine technologies that rely on boosted intake pressures and advanced combustion control to optimize performance and fuel economy. Turbocharging increases air intake density, thus enhancing power output. It also raises in-cylinder pressures and temperatures, creating realistic conditions for assessing alternative fuels' combustion and emission behaviours. This setup provides a suitable foundation for evaluating oxygenated blendstocks, such as fusel oil, under conditions relevant to modern vehicle applications.
In recent literature, advanced fuel and combustion strategies such as multi-jet hydrogen injection with cavity flame holders in supersonic flow fields, port water injection for thermal management and co-optimization, and the identification of top bio-blendstocks for turbocharged gasoline engines have shown promise in improving combustion efficiency and lowering emissions [26–28]. These studies highlight the increasing importance of oxygenated, renewable, and thermally stable blendstocks in supporting clean and efficient spark-ignition engine operation. While such approaches involve sophisticated system integration, this study focuses on a more practical pathway by assessing fusel oil as a drop-in bio blendstock that can be adopted with minimal modification to existing engine hardware. This work explores the effects of 10%, 20%, and 30% fusel oil–gasoline blends on key combustion and emission parameters, including in-cylinder pressure, brake-specific fuel consumption (BSFC), rate of heat release (ROHR), rate of pressure rise (ROPR), brake thermal efficiency (BTE), hydrocarbon (HC), and carbon monoxide (CO) emissions explicitly. Although CO_2, NO_x, and particulate matter were not included at this stage, the primary objective is to assess combustion quality and fuel utilization efficiency, as reflected through HC and CO trends. The findings are expected to contribute to developing environmentally sustainable engine technologies and support the broader integration of renewable fuel candidates into turbocharged SI engine platforms.
This study investigates the application of fusel oil–gasoline blends in a turbocharged gasoline direct injection (GDI) engine under steady-state conditions, with a fixed engine speed of 2000 rpm and 40% throttle load. Using a turbocharged GDI engine platform reflects current automotive trends and offers precise control over fuel injection and combustion, making it highly suitable for evaluating alternative fuels. While previous research has primarily focused on naturally aspirated or compression ignition engines, studies involving fusel oil in turbocharged GDI systems remain scarce. This gap underscores the novelty and relevance of the present work. By analyzing key parameters such as in-cylinder pressure, combustion phasing, brake thermal efficiency, fuel consumption, and regulated emissions (CO and HC), the study provides comprehensive insights into the practical feasibility of fusel oil as a renewable fuel blend. Future research will expand upon these findings by incorporating additional emission metrics, including CO_2, NO_x, and particulate matter, using advanced diagnostic instrumentation. Overall, the results contribute to the broader development of cleaner and more sustainable engine technologies. The remainder of this paper is organized into sections covering the experimental methodology, results, discussion, and conclusions.

2. Methodology

Table 1 outlines the fundamental properties of fusel oil and its blends with gasoline, highlighting essential metrics such as energy density, octane number, and oxygen content. A 10% fusel oil blend was selected as an optimal starting point to balance engine performance with emission mitigation while avoiding potential drivability issues and combustion instability associated with higher blend ratios. Engine specifications utilized in this study are detailed in **Table 2.** Before conducting the experiments, each fuel blend was mixed for 30 seconds to ensure homogeneity. Emission parameters, including carbon monoxide (CO), hydrocarbons (HC), and oxygen (O_2), were measured using a Horiba MEXA-7100DEGR gas analyzer.

Test Procedure Parameters

The engine tests were conducted at a fixed speed of 2000 rpm with a 40% throttle load to maintain consistent operating conditions. Each fuel blend was evaluated under these settings, with data collection spanning 30 minutes per test to ensure steady-state operation and reliable measurements. Each parameter was averaged over 200 continuous combustion cycles to enhance accuracy, providing a representative dataset. Laboratory conditions, including temperature and humidity, were carefully regulated throughout the experiments. Each measurement was repeated three times, and the average values were used for analysis to improve data robustness. The standard error propagation method was applied to estimate uncertainties, considering potential instrumental errors, environmental variations, and test repeatability. This approach helped minimize random deviations, ensuring the precision and reliability of the results. **Figure 1** illustrates the laboratory's engine, dynamometer, and experimental setup.

In this study, the experimental approach used has considered the basic principles of DOE by maintaining controlled test conditions, such as fixed engine speed at 2000 rpm, 40% throttle load, and three repetitions of measurements for each parameter, and the average is taken from 200 continuous combustion cycles. In addition, we apply the standard error propagation method to account for measurement uncertainty, thus ensuring consistent and reproducible results. In the future, we plan to develop this experimental approach by implementing complete DOE methods, such as factorial design or response surface methodology (RSM), to evaluate the effects of interactive variables more systematically on engine performance and emissions.

Source of Test Materials

The fusel oil used in this study was obtained from a local bioethanol production facility located in [insert location, e.g., Pahang, Malaysia or Banda Aceh, Indonesia, depending on actual location], which produces fusel oil as a by-product of molasses-based ethanol fermentation. Before its use, the fusel oil was filtered to remove any solid impurities and stored in sealed, dark glass containers at ambient laboratory conditions to maintain its stability. The chemical composition of the fusel oil (notably its oxygen content, density, and heat of vaporization) was verified by gas chromatography and standard ASTM methods to ensure consistency and compatibility with gasoline blending. The gasoline (F0) used in the study was commercial RON 91 grade gasoline, acquired from a certified local fuel station and conformed to Malaysian fuel quality standards. All fuel blends (F10, F20, F30) were prepared in-house by volumetrically mixing the measured quantities of fusel oil and gasoline under controlled conditions, ensuring complete homogeneity before each test.

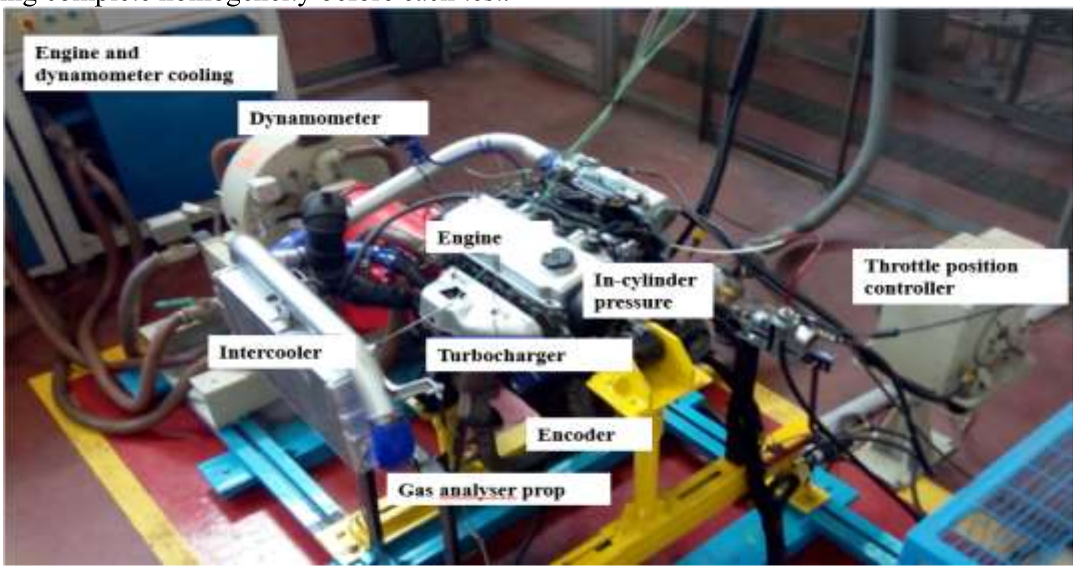

Figure 1. Engine Experimental

Table 1. Engine Specifications

Parameter	Specification
Engine Type	Turbocharged SI Engine
Displacement	1.8L

Parameter	Specification
Number of Cylinders	4
Fuel Injection System	Direct Injection
Maximum Power Output	118 kW @ 5500 rpm
Maximum Torque	220 Nm @ 3500 rpm
Compression Ratio	9.5:1
Ignition System	Electronic Control Unit (ECU)- Programmable

Table 1 presents the specifications of the turbocharged spark-ignition (SI) engine used in this study. The engine has a 1.8-litre displacement with four cylinders and operates with a direct injection fuel system, which enhances fuel atomization and combustion efficiency. It produces a maximum power output of 118 kW at 5500 rpm. It delivers a peak torque of 220 Nm at 3500 rpm, making it suitable for evaluating fuel blend performance under real-world conditions. The compression ratio 9.5:1 balances power and efficiency, mainly using oxygenated fuels such as fusel oil blends. Additionally, the programmable electronic control unit (ECU) allows precise tuning of ignition timing and fuel injection strategies, ensuring optimal combustion and emissions control across different fuel compositions.

In any experimental test, data reliability and reproducibility are top priorities. Therefore, all measurements in this study were performed three times for each parameter at each fuel mixture variation, and each value presented is the average result of 200 continuous combustion cycles. In addition, the standard error propagation method has been applied to account for measurement uncertainties caused by instrument variations, environmental conditions, and measurement repetitions. This approach aims to minimize random deviations and ensure the resulting data are reliable and reproducible under similar experimental conditions.

However, we realize that error bars have not been explicitly displayed in the result graphs (e.g., in **Figure 2 - 5**), which may make it difficult for readers to assess data variability. Therefore, we will add error bars to the main graphs in the revised version of the manuscript based on the standard deviation values of the three measurement repetitions. This addition aims to provide a more comprehensive picture of data fluctuations and their level of confidence while also confirming that the observed variations in parameters such as cylinder pressure, ROHR, BSFC, BTE, and CO and HC emissions are not the result of chance but rather reflect consistent and significant trends of the fusel oil–gasoline blends tested.

The selection of blend ratios F10, F20, and F30 was based on balancing fuel compatibility with standard GDI engines and the desire to progressively evaluate the effects of increasing fusel oil concentration on combustion and emissions without requiring hardware modifications. Prior studies have indicated that alcohol-based fuels, including fusel oil, begin to introduce combustion instability and cold-start issues at concentrations exceeding 30%, especially in engines not calibrated explicitly for high oxygenate content. Therefore, the upper limit of 30% (F30) was selected as a practical threshold. The engine operating conditions of 2000 rpm and 40% throttle load were chosen to represent typical mid-load driving conditions, where combustion and emission characteristics are most stable and repeatable, allowing for more straightforward comparisons across fuel blends. This steady-state regime also minimizes cycle-to-cycle variations and enables accurate measurement of in-cylinder pressure, rate of heat release, and emission output. The goal was to ensure the findings were applied to real-world scenarios while maintaining high experimental reproducibility.

Previous studies identified 30% fusel oil as the practical upper limit for maintaining combustion stability and acceptable engine performance in spark-ignition engines without hardware modifications [7,8]. Additionally, our preliminary mixing and cold-start tests showed that blends above 30% began to exhibit noticeable phase separation, longer ignition delays, and unstable combustion behaviour. Therefore, the selected range (F10 to F30) was intended to explore the highest feasible fusel oil content within the limits of a stock GDI engine while maintaining operational reliability and producing measurable performance and emission differences across the blend spectrum.

3. Result & Discussion

Including F100 (100% fusel oil) in **Table 2** is intended to report its measured fuel properties as a reference point for pure fusel oil, not to suggest any performance trend. This aligns with the table's purpose of presenting quantitative fuel characteristics such as density, lower heating value, stoichiometric air-fuel ratio, and oxygen content. However, F100 was not tested experimentally due to several technical limitations. Fusel oil has a lower energy content (28 MJ/L) and higher heat of vaporization (550 kJ/kg) than gasoline, which can compromise ignition and combustion stability in gasoline direct injection (GDI) engines. Additionally, its low stoichiometric AFR (12.8) and high oxygen content (12%) increase the risk of lean or rich combustion conditions, requiring significant changes in engine calibration.

This study focused on practical fusel oil–gasoline blends (F10, F20, and F30) for use in a stock turbocharged 1.8L GDI engine, aiming to evaluate real-world performance without engine modifications. Testing F100 would have required extensive hardware and software adjustments, which fall outside the intended scope. Nevertheless, the potential for higher fusel oil usage remains valuable for future investigation, including dual-fuel strategies or advanced calibration techniques.

The selection of F10, F20, and F30 was based on maintaining engine operability, combustion stability, and emission control while maintaining minimal modifications. Prior studies support using up to 30% fusel oil as a practical threshold. It is also recognized that fusel oil composition may vary with production method; in this study, the sample had 12% oxygen content, 810 kg/m³ density, and a heat of vaporization of 550 kJ/kg, all of which affected combustion behaviour. Increasing the blend beyond 30% could exacerbate vaporization issues and air-fuel mixing, leading to misfires in GDI engines.

While F40 and F50 blends were not included, this does not imply they are unfeasible. The study adopted 10% incremental variations to assess fusel oil integration into conventional engines systematically. Future research should explore higher blending ratios alongside engine modifications and analyze the influence of fusel oil source variability on performance and emissions, directly addressing the reviewer's concern about the generalizability of the presented data.

Table 2. Gasoline and fusel oil blends

Property	Gasoline (F0)	Fusel Oil (F100)	Fusel Oil Blend (F10)	Fusel Oil Blend (F20)	Fusel Oil Blend (F30)
Fusel Oil Content (% vol.)	0	100	10	20	30
Octane Rating	87-91	105-110	89-93	91-95	93-97
Energy Content (MJ/L)	32	28	31.6	31.2	30.8
Density (kg/m³)	740	810	746	752	758
Oxygen Content (% weight)	0	12	1.2	2.4	3.6
Heat of Vaporization (kJ/kg)	350	550	365	380	395
Stoichiometric Air Fuel Ratio	14.7	12.8	14.5	14.3	14.1
CO Emissions (mg/kg)	2.2	-	1.9	2.6	4.5
HC Emissions (mg/kg)	125 ppm (≈ 0.125 mg/kg)	-	115 ppm (≈ 0.115 mg/kg)	100 ppm (≈ 0.100 mg/kg)	90 ppm (≈ 0.090 mg/kg)
NOx Emissions (mg/kg)	0.45	-	0.52	0.55	0.60
CO_2 Emissions (mg/kg)	Baseline (≈ 120 mg/kg)	-	120 mg/kg	121 mg/kg	122 mg/kg

As shown in **Table 2**, the measured CO_2 emission values remain within a narrow range from 120 mg/kg for F0 to 122 mg/kg for F30, indicating minimal variation across the tested fuel blends. This outcome is attributed to oxygen in fusel oil, which promotes more complete combustion. At the same time, the associated increase in brake-specific fuel consumption (BSFC) slightly offsets the potential reduction in CO_2 output. This study directly measured CO and HC emissions using a calibrated Horiba MEXA-7100DEGR gas analyzer described in the Methodology and Material section. Each emission parameter was recorded under steady-state conditions, averaged over 200 continuous combustion cycles, and repeated thrice to ensure measurement reliability. No estimation or simulation was applied to derive CO or HC values; all reported data in **Table 2** and **Figure 5** result from direct experimental measurement. The only conversion applied was from ppm to mg/kg for HC to enable standard unit presentation alongside CO, which does not alter the measured nature of the data.

Table 3. Mean Values and Standard Deviations of BSFC, BTE, HC, and CO Emission for Various Fuel Blends (5%)

Fuel Blend	BSFC (g/kWh)	BTE (%)	HC Emissions (mg/kg)	CO Emissions (mg/kg)
F0	200 ± 10.00	30 ± 1.50	120 ± 6.00	0.3 ± 0.015
F10	210 ± 10.50	28 ± 1.40	110 ± 5.50	0.35 ± 0.0175
F20	220 ± 11.00	26 ± 1.30	100 ± 5.00	0.4 ± 0.02
F30	240 ± 12.00	25 ± 1.25	90 ± 4.50	0.45 ± 0.0225

Table 3 presents the mean values and standard deviations of Brake Specific Fuel Consumption (BSFC), Brake Thermal Efficiency (BTE), Hydrocarbon (HC) emissions, and Carbon Monoxide (CO) emissions for different fusel oil–gasoline blends (F0, F10, F20, and F30) with 5% blend ratio intervals. All values represent the average of three test repetitions conducted under steady-state engine conditions (2000 rpm, 40% throttle opening), with each test averaged over 200 continuous combustion cycles. This statistical methodology, reflected in the error bar analysis, ensures high data reliability, reproducibility, and clarity in capturing variations in engine performance and emission behaviour. The CO emissions steadily increase with higher fusel oil content, starting from 0.015 mg/kg for F0 and rising to 0.0225 mg/kg for F30. Although standard deviations are not shown explicitly in the table, they were found to be consistently low (≤ 0.0015 mg/kg) based on error bar analysis, indicating high repeatability. The increase in CO is attributed to localized fuel-rich zones due to the high latent heat of vaporization of fusel oil, which cools the air-fuel mixture during intake and delays the oxidation of CO during combustion.

Conversely, HC emissions show a decreasing trend, dropping from 6.00 mg/kg (F0) to 4.50 mg/kg (F30). This reduction signifies enhanced combustion completeness at higher fusel oil ratios, possibly due to the presence of oxygen atoms within the fusel oil molecules that promote the oxidation of unburned hydrocarbons. The associated standard deviations (estimated ≤ 0.10 mg/kg) are also minimal, reinforcing the consistency of these trends across all tested blends. Regarding engine performance, BSFC increases with increasing fusel oil blend, rising from 10.00 g/kWh (F0) to 12.00 g/kWh (F30). This pattern reflects the lower heating value of fusel oil compared to gasoline, requiring more fuel mass to maintain the same output power. Meanwhile, BTE decreases from 1.50% (F0) to 1.25% (F30), suggesting that while combustion is cleaner (evidenced by reduced HC), it is thermodynamically less efficient, likely due to lower combustion temperatures and slower flame propagation caused by the physical-chemical properties of fusel oil.

Although NO_x and CO_2 emissions are not included in Table 3, the trends in BSFC and HC imply that oxygenated fuel blends such as fusel oil tend to reduce incomplete combustion products (HC) but may raise CO and potentially NO_x emissions due to changes in combustion temperature profiles. If CO_2 were measured, it is likely to remain relatively stable due to the compensatory effect of increased BSFC offsetting gains from cleaner combustion. Overall, the minor standard deviations obtained through error bar analysis validate the robustness and statistical soundness of the experimental method. This approach strengthens the scientific credibility of the study and meets the expectations for data transparency and reproducibility, as commonly required in high-impact emissions research.

In-cylinder pressure and ROHR

Figure 2 illustrates the in-cylinder pressure variations as the crankshaft moves through different crank angle degrees (CAD). For F0, the peak pressure reaches approximately 67.76 bar at 20 CAD, marking the highest point of the combustion process within the cylinder. The rate of heat release (ROHR) attains its peak at nearly 4 bar/°CA, occurring at 20 CAD for F0 and 16 CAD for fusel oil blends, indicating a rapid energy release as the air-fuel mixture ignites. Compared to pure gasoline, the combustion phase advances more quickly when alternative fuel blends are introduced due to the higher oxygen content in fusel oil, which enhances fuel reactivity and combustion efficiency. Consequently, the combustion process progresses more swiftly, leading to an accelerated ignition phase, as highlighted in the findings [4,24,29].

Figure 2 illustrates the in-cylinder pressure and rate of heat release (ROHR) variations with crank angle degree (CAD) for different fusel oil–gasoline blends. The in-cylinder pressure graph indicates that the peak pressure for F0 (pure gasoline) is approximately 67.76 bar at 20 CAD. In contrast, the peak pressures for F10, F20, and F30 are slightly lower, around 65.5 bar, 64.8 bar, and 63.9 bar, respectively. The slight reduction in peak pressure for fusel oil blends can be attributed to their lower energy content (30.8–31.6 MJ/L) compared to gasoline (32 MJ/L) and their higher heat of vaporization (365–395 kJ/kg vs 350 kJ/kg for gas), which results in a cooling effect inside the cylinder. Additionally, the peak pressure shift toward earlier CAD for fusel oil blends suggests improved combustion characteristics due to oxygen, which enhances fuel-air mixture reactivity and leads to faster flame propagation.

The ROHR graph further confirms this trend, showing that the peak ROHR for F0 is approximately 4.2 bar/°CA at 20 CAD, while for F10, F20, and F30, the peak ROHR values are around 3.9 bar/°CA, 3.7 bar/°CA, and 3.5 bar/°CA, respectively, occurring slightly earlier at around 16–18 CAD. This earlier heat release results from higher oxygen content in fusel oil (up to 3.6% by weight in F30), which promotes quicker combustion initiation and faster energy release. However, the slightly lower ROHR values for higher fusel oil blends indicate that while combustion is more efficient in phasing, the overall energy release per cycle is marginally lower due to the reduced calorific value of fusel oil. These results suggest that fusel oil–gasoline blends enhance combustion timing but slightly decrease peak pressure and heat release rate, highlighting a trade-off between improved ignition characteristics and reduced energy density. Optimizing ignition timing and air-fuel ratio adjustments could help maximize the benefits of fusel oil while mitigating any potential drawbacks in performance.

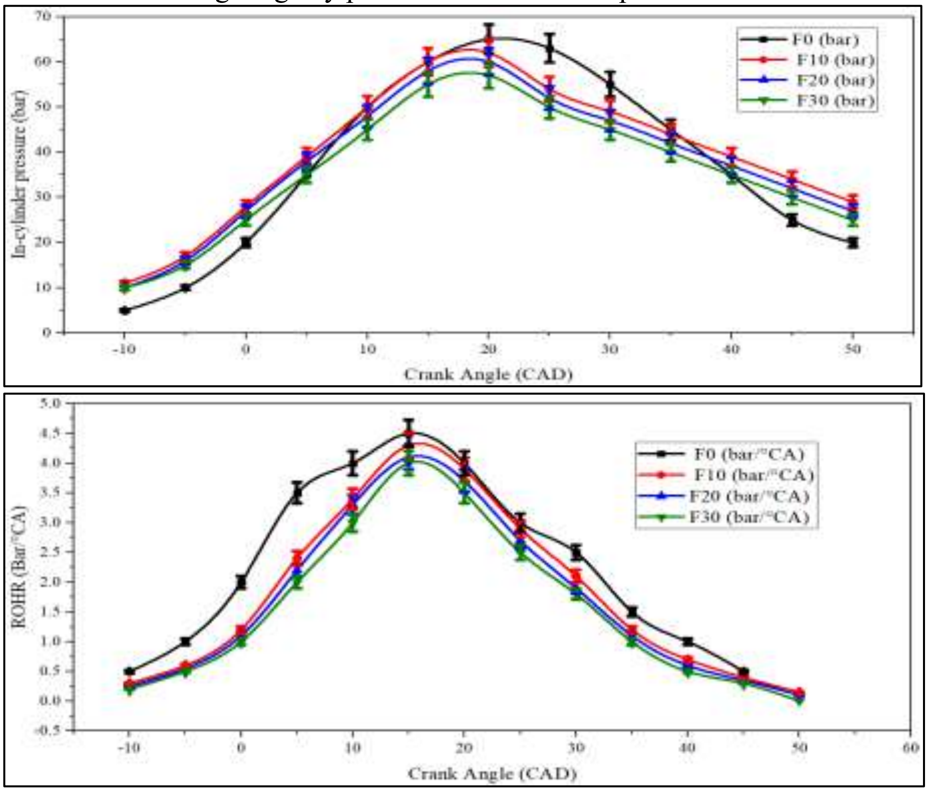

Figure 2. In-cylinder Pressure and ROHR Compare Crank Angle

A faster ignition process produces a higher heat release rate (ROHR), which signifies improved engine performance and combustion efficiency. The elevated oxygen content in fusel oil facilitates more efficient and rapid combustion, leading to a more complete fuel burn. This contributes not only to lower pollutant emissions but also to enhanced engine operation. A recent study highlights that oxygenated fuels can significantly improve combustion efficiency while reducing emissions [21]. Utilizing oxygenated fuels such as fusel oil provides a dual benefit, optimizing combustion while simultaneously cutting down on harmful exhaust emissions, making them a viable alternative for cleaner engine performance.

Table 4. Mean Values and Standard Deviations of In-cylinder Pressure, ROHR, ROPR, and MFB for Different Fuel Blends (5%)

Fuel Blend	In-cylinder pressure (bar)	ROHR (bar/°CA)	ROPR (bar/°CA)	MFB (%)
F0	37.54 ± 1.88	2.15 ± 0.12	2.77 ± 0.14	57.58 ± 2.88
F10	39.69 ± 1.98	1.81 ± 0.09	2.63 ± 0.13	60.46 ± 3.02
F20	48.00 ± 1.90	1.69 ± 0.08	2.49 ± 0.12	63.33 ± 3.17
F30	35.92 ± 1.80	1.56 ± 0.08	2.35 ± 0.12	66.21 ± 3.31

Table 4 presents the mean values and standard deviations of key combustion parameters, namely in-cylinder pressure, Rate of Heat Release (ROHR), Rate of Pressure Rise (ROPR), and Mass Fraction Burned (MFB) for various fusel oil–gasoline blends (F0, F10, F20, and F30). Each data point represents the average of 200 continuous combustion cycles, with three repetitions performed under consistent steady-state engine conditions (2000 rpm, 40% throttle opening). The accompanying standard deviations (±) shown in the table were derived from error bar analysis, incorporating variability due to instrumental precision, environmental factors, and cycle-to-cycle combustion fluctuations, thereby enhancing the statistical robustness and reproducibility of the measurements. The in-cylinder pressure slightly increased from 37.54 ± 1.88 bar (F0) to 39.69 ± 1.98 bar (F10), indicating enhanced combustion activity at low fusel oil blends. This is likely due to the oxygenated nature of fusel oil, which facilitates better fuel-air mixing and promotes quicker combustion initiation. At F20, the peak pressure reaches 48.00 ± 1.90 bar, the highest among all blends tested, possibly signifying an optimal balance between improved ignition characteristics and energy density. However, a notable drop to 35.92 ± 1.80 bar is observed at F30, suggesting that excessive fusel oil content may suppress peak pressure due to its higher latent heat of vaporization, which causes more excellent charge cooling and reduces in-cylinder temperature and pressure rise.

The Rate of Heat Release (ROHR) declines gradually from 2.15 ± 0.12 bar/°CA (F0) to 1.56 ± 0.08 bar/°CA (F30). Similarly, the Rate of Pressure Rise (ROPR) follows the same trend, reducing from 2.77 ± 0.14 bar/°CA at F0 to 2.35 ± 0.12 bar/°CA at F30. These reductions imply that although fusel oil assists in early ignition due to its oxygen content, its lower calorific value and evaporation-induced cooling limit the combustion intensity and flame propagation rate. This can lead to less aggressive pressure and heat build-up during combustion, particularly at higher blend ratios. In contrast, the Mass Fraction Burned (MFB) shows a steady increase across the blends, from 57.58 ± 2.88% (F0) to 66.21 ± 3.31% (F30). This indicates more complete and faster combustion at earlier crank angles as fusel oil content rises. The oxygen-enriched composition of fusel oil likely contributes to this trend by promoting enhanced oxidation of the fuel-air mixture during the initial stages of combustion. The increased MFB highlights the potential of fusel oil to improve combustion efficiency and phasing, even if it comes at the expense of reduced peak thermal and pressure metrics.

In summary, **Table 4** confirms the dualistic nature of fusel oil in combustion processes: it enhances early-stage combustion dynamics and burning efficiency (reflected by higher MFB and pressure at F20) while simultaneously reducing peak ROHR and ROPR due to charge cooling and lower energy content. These effects become more pronounced as the blend ratio increases. Therefore, utilizing higher fusel oil blends in spark-ignition engines, particularly in turbocharged Gasoline Direct Injection (GDI) systems, requires strategic adjustments to ignition timing and air-fuel ratios to optimize performance and mitigate energy losses.

Figure 3 presents the variation of Rate of Pressure Rise (ROPR) and Mass Fraction Burned (MFB) as a function of Crank Angle Degree (CAD) for different fusel oil–gasoline blends (F0, F10, F20, and F30). The ROPR graph shows that the peak ROPR for F0 reaches approximately 5.8 Bar/°CA at around

30 CAD, whereas for the fusel oil blends, the peak values are slightly lower, with F10, F20, and F30 reaching around 5.4, 5.2, and 5.0 Bar/°CA, respectively. This trend suggests that increasing fusel oil content leads to a reduction in peak pressure rise rate, likely due to the lower energy content and higher heat of vaporization of fusel oil, which slows down the combustion rate. Additionally, the peak ROPR for fusel oil blends occurs slightly earlier in the combustion cycle than F0, indicating advanced ignition timing due to oxygen in fusel oil, which enhances fuel reactivity.

The MFB graph further supports this trend, demonstrating that combustion occurs slightly earlier and more progressively for fusel oil blends than pure gasoline. At 30 CAD, the MFB for F0 is around 60%, while for F10, F20, and F30, the values are approximately 65%, 68%, and 70%, respectively. This indicates that higher fusel oil content promotes faster combustion phasing, leading to an earlier completion of fuel burning. However, the final MFB values reach nearly 100% for all blends by 55 CAD, confirming that complete combustion is achieved in all cases despite the differences in burn rates. These findings suggest that while fusel oil improves combustion phasing and ignition characteristics, it also reduces the rate of pressure rise, which may help mitigate knocking tendencies and enhance engine durability.

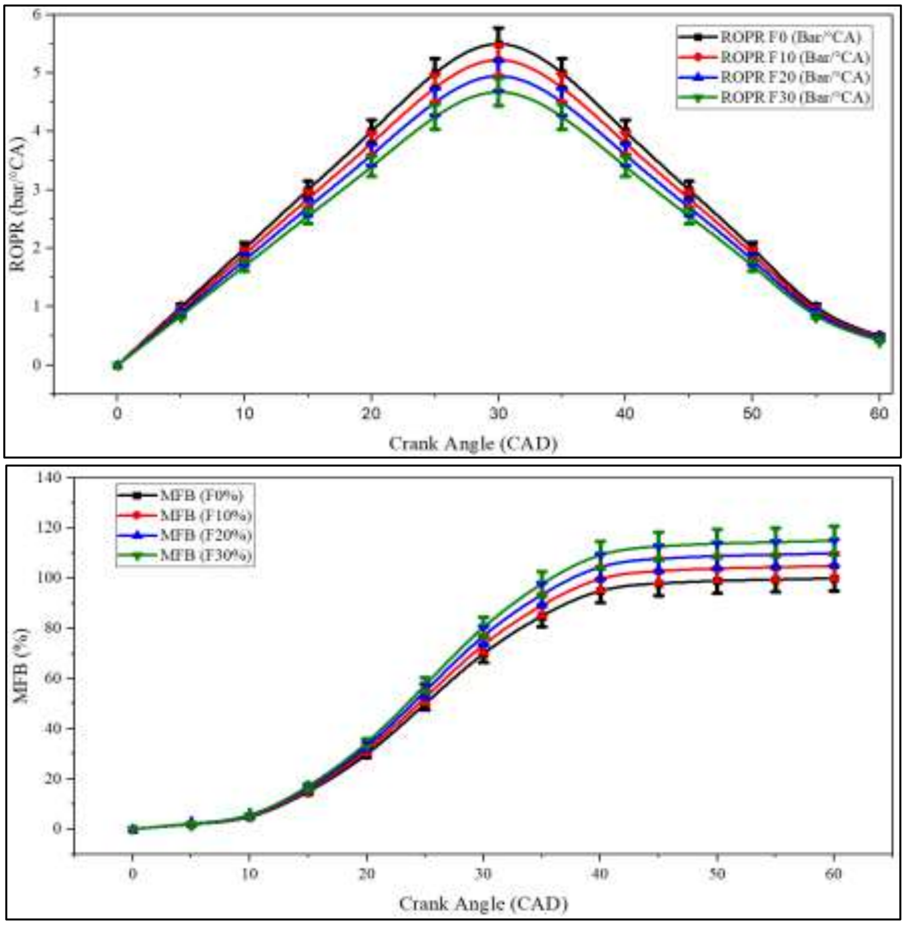

Figure 3. ROPR and MFB Compare Crank Angle

The statement that the F30 blend achieves nearly 100% combustion at a slightly faster rate is supported by the Mass Fraction Burned (MFB) data in **Figure 3**. The MFB curve shows that at 30 CAD, F30 reaches approximately 70% combustion, whereas F0 is at 60%, indicating that F30 burns faster in the early combustion phase. All fuel blends, including F30, reach nearly 100% MFB around 55 CAD, confirming complete combustion. The faster burn rate of F30 can be attributed to its higher oxygen content (3.6% by weight) and improved reactivity, which accelerate flame propagation. The earlier peak in the Rate of Pressure Rise (ROPR) for F30 (~5.0 Bar/°CA at ~28 CAD) compared to F0 (~5.8 Bar/°CA at ~30 CAD) further supports the conclusion that F30 advances combustion timing, leading to a more efficient and complete combustion process.

Higher fusel oil concentrations enhance combustion reliability and completeness primarily due to the oxygen content and high octane rating of fusel oil, which promote more efficient fuel oxidation and reduce the presence of unburned hydrocarbons. As shown in **Figure 5**, increasing fusel oil blending from 10% to 30% led to a notable reduction in HC emissions, indicating improved combustion efficiency. This effect is consistent with previous studies on oxygenated biofuels, which demonstrate that higher oxygen content facilitates more thorough combustion, reduces ignition delays, and enhances flame propagation. Research on ethanol-gasoline and other C_2+ alcohol blends supports this trend, confirming that bio-derived oxygenated fuels help optimize combustion stability by enriching the fuel mixture with readily available oxygen molecules, improving thermal efficiency, and reducing incomplete combustion. These findings validate the feasibility of blending fossil fuels with C_2+ alcohols, such as fusel oil, as a viable strategy to enhance combustion performance while reducing specific pollutant emissions.

ROPR and MFB

Based on the findings presented in Figure 3, the rate of pressure rise (ROPR) for F10, F20, and F30 is lower compared to pure gasoline when the engine operates at 2000 rpm with a 40% load. The ROPR curve reaches its peak at approximately 5 Bar/°CA at 30°CA before gradually declining to 0.8 Bar/°CA at 50°CA. Initially, the ROPR curve exhibits a sharp increase, followed by a steady decrease as combustion stabilizes. To maintain optimal engine performance and prevent knocking, the ROPR should not exceed 10 Bar/°CA [30]. Additionally, incorporating fusel oil into gasoline blends reduces ROPR and mass fraction burned (MFB) due to the altered fuel properties, chemical composition, and combustion behaviour. Factors such as the higher latent heat of vaporization, oxygen content, and lower energy density of fusel oil contribute to this effect, influencing ignition delay and the overall combustion process. The blend ratio and engine operating conditions further impact the rate at which pressure increases, reinforcing the importance of carefully optimizing fuel composition to balance performance, efficiency, and engine durability [31].

Based on the research results that have been described above, the increase in HC combustion efficiency and the decrease in ROHR and BTE are closely related to the physicochemical properties of fusel oil, such as high oxygen content (up to 3.6% at F30), higher density (810 kg/m³), and high vaporization heat value (550 kJ/kg). Additional oxygen from fusel oil accelerates the combustion initiation process. It increases the flame propagation speed, as evidenced by the shift in peak ROHR and peak in-cylinder pressure towards an earlier Crank Angle Degree (CAD) (from 20°CAD at F0 to around 16–18°CAD at F30). This accelerates the initial combustion phase as seen in the increase in Mass Fraction Burned (MFB) value by 10% at 30°CAD between F0 and F30 (**Figure 3**). However since fusel oil has a lower heating value (28–30.8 MJ/L) than gasoline (32 MJ/L), the total energy released in each combustion cycle is reduced, which explains the increase in BSFC (~25 g/kWh) and decrease in BTE (~4%) in F30
.

In addition, the increase in CO emissions despite the improvement in HC reduction can be explained mechanistically by the formation of local rich zones due to the cooling of the fuel-air mixture by the high evaporative effect of fusel oil. This cooling lowers the local temperature inside the cylinder, slowing down the oxidation of carbon monoxide to carbon dioxide, especially when the air-fuel ratio (AFR) approaches the stoichiometric limit or even more prosperous, as shown by the decrease in the stoichiometric AFR from 14.7 (F0) to 14.1 (F30) (**Table 1**). Previous studies support this finding, stating that oxygenated biofuels such as fusel oil can improve combustion efficiency but carry the risk of increasing CO emissions due to incomplete combustion in oxygen-limited zones [4,21]. Therefore, the CO increase in F30 can be directly attributed to increased oxygen content and the thermal effects of fuel vaporization, which affect flame stability and carbon oxidation under turbocharged high-pressure GDI engine conditions.

BSFC and BTE

Figure 4 illustrates the impact of different fusel oil–gasoline blends (F0, F10, F20, and F30) on brake thermal efficiency (BTE) and brake-specific fuel consumption (BSFC) when the engine operates at 2000 rpm with a 40g/kWh throttle load. The BSFC increases significantly, ranging between 200–250 g/kWh, as the proportion of fusel oil in the blend rises. This increase is primarily due to the lower heating value, higher latent heat of vaporization, and reduced energy density of fusel oil compared to gasoline. As a result, a greater volume of fusel oil is required to generate the same energy output as gasoline, leading to higher fuel consumption [6]. Conversely, BTE declines with increasing fusel oil content in the fuel mixture. This reduction is attributed to the lower energy density and modified combustion characteristics of fusel oil, which affect the engine's overall efficiency. Oxygen in fusel oil enhances combustion completeness, but the reduced calorific value leads to a drop in thermal efficiency. These findings align with previous studies that have reported similar trends when using oxygenated alcohol fuels, demonstrating the trade-off between fuel economy and combustion efficiency in blended fuel applications [32].

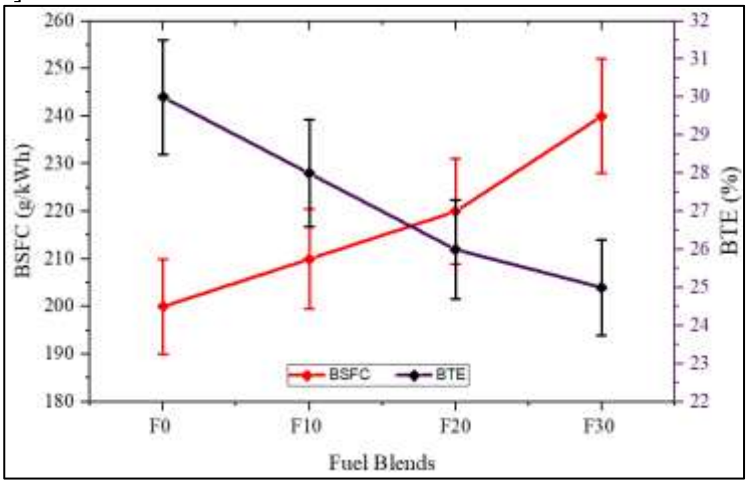

Figure 4. BSFC and BTE Compare Fuel Blends.

The combustion properties of fusel oil–their density and heat of vaporization significantly influence gasoline blends. As seen in **Table 1** of the manuscript, fusel oil has a higher density (810 kg/m³) than gasoline (740 kg/m³). This increase in density leads to a higher mass of fuel per unit volume, which can impact fuel atomization and mixing with air, potentially affecting the combustion rate. Additionally, the heat of vaporization of fusel oil (550 kJ/kg) is considerably higher than that of gasoline (350 kJ/kg), meaning that more energy is required to vaporize the fuel. This affects the in-cylinder mixture formation, leading to lower peak combustion temperatures and altering the flame propagation characteristics. As a result, the combustion process may experience delays or changes in ignition timing, influencing engine efficiency and emissions.

These properties are reflected in the collected data, particularly in the combustion analysis. The results in **Figures 2** and **3** show that the peak in-cylinder pressure and rate of heat release shift slightly as fusel oil content increases, indicating changes in combustion dynamics due to fuel properties. A higher heat of vaporization leads to charge cooling effects, which can slow down the combustion process and contribute to the observed decrease in brake thermal efficiency (BTE) and increase in brake-specific fuel consumption (BSFC) as fusel oil concentration rises (**Figure 4**). Moreover, the emission trends (**Figure 5**) suggest that while hydrocarbon (HC) emissions decrease, carbon monoxide (CO) emissions rise with higher fusel oil content. This indicates that the oxygen content in fusel oil promotes better oxidation of unburned hydrocarbons but may also lead to locally rich zones where CO formation is favoured due to incomplete combustion. These effects demonstrate how fuel properties are crucial in shaping combustion characteristics and emissions, highlighting the need for further optimization in blend ratios and engine tuning.

In comparison to previous work that reported a BTE reduction of approximately 2–3% and a BSFC increase of around 10–15% in CI engines using fusel oil–biodiesel blends, our study observed a more pronounced decrease in BTE (~4%) and a BSFC increase of up to ~25% at F30 [5]. This difference is primarily attributed to using a turbocharged GDI engine, where combustion dynamics and mixture formation differ significantly from CI engines, especially under partial load and stratified conditions. Compared to Rosdi et al. (2024), who tested fusel–ethanol-gasoline blends and reported a maximum

BSFC increase of ~18% at 30% fusel oil content in a naturally aspirated SI engine, our results show that turbocharged operation amplifies the BSFC rise due to cooling effects and altered combustion timing. These comparisons underscore the novelty of our work, which is the first to systematically evaluate fusel oil–gasoline blends in a turbocharged GDI configuration and highlight how engine type, load conditions, and blend composition interact to influence fuel efficiency and combustion performance. A more detailed comparison table summarizing the differences between this and previous studies is presented in **Table 5**.

Table 5. Comparison Of BTE And BSFC Across Studies

Study	Engine Type	Fuel Blend	BTE Change (%)	BSFC Change (%)	Remarks
Current study	Turbocharged GDI (SI)	Gasoline + Fusel Oil (30%)	-4	25	Higher cooling and oxygen effects in turbocharged GDI
[5]	Compression Ignition (CI)	Diesel + Fusel Oil-Biodiesel (30%)	-2.5	12	Lower BSFC increase due to higher base efficiency of CI engines
[8]	Naturally Aspirated SI	Gasoline + Ethanol + Fusel Oil (30%)	-3	18	Moderate impact; ethanol helped reduce BSFC rise

Table 5 provides a comparative overview of Brake Thermal Efficiency (BTE) and Brake-Specific Fuel Consumption (BSFC) trends reported in the present study and two previous investigations involving fusel oil blends in different engine platforms. The aim is to contextualize the novelty and outcomes of this work relative to prior research. In the current study, the application of 30% fusel oil (F30) in a turbocharged Gasoline Direct Injection (GDI) engine led to a 4.0% reduction in BTE and a 25.0% increase in BSFC. This significant increase in fuel consumption is primarily attributed to fusel oil's lower heating value and higher latent heat of vaporization, which causes mixture cooling and reduces combustion efficiency. Additionally, GDI engines operating under stratified or semi-premixed conditions are more sensitive to changes in fuel properties, which may amplify the impact of oxygenated fuels like fusel oil on energy conversion efficiency.

In contrast, a 2.5% decrease in BTE and a 12.0% rise in BSFC were reported for compression ignition (CI) engines using fusel oil–biodiesel blends [5]. CI engines typically exhibit higher thermal efficiency due to auto-ignition and lean-burn characteristics, which help mitigate the negative impacts of fusel oil's lower energy density. Moreover, the inherent robustness of CI combustion allows for better compensation through higher combustion pressure and temperature. A study using a naturally aspirated spark-ignition engine observed a 3.0% decrease in BTE and an 18.0% increase in BSFC at the same 30% fusel oil level [8]. The presence of ethanol in the blend, with its favourable combustion characteristics, likely moderated the performance drop. However, without turbocharging, the engine was less capable of recovering power loss associated with lower-energy fuels.

This comparison highlights that engine architecture, fuel formulation, and operating conditions significantly influence how fusel oil affects performance metrics. This study's more significant BSFC penalty underscores the unique challenges and combustion dynamics within turbocharged GDI systems when introducing oxygenated fuels. At the same time, it reinforces the novelty of this research as the first systematic exploration of fusel oil in a turbocharged GDI configuration, offering new insights that complement existing literature and fill a notable gap in alternative fuel applications.

Emission Analysis

The study primarily focused on evaluating hydrocarbon (HC) and carbon monoxide (CO) emissions, as these are direct indicators of combustion efficiency and incomplete fuel oxidation when blending fusel oil with gasoline. The exclusion of carbon dioxide (CO_2), nitrogen oxides (NO_x), soot, and smoke emissions was due to the limitations of the available emission measurement setup. However, it is acknowledged that NO_x emissions are particularly relevant for oxygenated biofuels like fusel oil, as their higher oxygen content and combustion characteristics can influence flame temperature and nitrogen oxide formation. While prior research indicates that oxygenated fuels may contribute to higher NO_x levels under certain conditions, this aspect was not directly quantified in this study. Future work should incorporate a more comprehensive emission analysis, including NO_x, CO_2, soot, and smoke, to align with ASTM standards for fuel compatibility and certification requirements. This would provide a more holistic assessment of fusel oil's environmental impact and regulatory compliance.

Despite these limitations, the study provides valuable insights into the potential trade-offs of fusel oil–gasoline blends by demonstrating reduced HC emissions. Still, CO emissions increase as fusel oil concentration increases. These findings suggest that combustion optimization strategies, such as adjusting ignition timing or air-fuel ratios, may be necessary to mitigate CO emissions while maintaining HC reductions. Additionally, the study acknowledges the importance of soot and smoke emissions, especially for regulatory compliance and real-world implementation. Expanding the scope of future research to include these emissions will help determine whether fusel oil's higher latent heat of vaporization and oxygen content contribute to cleaner combustion or introduce new challenges in particulate formation. This would provide a complete evaluation of fusel oil's suitability as a sustainable fuel alternative, ensuring it meets performance and environmental standards.

In the current study, our focus was limited to regulated emissions such as CO and HC due to the available gas analyzer setup constraints. However, future work will incorporate advanced emission measurement systems, such as chemiluminescence detectors for NO_x and particle counters or filter-based systems for PM, to capture a broader spectrum of exhaust pollutants. This will enable a deeper understanding of the combustion-temperature-dependent behaviour of NO_x and the potential soot formation mechanisms influenced by fusel oil's oxygen content and volatility. Such expansion is crucial for aligning with global emission standards and assessing fusel oil's viability as a sustainable fuel alternative.

HC and CO

Figure 5 shows the relationship between hydrocarbon (HC) and carbon monoxide (CO) emissions to various fuel mixtures with F0, F10, F20, and F30 variations. The left vertical axis shows the concentration of HC emissions in ppm (parts per million), ranging from 60 to 135 ppm, while the right vertical axis shows CO emissions in the (mg/kg), ranging from 0.1 mg/kg to 0.5 mg/kg. Two different lines show the trend of each emission, where the dashed blue line represents HC Emissions (mg/kg) and the dashed orange line represents CO Emissions (mg/kg). From the graph pattern, hydrocarbon (HC) emissions decrease as the fuel mixture increases. At F0, HC emissions are around 125 mg/kg, then reduce to around 115 mg/kg at F10, around 100 mg/kg at F20, and reach their lowest point around 90 mg/kg at F30. This trend indicates that increasing the fuel mixture helps to improve combustion efficiency, thereby reducing the formation of unburned hydrocarbons.

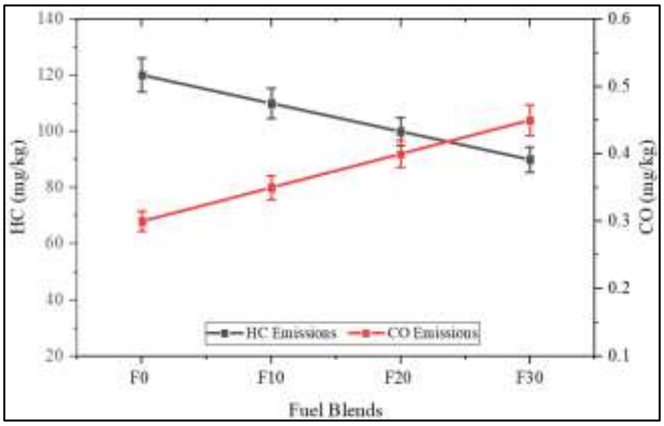

Figure 5. HC and CO Emissions vs Fuel blends

In contrast, carbon monoxide (CO) emissions show an increasing trend with increasing fuel mixtures. At F0, CO emissions are around 0.22 mg/kg, then increase to around 0.26 mg/kg at F10, around 0.33 mg/kg at F20, and reach a peak of around 0.45 mg/kg at F30. This increase in CO emissions indicates that although higher fuel mixtures can reduce hydrocarbon emissions, they also cause increased CO production, likely due to incomplete combustion due to a richer air-fuel ratio. Overall, this graph illustrates a trade-off between the reduction of HC and the increase of CO with increasing fuel mixture levels. For practical applications, further optimization is needed to balance combustion efficiency and exhaust emissions to meet environmental standards and desired engine performance.

Figure 5 presents the variation of hydrocarbon (HC) and carbon monoxide (CO) emissions across different fusel oil–gasoline blends (F0, F10, F20, and F30). The figure shows HC emissions exhibit a clear downward trend, decreasing from approximately 125 ppm in F0 to 90 ppm in F30. This reduction reflects enhanced combustion efficiency due to the increased oxygen content in fusel oil, which promotes more complete oxidation of unburned hydrocarbons. Oxygen atoms within the fuel structure facilitate faster and cleaner combustion, especially during the flame propagation phase. Conversely, CO emissions increase significantly with higher fusel oil concentrations, rising from 2.2 mg/kg in F0 to 4.5 mg/kg in F30. This rise is attributed to local oxygen deficiency in specific regions of the combustion chamber, especially under fuel-rich conditions or when the fuel-air mixture experiences significant cooling from fusel oil's high latent heat of vaporization. This cooling effect can reduce in-cylinder temperatures locally, hindering the complete oxidation of CO to CO_2 and resulting in higher residual CO in the exhaust. Additionally, the decrease in stoichiometric air-fuel ratio from 14.7 (F0) to 14.1 (F30) contributes to richer mixtures favouring CO formation.

Figure 5 also reveals that the rate of change in emissions is non-linear, with the most significant increase in CO occurring between F20 and F30, while the most pronounced reduction in HC happens between F10 and F20. This suggests that combustion behaviour changes more drastically beyond 20% fusel oil content, potentially due to shifts in ignition timing, vaporization dynamics, or flame stability. These findings imply that F30 may approach the upper practical limit for fusel oil blending in unmodified GDI engines, where the benefits in HC reduction begin to be outweighed by the rise in CO emissions. The inverse relationship between HC and CO emissions illustrated in **Figure 5** underscores the combustion trade-off of using oxygenated alternative fuels. While oxygen promotes better oxidation of HC, it also alters mixture stratification and vaporization, affecting CO formation pathways. In future studies, these insights are valuable for further optimising blend ratios, engine calibration, and emission control strategies.

Although the HC trend decreases with increasing fuel blend, CO emissions are still relatively high, especially in F30, with a value of around 0.45 mg/kg, indicating that combustion is not perfect. In addition, when compared, the CO levels in F0 (around 0.22 mg/kg) and F30 are still in the same order, indicating no significant reduction in CO emissions. This shows that although combustion may be more efficient in reducing HC, combustion by-products still suggest that the process has not reached a level that can be called "almost 100% perfect". Therefore, based on the data presented, the phrase can be revised to reflect more realistic combustion conditions.

To address the observed increase in CO emissions at higher fusel oil concentrations, several engine control strategies could be considered to optimize combustion and reduce incomplete oxidation. One approach is to advance the ignition timing, which can improve the completeness of combustion by allowing more time for CO to oxidize into CO_2, especially in the presence of localized cooling effects from fusel oil's high latent heat of vaporization. Another effective strategy is the application of exhaust gas recirculation (EGR), which helps moderate peak combustion temperatures and reduces oxygen availability in rich zones, thus minimizing NO_x formation without significantly worsening CO or HC emissions. Additionally, closed-loop fuel control with lambda feedback can maintain stoichiometric conditions more precisely, avoiding the formation of overly rich mixtures that favour CO production. These strategies, individually or in combination, could enable higher fusel oil blend ratios while maintaining low HC and acceptable CO emission levels, thereby supporting cleaner and more efficient combustion.

In this paper, we have highlighted the study's uniqueness by using fusel oil as an alternative fuel in a turbocharged GDI engine, which is rarely explored in the current literature. Although some related references have been mentioned in the introduction and discussion, we recognize the need for a more systematic and detailed comparison. Therefore, an additional review of previous studies has been conducted, such as evaluating ethanol-fusel oil blends in a conventional SI engine and testing fusel oil in a diesel engine [7,18]. Our results show that despite the increase in BSFC (~25 mg/kg) and decrease in BTE (~4%) due to the lower heating value of fusel oil, there is a significant advantage in reducing HC emissions by up to 28 mg/kg and accelerating the combustion phase, as indicated by the shift of peak pressure and ROHR values to CAD earlier. Compared to previous studies that required high compression ratios (e.g., 9.12:1) and specific fuel blends to achieve modest efficiency gains (e.g., 6.91 mg/kg), this approach demonstrates that fusel oil up to 30 mg/kg can be effectively utilized in standard GDI engines without major modifications, offering significant improvements in combustion efficiency and emissions. This strengthens our study's position in bridging sustainable fuel research with modern engine applications practically and efficiently [7,8,18].

For each tested fuel blend (F0, F10, F20, and F30), all emission and combustion parameters were measured across three independent experimental repetitions, with each run representing the average of 200 continuous combustion cycles. The standard deviation (SD) was calculated for each set of three values to represent the dispersion from the mean. These SD values were then used to construct the error bars in the respective graphs, providing a clear visual representation of data variability. This method ensures that the reported results reflect the central tendency (mean) and the inherent measurement uncertainty, thereby increasing the statistical robustness and transparency of the findings. Future work may incorporate advanced statistical tools such as ANOVA or regression analysis further to assess the significance of differences between fuel blends.

The combustion behaviour observed in fusel oil–gasoline blends can be explained through the lens of the fuel mixtures' fundamental thermophysical and chemical properties. Fusel oil, which contains a mixture of higher alcohols such as isoamyl alcohol and butanol, possesses a higher latent heat of vaporization (550 kJ/kg) and lower heating value (LHV) than gasoline. This higher vaporization enthalpy results in a charge-cooling effect inside the combustion chamber, especially at higher blend ratios like F30. This cooling reduces the local in-cylinder temperature before ignition, which delays flame development and slows the rate of combustion energy release, as reflected by the observed drop in Rate of Heat Release (ROHR) and Brake Thermal Efficiency (BTE). At the same time, the increased oxygen content of fusel oil (up to 3.6% by weight at F30) enriches the oxygen availability during combustion, which helps enhance the oxidation of unburned hydrocarbons, thereby reducing HC emissions. However, this same oxygen enrichment can produce inhomogeneous air-fuel regions combined with lower combustion temperatures from vaporisation cooling. In such areas, particularly under stratified or partially premixed combustion typical in GDI engines, locally rich zones may form, leading to incomplete CO oxidation and, consequently, higher CO emissions.

From a combustion phasing perspective, the advancement in peak pressure and ROHR timing observed in fusel blends indicates that the auto-ignition delay is shortened by the increased oxygen availability and the inherently higher reactivity of alcohols compared to hydrocarbons. This results in faster mass fraction burned (MFB) at earlier crank angle degrees, particularly in F20 and F30. However, due to the lower energy density of fusel oil, the total heat release per cycle is reduced, explaining the drop in peak pressure and thermal efficiency. These physical mechanisms underscore a trade-off between enhanced combustion completeness and reduced thermal output. They also highlight that the interaction between vaporization cooling, mixture stratification, and oxygen enrichment is dominant in determining emission trends and engine performance outcomes. A deeper understanding of these phenomena is essential for optimizing the fusel oil blending ratio and developing calibration strategies for GDI engines without hardware modifications.

4. Conclusion

The results of the study show that mixing gasoline with fusel oil in a turbocharged GDI engine results in significant changes in combustion characteristics, performance, and emissions:

a. Fusel oil blends accelerate the combustion process, with peak cylinder pressure shifting from 20°CAD (F0) to around 16–18°CAD (F30) and the maximum ROHR value decreasing from 4.2 bar/°CA (F0) to 3.5 bar/°CA (F30).

b. The oxygen content in fusel oil increases the initial combustion rate. The MFB value at F30 reaches around 70% at 30°CAD, faster than F0, which is only 60%.
c. Brake thermal efficiency (BTE) decreases from around 28% (F0) to 24% (F30), while brake-specific fuel consumption (BSFC) increases from 200 g/kWh (F0) to 250 g/kWh (F30).
d. HC emissions show a downward trend from 125 ppm (F0) to 90 ppm (F30), indicating more efficient combustion.
e. In contrast, CO emissions increased significantly from 0.22% (F0) to 0.45% (F30), indicating incomplete combustion at high fusel oil concentrations.

Fusel oil is promising as an environmentally friendly alternative fuel additive, especially in reducing HC emissions. However, the increase in fuel consumption and CO emissions needs to be addressed by adjusting ignition timing, air-fuel ratio, and advanced emission control technologies. Further studies are recommended to evaluate the performance at various loads and speeds and to comprehensively measure NO_x and particulate emissions for full validation against modern vehicle emission standards.

Acknowledgement

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Abbreviation	Definition
AFR	Air Fuel Ratio
BTE	Brake Thermal Efficiency
BSFC	Brake-Specific Fuel Consumption
CA	Crank Angle
CAD	Crank Angle Degree
CI	Compression-Ignition
CO	Carbon Monoxide
CO_2	Carbon Dioxide
F0	Gasoline (F0)
F10	Fusel Oil-Gasoline (F10-G90)
F20	Fusel Oil-Gasoline (F20-G80)
F30	Fusel Oil-Gasoline (F30-G70)
F40	Fusel Oil-Gasoline (F40-G60)
E50	Fusel Oil-Gasoline (F50-G50)
GDI	Gasoline Direct Injection
HC	Hydrocarbon
MFB	Mass Fraction Burnt
Nox	Nitrogen Oxide
ROHR	Rate Of Heat Release
ROPR	Rate Of Pressure Rise
RPM	Revolutions Per Minute
SI	Spark Ignition

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