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Techno-Economic and Environmental Assessment of an RCCI Diesel Engine Fuelled with Fusel Oil Addition

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Abstract

RCCI is a promising combustion strategy that can improve the controllability of combustion phasing. This study evaluates fusel oil (an inexpensive industrial by-product) as a low-reactivity supplementary fuel in an RCCI diesel engine. Fusel oil was injected into the intake air at 4, 6, 10, 12, and 16 g/min (DF4-DF16), and experiments were conducted on a four-cylinder, four-stroke diesel engine at 1750 rpm under 40, 60, 80, and 100 Nm. In-cylinder temperature/pressure-based combustion behaviour, air excess ratio (λ), NO and smoke emissions were assessed. The influence of fusel oil on combustion was strongest at low load. At 40 Nm, the highest fusel-oil energy share increased peak cylinder pressure by 14% and peak in-cylinder temperature by 4% compared to diesel fuel tests, while at 100 Nm the corresponding increases were 4% and less than 1%. NO increased at 40 Nm, with a maximum rise of 17.9% at the highest fusel-oil energy share, but decreased at medium and high loads, falling by 7.13 to 13.54% between 60 and 100 Nm. Smoke increased consistently with fusel oil, reaching about 42% at 40 Nm and remaining below 22% at 100 Nm. A techno-economic assessment showed that although capital costs increased slightly, the low price of fusel oil decreased operating costs by up to 33% and reduced life-cycle costs by up to 42%. Overall, fusel oil-assisted RCCI operation can provide notable cost benefits and conditional NO reductions, though with a smoke penalty that should be considered in application and after-treatment strategies.

Keywords: RCCI; fusel oil; diesel fuel; emissions; economic analysis; alternative fuels

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1. Introduction

Internal combustion engines are widely used in transportation, agriculture, and industrial applications, and significantly impact global energy and emissions [1–4]. However, it is well-known that these engines are heavily reliant on the utilisation of fossil-based fuels, with an approximate ratio of 84% [5]. Rising energy demands, limited fossil fuel reserves, and increasingly stringent exhaust emission regulations necessitate the development of innovative combustion strategies in engine technologies that deliver higher efficiency and lower emissions.

However, due to the disadvantages observed in HCCI engines, such as unstable operation at high loads, knocking, uncontrollable combustion, and a limited engine operating speed range [6,7] researchers have recently turned their attention to the RCCI combustion strategy.

In the RCCI strategy, low-reactivity fuel is either injected into the intake manifold or directly into the cylinder to create a homogeneous mixture, while high-reactivity fuel is injected directly into the cylinder to control ignition. This method provides enhanced precision in the control of the combustion phase, leading to a reduction in combustion temperatures and a substantial limitation on NOx and smoke formation [8–11].

Gozen [12] investigated the effects of different fuel mixture ratios, injection timings, and EGR on NOx and soot emissions in an RCCI diesel engine running on methanol and diesel fuels. The study emphasised that methanol has a key role in reducing NOx emissions. The author found that increasing the methanol and EGR ratios reduced NOx emissions but increased soot emissions.

Gharehghani et al. [13] investigated the combustion and performance parameters of an RCCI engine operating with natural gas and diesel fuel. The authors added ozone gas to the intake at various rates (10 ppm, 100 ppm, and 1000 ppm). In the study, the data obtained were used to validate the digital model. The engine speed was assumed to be 800 rpm, and the resulting findings were evaluated. The researchers noted that the addition of ozone gas accelerated the high-tech combustion process, and that with the expansion of the ozone gas amount, the RCCI process led to earlier combustion and increased combustion efficiency. In addition, they observed a decrease in CO and HC emissions and an increase in NOx emissions.

Isık [14] analysed the performance and emission characteristics of a biodiesel fueled CI engine by adopting the RCCI strategy. In the study, the main fuel was injected into the cylinders from the injectors of the fuel system on the engine towards the end of the compression stroke, while the secondary fuel was injected through the intake manifold. They stated that the LTC system is a reliable strategy for reducing exhaust emissions and improving fuel economy. The authors observed improvements in most of the combustion and performance parameters for ethanol and gasoline fueled mixtures in different ratios. They also reported that while fuel consumption increased at all loads, NOx emissions decreased significantly, and although there were partial increases, CO and HC emissions remained at low levels.

Biofuels produced from plant or animal sources are classified as viable alternative fuels to be used in CI engines, either in pure form or mixed with diesel fuel [15–18]. Their utilisation in RCCI applications offers significant advantages in terms of both environmental sustainability and energy supply security [19,20].

Altun et al. [21] investigated the use of biodiesel and diesel as high-reactivity fuels and methanol as a low-reactivity fuel in an RCCI engine. Methanol injection into the cylinder was compared with manifold injection, and direct injection was proposed as an alternative to the manifold injection approach commonly employed in RCCI engines. The effects of different low-reactivity fuel injection strategies on engine performance and emission characteristics were examined. The results showed that, compared to conventional diesel–biodiesel operation, the use of manifold injection in RCCI reduced NOx emissions but led to increases in CO and HC emissions.

Fusel oil, one of these biofuels, is a waste product in sugar factories [22,23]. For every 1000 L of ethyl alcohol produced after distillation, approximately 5 L of fusel oil are obtained. It is a promising alternative fuel due to its biological origin, high oxygen content, and its production as a byproduct of ethanol production processes [24–26].

Sriumpunpuk et al. [27] experimentally investigated the use of fusel oil as an alternative fuel in diesel engines in terms of combustion characteristics and exhaust emissions. Due to its alcohol-based structure and oxygen-containing components, fusel oil was tested in the engine by mixing it with diesel fuel at 10% and 20% by volume. The low reactivity and high heat of vaporisation of fusel oil contributed to a more widespread and controlled progression of the combustion phases; consequently, the rate of increase of in-cylinder pressure tended to decrease. However, it was noted that due to the low lower calorific value of fusel oil, more fuel is required in order to generate the same amount of power. Emission analysis showed that fuel blends containing fusel oil reduced CO and HC emissions, primarily due to the fuel's oxygen-containing structure and more homogeneous combustion environment. Conversely, it was stated that the fusel oil ratio has a low impact on NOx emissions.

Awad et al. [28], examined the performance and emission parameters for a fusel oil-diesel dual-fuel mixture at different engine speeds and engine loads. In the experiments, the engine speed increased in 300 rpm increments between 1200 rpm and 2400 rpm whilst the engine load varied between 50% to 75%. In studies conducted with mixed fuels, the presence of water in fusel oil caused a decrease in torque and power and an increase in specific fuel consumption due to its low calorific value and cetane number. The study concluded that water in the fuel mixture caused an increase in CO and CO_2 emissions and a decrease in NOx emissions.

Liu et al. [29], investigated the properties of fusel oil by analysing its water capability. Combustion and emission characteristics in a diesel engine were analysed according to three types of fuel (20% fusel oil with 0% water, 20% fusel oil with 6% water, and pure diesel). The results showed that moisture absorption from the air had no noticeable effect on the stability of the fuel, and the fuel remained single-phase. Adding 20% fusel oil with 6% water content increased brake thermal efficiency by approximately 0.7%. They also reported that emitted NOx by the water-containing fuel could be reduced by approximately 6.4%. In total, adding 20% fusel oil to diesel did not negatively affect the fuel.

Ciftci et al. [30], investigated the effects of fusel oil and waste biodiesel fuel mixtures on the performance, combustion, and exhaust emission characteristics of a diesel engine. In the experiments, fuel mixtures created by adding different ratios of fusel oil (5–10%) and waste biodiesel (30–50%) to diesel fuel were used, and the engine was tested under different load conditions. In terms of performance, it was reported that the specific fuel consumption increased due to the low lower calorific values of fusel oil and waste biodiesel, but the engine was able to operate stably at appropriate blend ratios. NOx emissions varied depending on engine load and fuel composition; while an increasing trend was observed under some operating conditions, they remained at limited levels under low and medium loads.

Although there are several studies in the literature for fusel oil use as single-fuel or dual-fuel systems in diesel engines [31–35], its performance assessment in terms of environmentally and economically under RCCI combustion strategy is limited. Therefore, the objective of this study is to propose a novel aspect to the existing literature by conducting an experimental investigation into the feasibility of a fusel oil-enriched diesel engine in environmental and economic contexts.

2. Materials and Methods

Experimental studies were conducted in the Automotive Laboratory of the Mechanical Engineering Department of Marmara University Faculty of Technology. The experimental setup and instrumentation for RCCI tests is shown in Figure 1.

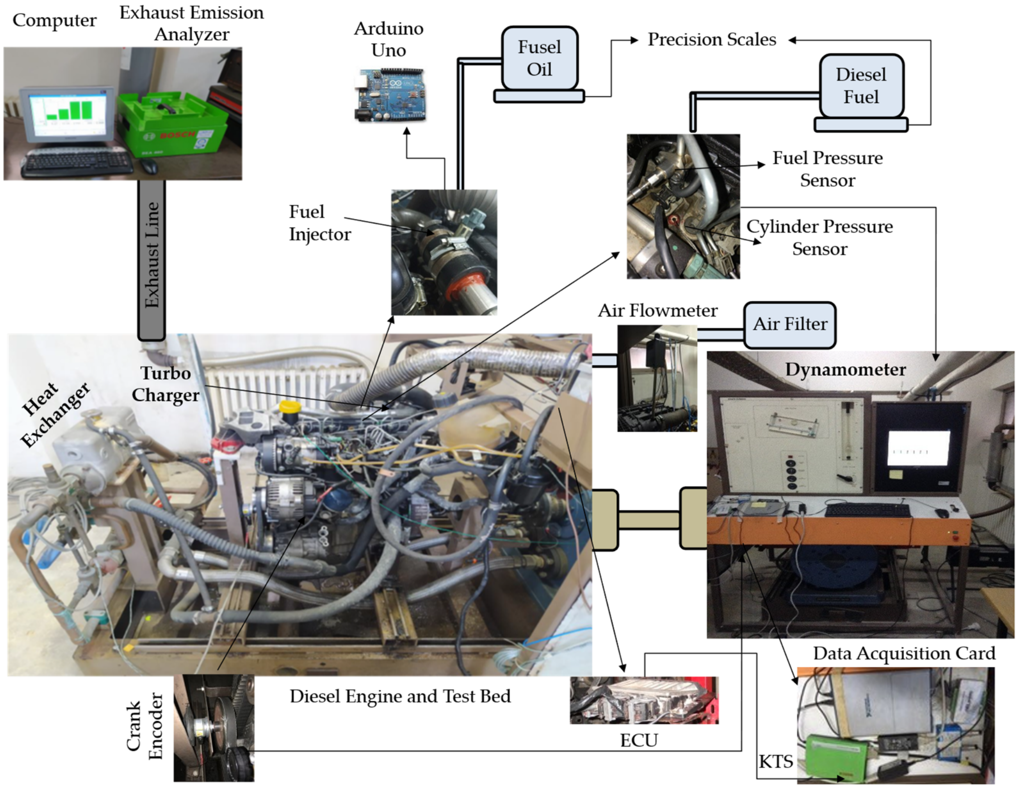

Figure 1. View of the engine test setup.

The experiments were conducted on a K9K-700, 1.461 L, four-cylinder, four-stroke, water-cooled, turbocharged common-rail diesel engine with a compression ratio of 18.25. The engine can produce a maximum torque of 160 Nm at 1750 rpm and a maximum power of 48 kW at 4000 rpm. Engine loading was applied using an eddy-current dynamometer (Cussons P8602) equipped with a city-water heat exchanger, allowing simultaneous cooling of both the engine and the dynamometer. All tests were performed at constant engine speed with electronically controlled load application.

During the experiments, air mass flow rate, cooling water temperature, fuel temperature, engine oil temperature, and exhaust gas temperature were continuously recorded by the dynamometer control system. Temperature measurements were carried out using K-type thermocouples (Omega). The intake air flow rate was measured using an air flow meter (New Flow), while the mass flow rates of liquid fuels were determined using a precision balance and a stopwatch. In addition, the mass consumption of air and diesel fuel per cylinder per cycle was obtained from the engine ECU via a fault detection device (Bosch KTS 570).

The combustion analysis system consisted of an in-cylinder pressure sensor (Oprant Model 33288 GPA), a fuel line pressure sensor (Kistler 4067C2000S), a crank angle encoder (Kubler), a data acquisition card (National Instruments, Model 6343), and combustion analysis software (Febris). The pressure sensors were installed in the glow-plug housing and at the fuel line inlet of the injector of the first cylinder. In-cylinder and fuel line pressure data were recorded with a resolution of 1 crank angle degree. The accuracy specifications of all measurement devices are provided in Table 1.

Table 1. Used device accuracy.

Measurement	Device	Accuracy
Engine load	Load cell	±0.25
Engine speed	Speed sensor	±0.1%
Crank Angle	Optical encoder	±0.2%
Cylinder pressure	Piezoelectric Sensor	±0.8%

Exhaust emissions, including NO, were measured using a Bosch BEA 460 emissions analyser. The NO measurement range of the emissions analyser was 0–5000 ppm, with a sensitivity of 1 ppm. The calculated measurement uncertainty values are provided in Table 2.

Table 2. Calculated uncertainty values.

	Uncertainty (%)
Torque	0.43
Power	0.55
BSFC	0.24
Cylinder pressure	1
Air flow rate	1
Smoke	1
Nitrogen oxide (NO)	1
Total	5.22

Additionally, a specially manufactured injection chamber was installed between the turbocharger outlet pipe and the intake manifold to serve as the fuel supply system for RCCI combustion. A fuel injector was mounted on the injection chamber to deliver the low-reactivity fuel. The fuel line pressure was maintained at 2.8 bar, and the injector injection duration was precisely controlled using an Arduino Uno microcontroller.

The test fuels were designated based on the mass of fusel oil injected into the cylinders (in grams). As the reference fuel, commercial diesel obtained from fuel stations was denoted as D. The tests conducted with the injection of 4 g/min, 6 g/min, 10 g/min, 12 g/min, and 16 g/min of fusel oil were designated as DF4, DF6, DF10, DF12, and DF16 respectively. Fusel oil was directly injected into the intake air with fuel injector and entered the cylinders. The determination of the physical properties of fusel oil was carried out at TÜBİTAK-MAM.

Fusel oil is an oxygenated higher-alcohol byproduct, and its physicochemical characteristics make it suitable as the low-reactivity supplementary fuel in RCCI operation. As summarised in Table 3, the measured density (832.2 kg/m^3) and kinematic viscosity (2.605 mm^2/s at 40 °C) support stable intake injection and mixture preparation, while the very low pour point (<-51 °C) facilitates fuel handling under cold conditions. The oxygenated structure can promote more homogeneous premixing in the intake charge, which is beneficial for RCCI reactivity stratification and stable combustion. In addition, the inherent water fraction (7.43% m/m) and the high latent heat typically associated with alcohol-based fuels can contribute to charge cooling and lower peak temperatures, which may help suppress thermal NO formation depending on the operating load and air excess ratio. In addition, the heating values of fusel oil were calculated based on its known chemical composition [36–38].

Table 3. Fusel oil properties used in the experiments.

	Fusel Oil	Method
Density (g/dm^3)	832.2	ASTM D 4052
Viscosity (mm^2/s @40 °C)	2.605	ASTM D 445
Pour Point (°C)	<−51	ASTM D 97
Flash Point (°C)	39.5	ASTM D 93
Water % (m/m)	7.43	ASTM D 6304
HHV (MJ/kg)	32.55	Calculated [36,37]
LHV (MJ/kg)	27.88	Calculated [38]

The study commenced with the fabrication and installation of an intermediate component incorporating a fuel injector, which was positioned between the turbocharger outlet and the intake manifold to enable the RCCI operation of the test engine. The injection timing was adjustable via an Arduino-based control unit, allowing precise regulation of the fuel quantity delivered to the cylinders. The experiments were carried out at engine loads of 40, 60, 80, and 100 Nm while maintaining a constant engine speed of 1750 rpm. Baseline data were first obtained by operating the test engine under conventional diesel combustion. Subsequently, RCCI experiments were performed by progressively reducing the diesel fuel quantity and injecting fusel oil at rates of 4, 6, 10, 12, and 16 g/min for each load condition. Each experimental condition was repeated three times, and the reported results represent the averaged values.

The fusel oil share is the proportion of fusel oil energy in the overall fuel input to the engine. m_{diesel} and $m_{fuseloil}$ are the mass flow rates of diesel and fusel oil, respectively, while LHV_{diesel} and $LHV_{fuseloil}$ are the lower heating values of diesel and fusel oil respectively. The fusel oil energy share is calculated based on Equation (1).

$$\%\ \text{Fusel oil energy share} = \frac{(m_{fuseloil} \times LHV_{fuseloil})}{(m_{diesel} \times LHV_{diesel}) + (m_{fuseloil} \times LHV_{fuseloil})} \quad (1)$$

The air-fuel ratio (AFR) is defined as the mass ratio of the air supplied to the engine to the mass of fuel participating in the combustion process [39]:

$$AFR = m_{air}/m_{fuel} \quad (2)$$

There is a direct relationship between the λ and AFR. The air-fuel ratio corresponding to a given λ value can be obtained by multiplying λ by the stoichiometric air–fuel ratio of the fuel. Conversely, λ can be determined by dividing the air-fuel ratio by the stoichiometric air-fuel ratio of the fuel.

$$\lambda = \frac{AFR}{AFR_{stoich}} \quad (3)$$

Uncertainty analysis was calculated using the root-sum squares (RSS) method of the combined uncertainty of the independent measurement variables [40].

$$U = \sqrt{U_1^2 + U_2^2 + \ldots} \quad (4)$$

A detailed cost analysis was employed on a diesel engine. For a complete financial overview, capital (CAPEX) and operating (OPEX) expenditures were determined. Additionally, a life cycle cost (LCC) study compared the long-term cost effects of diesel versus diesel–fusel oil fuelling. The data could help establish the total cost viability and profitability of fusel oil for engines. At a set discount rate (R), CAPEX was the fixed annual instalment that equally divides the original investment cost (IC) throughout the lifespan of investment (n). The following equation defines it.

$$\mathrm{CAPEX} = \mathrm{IC}\frac{\mathrm{R}(1+\mathrm{R})^{\mathrm{n}}}{(1+\mathrm{R})^{\mathrm{n}}-1} \quad (5)$$

A diesel engine has three main expenses: fuel consumption cost (FCC), lubricating oil cost (LOC), and maintenance cost (MC). In four-stroke engines, lubrication costs are negligible compared to other costs. Therefore, the OPEX value is calculated as follows.

$$\mathrm{OPEX} = \mathrm{FCC} + \mathrm{MC} \quad (6)$$

The FCC,

$$\mathrm{FCC}_{\mathrm{annual}} = \mathrm{FP}_{\mathrm{fuel}}\sum\nolimits_{\mathrm{i}=1}^{\mathrm{n}}(\mathrm{SFC}_{\mathrm{i}}\mathrm{P}_{\mathrm{i}}\mathrm{T}_{\mathrm{i}}) \quad (7)$$

The MC,

$$\mathrm{MC}_{\mathrm{annual}} = \sum\nolimits_{\mathrm{i}=1}^{\mathrm{n}}(\mathrm{SMC}_{\mathrm{i}}\mathrm{P}_{\mathrm{i}}\mathrm{T}_{\mathrm{i}}) \quad (8)$$

The LCC includes CAPEX and OPEX over an engine's financial lifespan. Calculating OPEX values and discounting them is done using an appropriate discounting function that takes into account the operational period (t) and the discount rate (r). The following equation is utilised for the LCC indicator via discounted operational costs and CAPEX.

$$\mathrm{LCC} = \mathrm{CAPEX} + \sum\nolimits_{\mathrm{t}=0}^{\mathrm{n}}\frac{\mathrm{OPEX}_{\mathrm{t}}}{(1+\mathrm{r})^{\mathrm{t}}} \quad (9)$$

where

FP: Price of Fuel [USD/kg]
T: Operating Time [hours]
SFC: Specific Consumption of Fuel [kg/kWh]
SMC: Specific Maintenance and Repair Cost [USD/kWh]

3. Results

3.1. Combustion, Performance and Emission Analysis

Figure 2 shows the cylinder pressure values with respect to cylinder volumes for tested engine loads. Additionally, maximum cylinder pressures ($\mathrm{CP}_{\mathrm{max}}$) and the CA values at which these pressures occur are provided in Table 4. As can be seen, cylinder pressures increased with load. At a load of 40 Nm, maximum cylinder pressures occurred near TDC. This was due to the low amount of fuel delivered to the cylinder during the main injection phase. However, the increase in the amount of fuel delivered to the cylinder during the main injection phase due to the increased load caused the maximum cylinder pressures to move away from TDC. As the amount of fuel burned during the pilot combustion phase increased, the amount of diesel fuel burned during the main combustion phase decreased. Therefore, as the fusel oil amount increased, maximum pressures occurred near TDC. The combustion of the pre-mixed fusel oil–air mixture ignited by the pilot diesel fuel in a small chamber increased cylinder pressure. As the energy share of the fusel oil decreased, the amount of diesel fuel burned during the main combustion phase increased. Thus, maximum cylinder pressures moved away from TDC. The decrease in energy share reduced the effect of fusel oil on cylinder pressures. At 100 Nm engine load, this effect shifted the point at which maximum pressures occur by 1–2 CA while increasing maximum cylinder pressure by 1–4%.

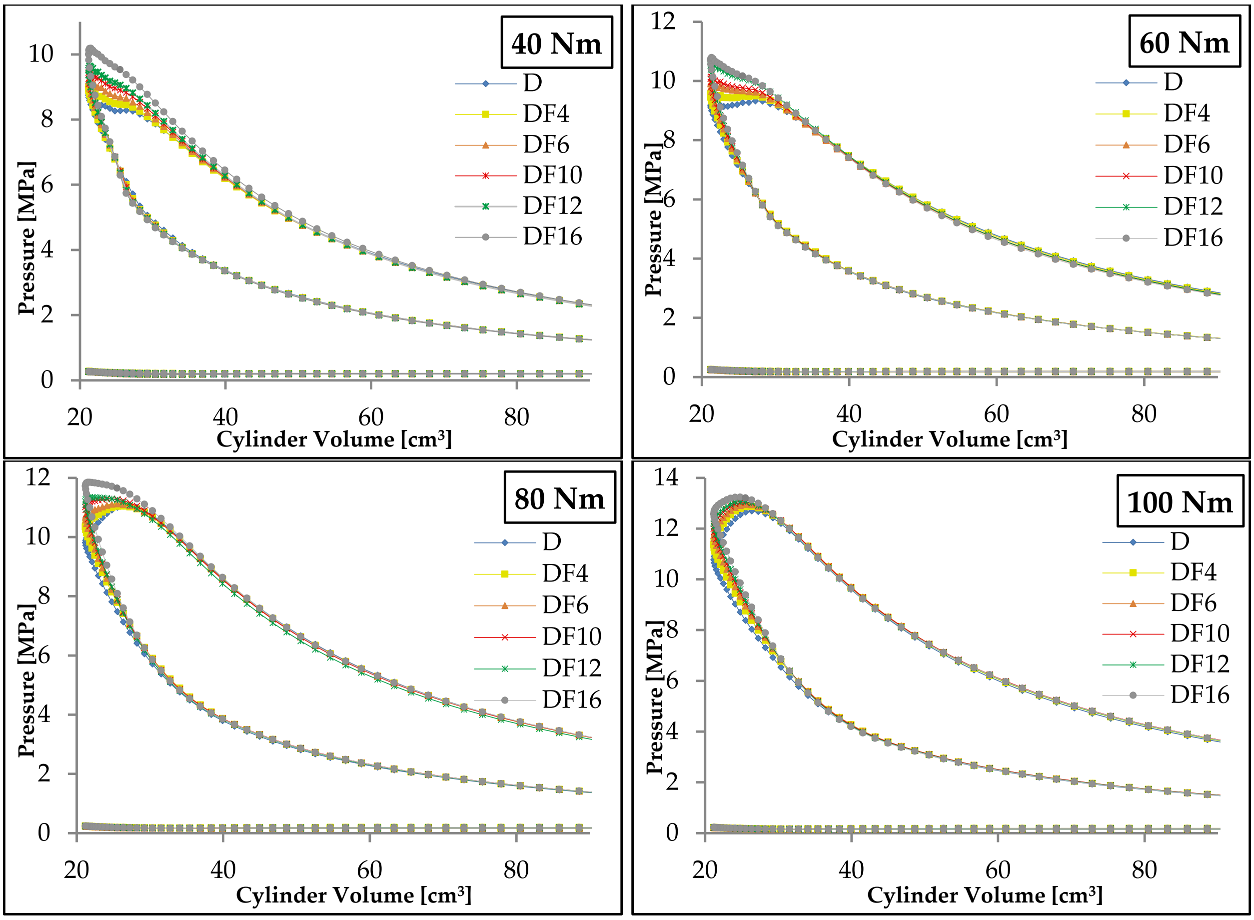

Figure 2. In-cylinder pressure versus cylinder volume at various engine loads for diesel and fusel oil.

Table 4. Effects of engine load and fuel type on maximum cylinder pressure and crank angle.

		D	DF4	DF6	DF10	DF12	DF16
40 Nm	CP_{max}	8756	9005	9259	9479	9642	10,185
	CA	361	361	362	362	362	363
60 Nm	CP_{max}	9324	9568	9970	10,140	10,608	10,792
	CA	373	361	362	362	362	362
80 Nm	CP_{max}	11,053	11,041	11,120	11,289	11,338	11,849
	CA	373	372	371	370	364	364
100 Nm	CP_{max}	12,694	12,870	12,948	13,036	13,068	13,238
	CA	372	372	371	371	370	370

The energy shares of the test fuels can be seen in Figure 3. The highest fusel oil energy share was observed in DF16 at a motor load of 40 Nm. Engine stability was unable to be maintained at a fusel oil mass flow rate exceeding 16 g/min. Fusel oil energy shares decreased in line with the increased engine load. This was due to the constant mass flow rates of fusel oil for each test condition. Increasing the fusel oil ratio to meet the engine's energy requirements increased the maximum cylinder pressure rise rates. Thus, the test engine began to run noisily and unstably.

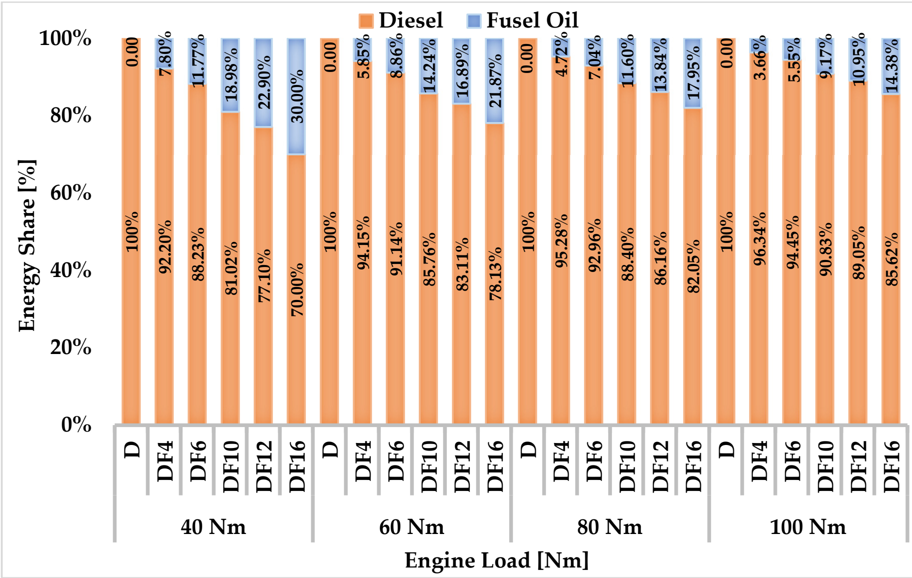

Figure 3. Energy shares of fusel oil and diesel fuels.

Figure 4 shows the lambda values for all tests. Lambda is the ratio of the actual air–fuel ratio to the theoretical air–fuel ratio. Lambda values decreased with increasing engine load. Although turbocharger efficiency was lower at high engine loads, the lambda value was high at low engine loads. Due to this, the engine could reach the target load of 40 Nm with less fuel mass. Accordingly, more fuel was sent to the cylinders to achieve higher loads. Thus, cylinder temperatures, exhaust gas enthalpy, and turbocharger efficiency increased. On the other hand, increased fuel mass decreased lambda values. The theoretical air–fuel ratio of fusel oil was lower than that of diesel fuel. However, due to the low heating value of fusel oil, more fusel oil was sent to meet the engine's energy requirements. The lambda values obtained in the DF 16 tests decreased by approximately 10% compared to the tests conducted with diesel fuel under other test conditions, except for the 100 Nm engine load test. The decrease under the 100 Nm test condition was approximately 7%.

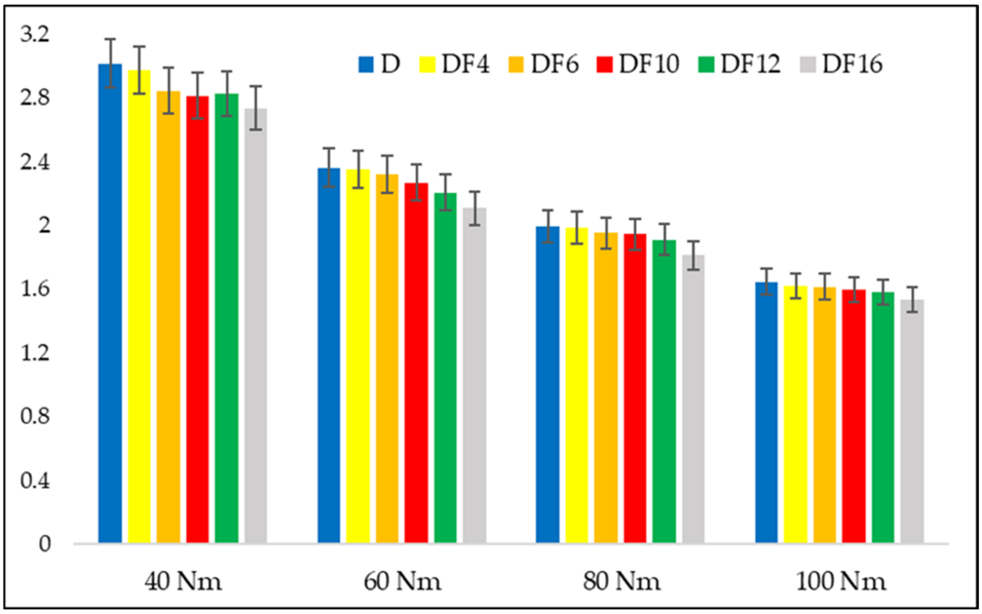

Figure 4. Calculated lambda values for each test condition.

Figure 5 shows the cylinder temperature changes occurring at 40 Nm load under different fuels. The maximum cylinder temperatures were 1962 K at 381 °CA with D and 2088 K at 378 °CA with DF16. Analysis of the data revealed that the maximum cylinder temperatures increased by 6.43% with DF16 compared to when the engine was running on pure diesel fuel.

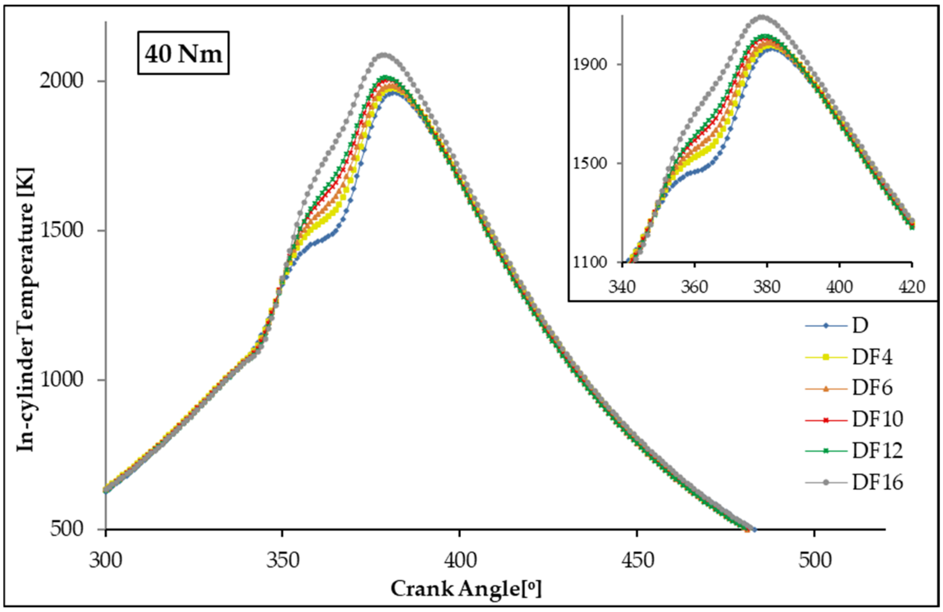

Figure 5. Cylinder temperatures at a load of 40 Nm.

Maximum cylinder temperatures occurring under 60 Nm load with different fuels were measured as 2273 K at 384 °CA with D and 2296 K at 382 °CA with DF16. Analysis of the data revealed that maximum cylinder temperatures with DF16 increased by 1.02% compared to when the engine was running on diesel fuel only.

At a load of 80 Nm, the maximum cylinder temperatures occurring with different fuels were measured at 382 °CA, 2527 K in D, and 2573 K in DF16. Upon analysis of the data, it was determined that the maximum cylinder temperatures with DF16 increased by 1.02% compared to when the engine was running solely on diesel fuel.

At a load of 100 Nm, the maximum cylinder temperatures occurring with different fuels were 381 °CA, D 2709 K, and DF16 2734 K. Upon examination of the data, it was determined that the maximum cylinder temperatures increased by 0.94% with DF16 compared to when the engine was running on diesel fuel only.

Based on examining the cylinder temperature graph, it was observed that the increase in fusel quantity had a slight effect on maximum cylinder temperatures and positions. The test engine's common rail system had two injection stages. Fuel oil was injected into the mixing chamber to form a homogeneous mixture. This mixture was ignited with pilot diesel fuel. As the amount of fuel oil increased, more heat was released during the pilot combustion stage. This effect could be observed before TDC under all test conditions. On the other hand, it was observed that the amount of diesel in the main injection stage had a negligible effect on maximum cylinder temperatures. Although maximum cylinder temperatures during the pilot combustion stage increased with fusel oil usage, the maximum temperatures resulting from main combustion were similar under all test conditions. This situation could be attributed to the water content of fusel oil.

Overall, across the investigated torque range (40–100 Nm), increasing engine load increased the overall fueling demand and reduced the λ (Figure 4), which led to higher in-cylinder temperatures in all cases (Figures 5–8). The fusel oil energy share was highest at low load (40 Nm) and decreased as load increased (Figure 3), primarily because the fusel oil mass flow rate was kept constant for each test point, while additional energy demand at higher loads was met by increasing the total fuel input. Consequently, the relative impact of fusel oil on peak in-cylinder temperature was strongest at 40 Nm and progressively diminished at higher loads.

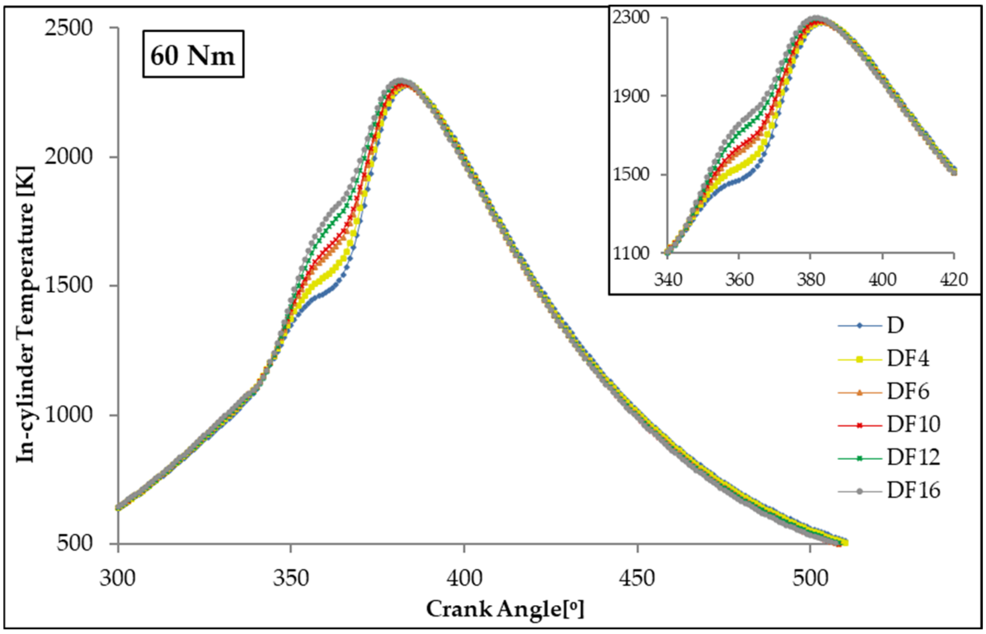

Figure 6. Cylinder temperatures at a load of 60 Nm.

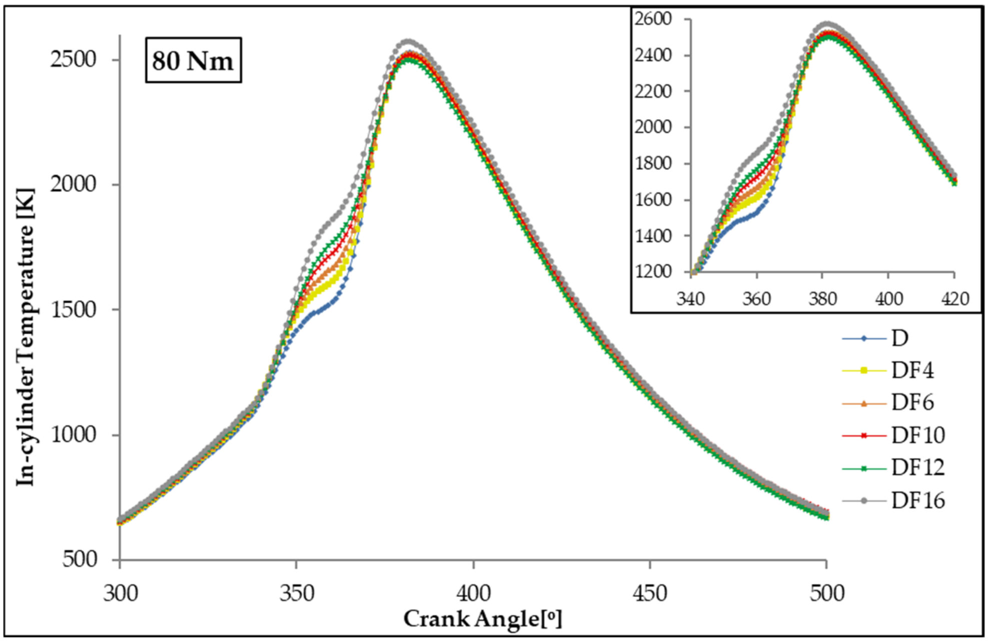

Figure 7. Cylinder temperatures at a load of 80 Nm.

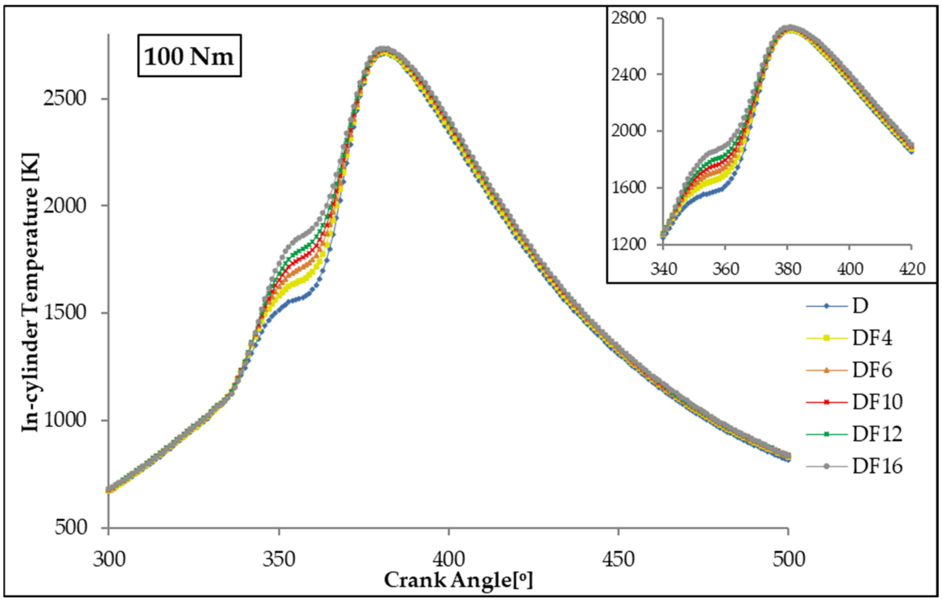

Figure 8. Cylinder temperatures at a load of 100 Nm.

Figure 9 illustrates the variation of NO emissions with engine load for different fusel oil ratios. At the lowest load (40 Nm), NO emissions showed an increasing trend with increasing fusel oil addition, reaching a maximum increase of 17.9% at DF16 compared to diesel operation. In contrast, at medium and high loads (60–100 Nm), the addition of fusel oil led to a consistent reduction in NO emissions. The most pronounced decrease was observed at 80 Nm, where NO emissions were reduced by up to 13.54% at the highest fusel oil rate. At 60 Nm and 100 Nm loads, NO reductions of 7.13% and 5.25%, respectively, were achieved with DF16. Overall, these results indicate that fusel oil addition increases NO formation at low loads, while providing a clear NO reduction benefit under medium and high load conditions.

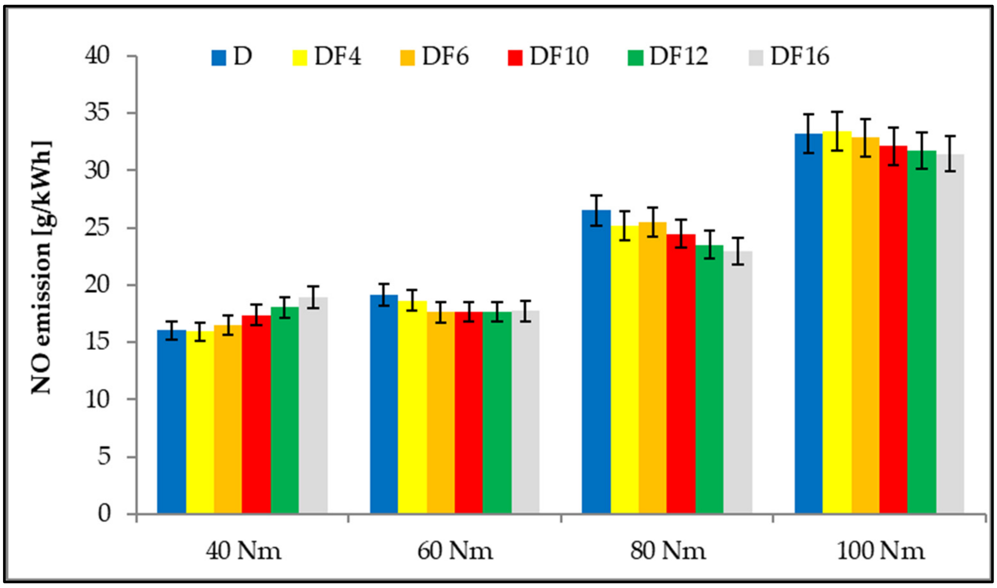

Figure 9. NO emissions with respect to different engine load conditions.

When cylinder temperatures and lambda graphs were examined, it was observed that temperatures increased as the fusel oil quantity increased, while lambda values decreased. However, even though lambda values decreased, they were relatively high, especially at

40 Nm engine load, compared to other engine loads. Both high temperatures and high lambda values increased NO emissions. At loads above 40 Nm, cylinder temperatures increased depending on the load. However, the effect of fusel oil on this increase diminished. In this case, the lambda values began to play a significant role in NO formation. After a load of 60 Nm, the temperature difference decreased by 20 K. In this scenario, the effect of air excess ratio on emissions was more pronounced than the effect of temperature. For DF16, λ decreased from 2.74 at a load of 40 Nm to 1.53 at 100 Nm. This demonstrated that air excess ratio played a significant role in NO formation. As the amount of fusel oil increased, the combustion time and air excess ratio decreased. Consequently, NO formation decreased.

In diesel engines, during the premixed uncontrolled combustion phase, hydrogen atoms react primarily with oxygen. The main reason for this is that hydrogen has a higher reactivity than carbon. In contrast, carbon atoms can carbonise without oxidising under the influence of high temperatures and can cause soot and smoke formation, particularly in areas where combustion is incomplete due to insufficient oxygen [41,42].

Figure 10 presents the variation of smoke emissions with engine load for different fusel oil ratios. For all operating conditions, smoke emissions increased with increasing fusel oil addition. At low load (40 Nm), the highest fusel oil ratio (DF16) resulted in a pronounced increase in smoke emissions, reaching approximately 42% compared to diesel operation. A similar increasing trend was observed at 60 Nm, where smoke emissions rose by about 30% at DF16. At medium load (80 Nm), smoke formation was further intensified, with an increase of nearly 39% at the highest fusel oil rate. At the highest load (100 Nm), although absolute smoke levels were higher, the relative increase due to fusel oil addition was more moderate, remaining below 22%. Overall, the results indicated that fusel oil addition consistently promoted smoke formation across the entire load range, with the strongest relative impact observed at low and medium loads.

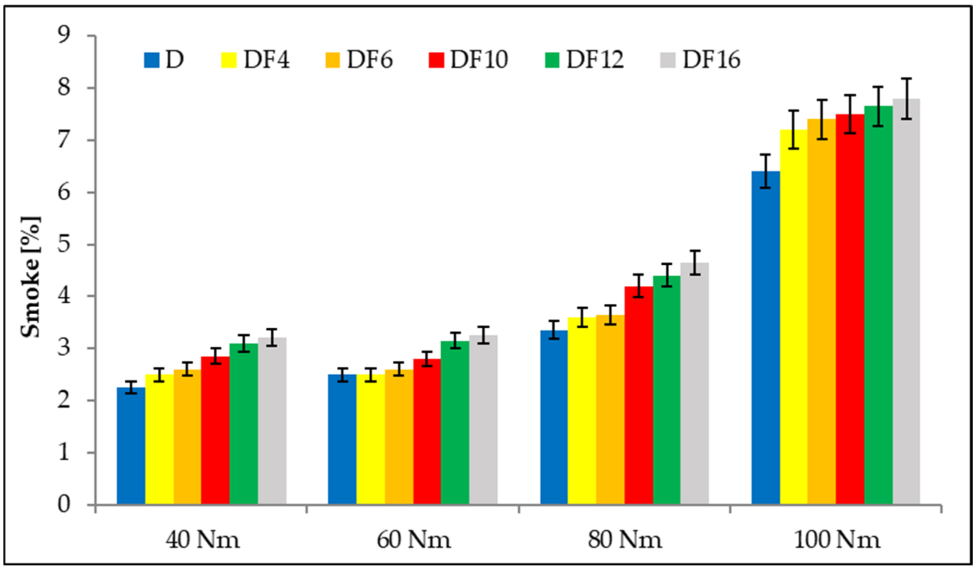

Figure 10. Smoke emissions with respect to different engine load conditions.

In RCCI mode, the fusel oil–air mixture in the combustion chamber was ignited by pilot diesel fuel. The combustion products formed as a result of the pilot combustion phase diluted the cylinder charge and reduced the oxygen concentration. The main injection was performed after the pilot combustion phase. In this case, it became difficult for the diesel fuel sprayed into the cylinder during the main injection to reach the oxygen. Thus, as the fuel oil ratio increased to meet the engine's energy requirements, an increase in smoke emissions was observed [25–27,29,31,43].

In summary, engine load markedly altered the emission response to fusel oil by changing the combined effects of temperature and oxygen availability. At low load (40 Nm), the coexistence of relatively high λ and the fusel oil-driven rise in peak in-cylinder temperature resulted in increased NO emissions (Figure 9). In contrast, at medium and high loads (60–100 Nm), λ decreased and the peak temperatures during the main combustion phase remained comparable across fuels, leading to net NO reductions of 7.13%, 13.54%, and 5.25% at 60, 80, and 100 Nm, respectively (Figure 9). Smoke emissions, however, increased with fusel oil addition at all loads (Figure 10). This behaviour is attributed to oxygen depletion and charge dilution caused by pilot-combustion products, which hindered soot oxidation during the main injection period. The relative smoke penalty was most pronounced at low and medium loads and became more moderate at the highest load, where baseline smoke levels were already higher.

3.2. Economic Analysis

Assessing economic viability relies on the values of LCC, OPEX, and CAPEX. The initial investment cost of the engine is dependent upon the equipment employed for fusel oil combustion. Table 5 lists the requirements for the engine regarding the combustion of fusel oil.

Table 5. IC costs for fusel oil combustion.

	IC	
Fuel Injector	40	USD
Fuel Filter	10	USD
Fuel Pump	40	USD
Installation	1000	USD
Total	1090	USD

With 10% CAPEX discount rate (R) and 10 years lifespan (n), the calculated CAPEX values for standard diesel fuel mode and fusel oil combustion mode were 2421 USD and 2599 USD, respectively. The SFC and power values were determined from the experiments. The test engine was an engine commonly used in passenger cars and light commercial vehicles. So, annual operation time was considered using three scenarios as 360, 720, 1080 h. The purchasing company's invoiced fuel costs were 1.79 USD/kg for diesel fuel and 0.42 USD/kg for fusel oil. Annual maintenance costs for 40, 60, 80, and 100 Nm loads were 50, 79, 100, and 129 USD, respectively [44]. In 4-stroke engines, lubrication costs were lower compared to fuel and maintenance cost, hence they were ignored in this study.

As engine load increased, both OPEX and LCC values rose across all cases, reflecting the higher fuel demand at elevated loads. In contrast, increasing the fusel oil share yielded a clear cost benefit: depending on the load level, OPEX values decreased by up to 33% and LCC by up to 42% relative to diesel fuel, as shown in Figure 11 and Figure 12 and Figure 13, respectively. These trends are reflected in Tables 6 and 7, indicating that fusel oil can provide a substantial economic advantage under the investigated operating conditions.

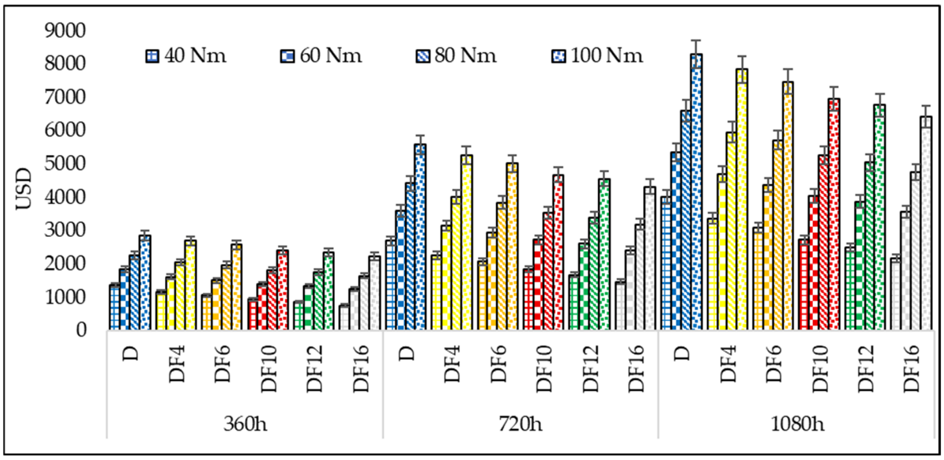

Figure 11. Annual OPEX values for each test.

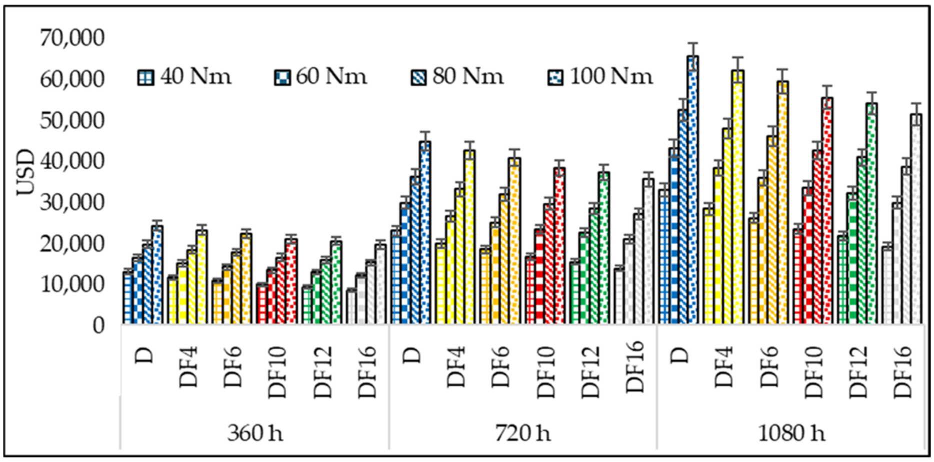

Figure 12. LCC values for 5% discount rate.

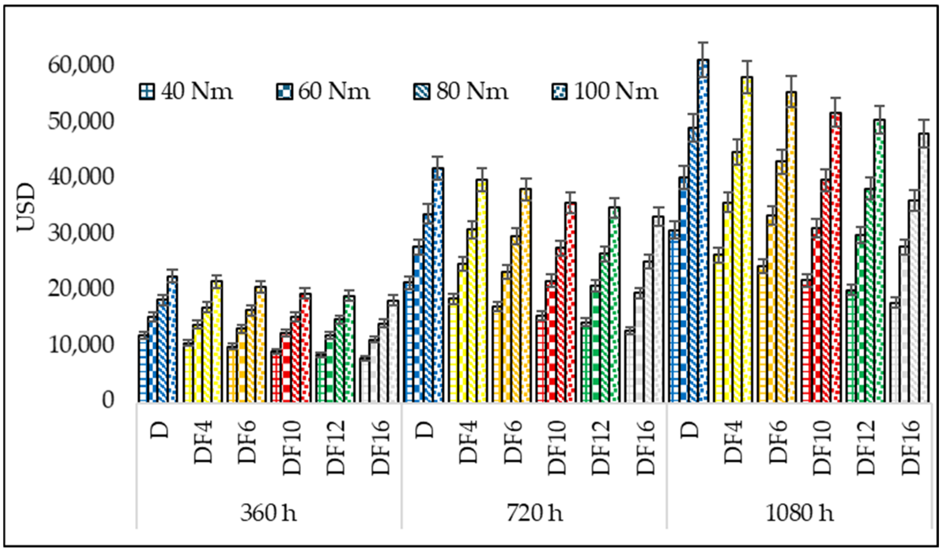

Figure 13. LCC values for 7% discount rate.

Table 6. DF16-induced reduction (%) in annual OPEX compared with diesel fuel (D).

	40 Nm	60 Nm	80 Nm	100 Nm
1. scenario	−44.92%	−32.35%	−27.20%	−21.98%
2. scenario	−45.76%	−33.06%	−27.81%	−22.49%
3. scenario	−46.05%	−33.30%	−28.03%	−22.67%

Table 7. DF16-induced reduction (%) in LCC compared with diesel fuel (D).

	40 Nm		60 Nm		80 Nm		100 Nm	
	r = 5%	r = 7%	r = 5%	r = 7%	r = 5%	r = 7%	r = 5%	r = 7%
Scenario 1	35.22%	34.45%	26.57%	26.08%	23.00%	22.64%	19.08%	18.82%
Scenario 2	40.24%	39.75%	29.82%	29.53%	25.49%	25.28%	20.91%	20.76%
Scenario 3	42.19%	41.84%	31.06%	30.85%	26.42%	26.27%	21.58%	21.47%

4. Discussion

The experimental findings indicated that fusel oil can be effectively utilised within an RCCI strategy, particularly at low and medium engine loads. At low loads, the higher energy contribution of fusel oil was observed; however, at higher engine loads and with increasing fusel oil ratios, this contribution was limited by combustion stability limitations. This behaviour is consistent with the general RCCI principle that the achievable low-reactivity fuel fraction is constrained by combustion phasing control and the pressure rise rate when the premixed portion of heat release increases. In practical terms, the observed stability boundary suggests that the fusel oil supply rate cannot be increased indefinitely and that an optimised balance between reactivity stratification and combustion stability is required for reliable operation. A slight increase in maximum in-cylinder temperatures is observed with fusel oil. This was primarily associated with enhanced heat release during the pilot combustion phase, whereas at higher loads, the maximum temperatures during the main combustion phase remained largely unchanged due to the high water content of fusel oil.

The emission results revealed that NO formation was governed by the combined effects of in-cylinder temperature, air fuel ratio (lambda), and the amount of injected fusel oil. At low engine loads, high lambda values and elevated temperatures tended to promote NO formation, while at higher loads, reduced lambda values and increased fusel oil fractions resulted in a decreasing trend in NO emissions. These results indicate that total fuel input alone does not determine NO trends; instead, oxygen availability and the temperature history around the main heat-release period are dominant. From an RCCI perspective, the reduced lambda at higher loads limits oxygen for thermal NO pathways, and the limited change in main-phase peak temperatures further suppresses NO formation even when additional fusel oil is supplied.

In contrast, smoke emissions consistently increased with fusel oil addition across all load conditions because of reduced oxygen availability in the cylinder caused by pilot combustion. A plausible explanation is that the increased pilot contribution and the associated pre-combustion products reduce the effective oxygen concentration available during the diffusion-controlled portion of the main diesel injection, thereby limiting soot oxidation. In addition, as fusel oil fraction increases and lambda decreases, soot burnout becomes more constrained even if ignition remains stable. The combined NO and smoke results therefore highlight a clear NO–smoke trade-off under the applied RCCI settings. This trade-off suggests that real-world application would benefit from calibration measures that target oxygen availability and soot oxidation, such as optimizing pilot and main injection timings and quantities, adjusting boost and intake temperature, or applying suitable aftertreatment solutions. The observed operability limitation at higher fusel-oil rates further supports the need for an integrated calibration approach, because strategies that reduce smoke may also influence pressure rise rate and combustion stability.

This study further demonstrates that fusel oil represents a promising alternative fuel from an economic perspective for diesel engines operating under RCCI conditions. Although fusel oil usage led to increase in in-cylinder temperatures and smoke emissions, its low fuel cost and wide availability resulted in substantial reductions in operating and life cycle costs. Overall, the results highlight a clear trade-off between economic benefits and combustion and emission performance, indicating that fusel oil is particularly suitable for applications in which cost reduction is prioritized and appropriate emission control strategies can be implemented.

5. Conclusions

In this study, fusel oil was used as a low-reactivity fuel for the RCCI mode in a four-cylinder diesel engine at 1750 rpm under loads ranging from 40 to 100 Nm. Test results showed that the energy share of fusel oil could be increased up to 30% at low loads. However, at higher fusel oil energy shares, the stable operation of the engine was limited.

- The effect of fusel oil on combustion was more pronounced at low engine loads. At 40 Nm, the use of fusel oil resulted in a 14% increase in maximum cylinder pressure and an approximately 4% increase in maximum cylinder temperature compared to diesel fuel tests. These increases were 4% and 1%, respectively, in the test conducted at 100 Nm.
- The use of fusel oil resulted in a reduction in NO emissions compared to diesel fuel tests, except for 40 nm tests. It was observed that the most important parameter in reducing NO emissions is the oxygen concentration.
- The use of fusel oil increased smoke emissions under all test conditions. In tests conducted at 40 Nm, where cylinder temperatures and injection pressures were low, smoke emissions increased by 42% in the test where fusel oil's energy share was highest.
- The techno-economic assessment showed that low-cost fusel oil provided significant cost advantages, despite a modest increase in capital costs for additional equipment. Operating costs decreased by up to 33%, while life-cycle costs decreased by up to 42%, depending on the operating scenario.

Future work will focus on expanding the low-reactivity fuel portfolio and improving emission balancing. Light alcohols such as ethanol and methanol can be used as alternatives to low-reactivity liquid fuels. Additionally, work can be conducted on RCCI engines where carbon-free gaseous fuels such as hydrogen and ammonia are added to the intake air.

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Abbreviations

The following abbreviations are used in this manuscript:

RCCI	Reactivity controlled compression ignition
HCCI	Homogeneous charge compression ignition
BSFC	Brake specific fuel consumption
HHV	High heating value
LHV	Low heating value
AFR	Air–fuel ratio
RSS	Root-sum squares
CAPEX	Capital expenditures
OPEX	Operating expenditures
LCC	Life-cycle cost
R	Discount rate
IC	Investment cost
FCC	Fuel consumption cost
LOC	Lubricating oil cost
MC	Maintenance cost
D	Diesel fuel
DF	Fusel oil
CP_{max}	Maximum cylinder pressure

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