



Optimizing fuel injection timing: Investigating the effects of injection timing on combustion performance and emission characteristics of high and low sulfur fuels in a diesel engine

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Abstract: Owing to environmental considerations, there is an increased interest in the Maritime Industry in regulating sulfur oxide emissions and maintaining engine performance. This necessitates installing exhaust gas cleaning systems onboard ships that are costly or use alternative fuels and low-sulfur marine fuels under 0.50%, as set by the International Maritime Organization. In addition, injection timing optimization greatly influences the performance and emission characteristics of diesel engines. Hence, this study investigates the effects of combining high-sulfur, low-sulfur, and ultra-low-sulfur fuels with fuel injection timing using a 3D simulation model from AVL FIRE R2022a while maintaining constant initial and boundary conditions. The results show that the use of light marine diesel fuel oil and Diesel-EN590 causes an increase in the in-cylinder temperature and pressure compared to heavy fuel oil. Moreover, the results show a significant decrease in emissions, especially soot emissions, and a slight decrease in NO_x and CO₂ emissions. Advanced injection timing results in increased peak pressure and in-cylinder temperature. Considering the emission characteristics, the optimal injection timing of 15CAD promotes CO₂ reduction, whereas 10CAD promotes NO_x emission reduction, although slight variations existed. Therefore, this research strategy effectively improves the engine performance and reduces emissions, thereby mitigating energy and environmental crises.

Keywords: High-sulfur fuel oil, Low-sulfur fuel oil, Ultra-low-sulfur fuel oil, Injection timing, Combustion performance, Emission

1. Introduction

The rapid growth of the maritime industry has resulted in a significant increase in the use of fossil-fueled propulsion systems. Over 80% of global trade is carried out through international shipping, amounting to a daily fuel consumption of over 640,000 tons and 49.5% of the world's residual fuel oil demand [1]. The increased fuel consumption in the marine industry has led to an energy crisis and increased emissions, resulting in the demand for higher fuel efficiency. Ship emissions include nitrogen oxides (NO_x), sulfur oxides (SO_x), carbon dioxide (CO₂), carbon monoxide (CO), and particulate matter (PM). These emissions contribute to greenhouse effects; serious health problems such as premature deaths; cardiovascular, respiratory, and pulmonary diseases; and an increase in sea levels [2]-[4]. Thus, significant challenges exist for the marine engine

sector in terms of reducing ship emissions while maintaining engine performance.

To reduce ship emissions, the International Maritime Organization (IMO) has introduced stricter emission regulations, including a Sulfur Cap for SO_x, Tier III for NO_x, and the implementation of a greenhouse gas strategy [5][6]. The global sulfur cap, which went into effect on January 1, 2020, aimed to reduce the sulfur limit of fuels from 3.50% to 0.50% for ships operating outside designated emission control areas and imposed a stricter limit of 0.10% in emission control areas (ECAs). The emission regulations have presented a great challenge in the shipping industry, and the three best strategies highlighted to comply with the IMO sulfur regulations include the use of fuels with lower amounts of sulfur; installation of exhaust gas cleaning systems (EGCS), including scrubbers; and use of alternative fuels, such

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as liquefied natural gas, alcohol-based fuels, ammonia, and hydrogen [7].

The use of EGCS, which can either be wet or dry scrubbers, to eliminate SOx from both engine and boiler exhaust gases has several advantages, including a reduction in operational costs by using cheaper, high-sulfur fuel. Moreover, the byproducts of dry scrubbers may be recycled and used to produce cement or fertilizers. However, this option incurs high installation and time costs resulting from ship downtime. In addition, the crew workload increases during the operation of the EGCS due to emission monitoring services and other areas have legal restrictions regarding the use of EGCS [8]. The other option of using low-sulfur fuel oils, which are products of refined oils, does not require the use of EGCS and can be used in heavy fuel oil (HFO) blends. In comparison, the alternative of using either marine fuels with lower amounts of sulfur, per the required limits, or other alternative fuels offers a suitable solution for the shipping industry to resolve future energy and environmental crises [9].

Several studies have proved that the use of conventional fuels with low sulfur contents, such as marine gas oil (MGO) and light marine diesel oil (MDO) with sulfur contents ranging from 0.1% to 1.0%, very-low-sulfur fuel oil (VLSFO) with a sulfur content of less than 0.5%, ultra-low sulfur (ULSFO) with a sulfur content equal to or less than 0.1%, and fuel blends of MGO and ULSFO with HFO, reduces not only sulfur oxides but also other pollutants [10][11]. However, certain considerations arise when employing treated or blended VLSFO and ULSFO in diesel engines, such as an increased pour point, reduced lubricity, and incompatibility.

In addition to the use of low-sulfur fuel oils, the continuous improvement of diesel engine combustion performance and reduction of emissions are linked to fuel injection system optimization. Fuel injection timing, injection pressure, and injection duration are the primary factors that significantly affect engine performance and emission characteristics [12][13]. Several studies have demonstrated that advanced injection timing enhances an engine's combustion performance, extends the ignition delay, and improves the air-fuel blend in the cylinders [14][15]. In addition, engine exhaust emissions can be significantly reduced via appropriate injection timing, which has great significance in solving environmental issues [16][17].

Therefore, the purpose of this study is to optimize diesel engine performance and reduce exhaust emissions by combining low-sulfur fuel oils with different fuel injection timing strategies

using the AVL FIRE R2022a simulation software, aiming to contribute to resolving environmental and energy crises.

2. Methodology

2.1 Experimental Setup

In this study, a DOOSAN P158LE-111 engine running on an MDO with a low sulfur content was used to analyze the effects of injection timing on combustion performance and emission characteristics. **Table 1** lists the specifications of the main engine.

Table 1: Principle engine specifications

Parameter	Value/Description
Engine type	4-Stroke, V-Type, 8-cylinder
Power (kW)	484
B.M.E.P (Bar)	23
Rated speed (rpm)	1800
Bore x Stroke (mm)	128 x 142
Connecting rod (mm)	177.5
Compression ratio	14.6:1
Engine Displacement (L)	14.618

Table 2 lists the properties of the MDO used in the experimental analyses.

Table 2: Fuel oil properties

Parameter	Value
Fuel type	MDO
Density (Kg/m ³) at 15°C	846.4
Sulfur (%)	0.034
Cetane number	44
Kinematic viscosity (at 40°C) (mm ² /s)	2.8940
Flashpoint (°C)	72

2.2 Three-Dimensional Simulation Models of the In-Cylinder Flow Combustion

In this analysis, three-dimensional (3D) simulations were conducted from the inlet valve closing (IVC) to exhaust valve opening (EVO) using the AVL FIRE R2022a simulation software. Highly advanced models in the AVL FIRE simulation software are appropriate for accurately simulating the combustion and emission characteristics of diesel, gasoline, and dual-fuel engines [18]-[21].

The generation of a computational mesh of the combustion chamber is essential when modeling the simulation. Owing to the axial symmetry of the engine and the presence of a fuel nozzle

with ten identical holes, one-tenth of the entire 3D computational mesh of the combustion chamber was modeled to reduce the computation time. **Figure 1** shows the two-dimensional (2D) and 3D sector computational meshes as well as the 2D sketch of the piston shape, injector position, and direction of injection.

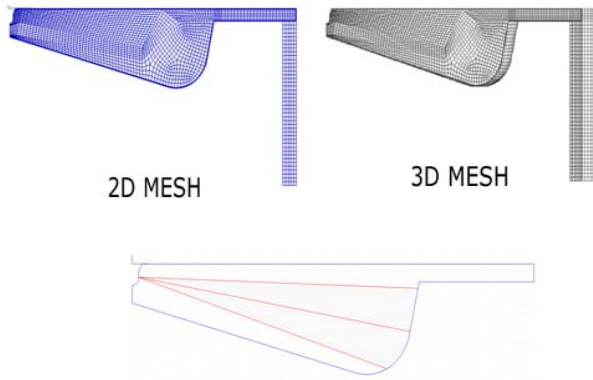


Figure 1: 2D and 3D sector computational meshes and 2D sketch of piston shape, injection nozzle position, and direction of injection

The engine operational boundary and initial condition parameters, which play critical roles in obtaining accurate, stable, and reliable results, were obtained from the experimental analysis to enhance the design optimization of the simulation model. **Tables 3 and 4** show the boundary and initial conditions, respectively.

Table 3: Boundary conditions

Parameter	Boundary type	Specific value
Piston	Mesh movement	570K
Cylinder head	Wall	570K
Liner	Wall	470K
Segment cut	Periodic	Inlet/Outlet

Table 4: Initial conditions

Parameter	Specific value
Charge air temperature(K)	360
Charge air pressure (Pa)	303975
Start of Injection Timing	16CAD BTDC
IVC	35CAD ABDC
EVO	62CAD BBDC

Various models, including turbulence, combustion, ignition, atomization, and emission (NOx and soot) models have been used to accurately perform engine simulations using AVL FIRE R2022a. The turbulence model used in the simulation is the k- ζ -f model, which is an improved four-equation turbulence model

based on the simple k- ϵ model that provides higher accuracy and stability turbulence prediction [22].

A 3-zones extended coherent flame species transport model was employed to study the combustion process. This model is outstanding compared with other coherent flame models, because of its partitioning of the combustion zone features into pre-flame, flame, and post-flame zones, which thereby increases the overall accuracy of the results [23].

Due to its high accuracy over a wide range of equivalent ratios, the extended Zeldovich model was employed to simulate the NOx emissions. These emission form in the engine combustion chamber when atmospheric nitrogen in the charge air reacts with atmospheric oxygen at high temperatures during the combustion process [24].

A kinetic soot mechanism model was used to model the formation of soot as a result of the incomplete combustion in internal combustion engines. By precisely calculating and predicting all chemical reactions, particle development, and particle combustion steps, this model can effectively represent the amount and characteristics of emissions, depending on the engine operating conditions [24].

An autoignition model was employed for ignition. Moreover, the WAVE, Dukowicz, and Walljet1 models were utilized in the atomization process, which involves breaking liquid fuel into smaller droplets, evaporation, and droplet wall interaction processes [25]. **Table 5** lists these models.

Table 5: Summary of the simulation model

Model	Description
Turbulence	k- ζ -f
Combustion	Extended coherent flame model (ECFM)
Emissions	NO - Extended Zeldovich Soot - Kinetic soot mechanism
Ignition	Auto-ignition Breakup - WAVE
Atomization	Evaporation - Dukowicz Droplet-Wall Interactions - Walljet1

2.3 Simulation Case Description

A comparative analysis was conducted to investigate the impacts of various levels of sulfur content in fuels and the effects of injection timing on diesel engine combustion, that is, the in-cylinder pressure, in-cylinder temperature, brake-specific fuel consumption, and emission characteristics for NOx, CO₂, and soot. To investigate the effects of various levels of sulfur content in fuels and injection timing, simulation models of HFO with high

sulfur content, MDO with low sulfur content, and Diesel EN590 (D-EN590) with ultra-low sulfur content were modeled. **Table 6** lists the fuel properties. Simultaneously, to investigate the effects of the injection timing, the variations in the injection timing from advanced to late ignition timing included 15CAD BTDC, 10CAD BTDC, and 5CAD BTDC. During the simulation, the total amount of fuel injected into the engine and all other boundary, initial conditions, and engine operating conditions remained constant.

Table 6: Fuel properties

Parameter	HFO	MDO	D - EN590
Density (Kg/m ³) at 15°C	990	846.4	844
Sulfur (%)	2.16	0.034	0.001
Cetane number	30	44	50
Kinematic viscosity (at 40°C) (mm ² /s)	350	2.8940	2.860
Flashpoint (°C)	135	72	70
Calorific value (MJ/kg)	40.2	44.64	43.4

2.4 Simulation Model Validation

The MDO simulation results were compared with the experimental results to verify the reliability and accuracy of the simulation model. The experimental values used for the validation were the brake mean effective pressure, indicated specific power, and carbon dioxide (CO₂) mass fractions. The deviations in the brake mean effective pressure, indicated specific power, and CO₂ mass fraction were 4.91%, 0.78%, and 4.68%, respectively. **Figure 2** shows a comparison of the experimental and simulation results.

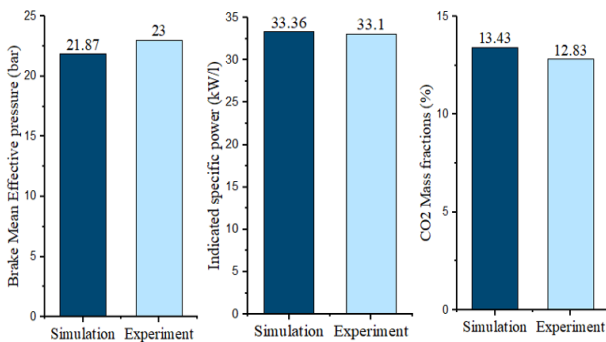


Figure 2: Comparison of the experimental and simulation results

2.5 Mesh Independence Analysis

Mesh independence analysis is an important approach for investigating whether the simulation results are independent of the geometric mesh size. Therefore, to establish the accuracy of the results and keep the computational costs low, combustion simulations were conducted on three developed meshes, that is coarse, medium, and fine, at the standard injection timing of 16CAD BTDC. **Table 7** lists the mesh properties and calculation times.

Table 7: Mesh properties and time of calculation

Category	Coarse Mesh	Medium Mesh	Fine Mesh
No. of faces	2030	2629	5213
No. of cells	100,484	141,600	221,868
Calculation time	0 h 45 min	1 h 9 min	2 h 3 min

Figure 3 shows the results of the in-cylinder temperature and pressure obtained when three mesh resolutions were used. The final results were independent of the three meshes used in this study. Therefore, the use of any of the three mesh configurations yielded accurate results. Mesh 2 (medium) was selected for use in the simulations for the entire study because of its reasonable calculation time and the accurate results obtained during the mesh independence study.

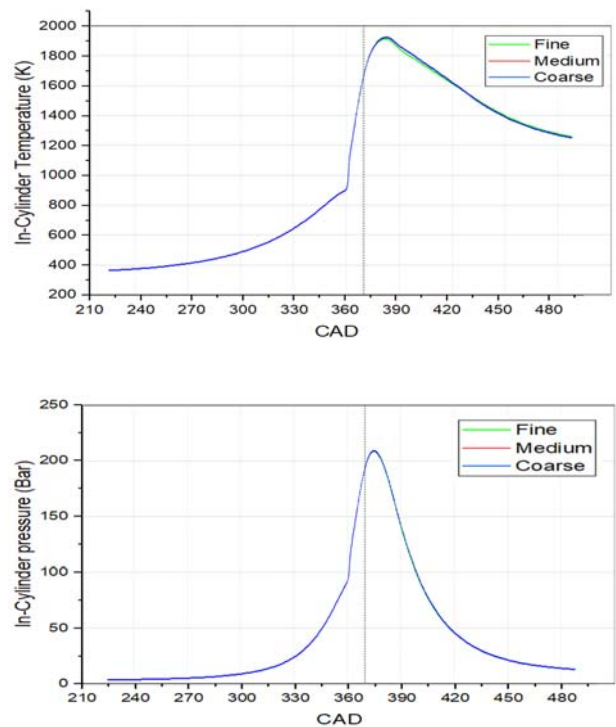


Figure 3: Results of the in-cylinder temperature and in-cylinder pressure, respectively, using the three different types of meshes

3. Simulation Results and Discussions

3.1 In-Cylinder Peak Pressure

Figure 4 shows the variations in the in-cylinder pressures of different fuels at different injection timings, using the same total amount of fuel injected. The in-cylinder pressure profiles of all fuels were similar, although there were small pressure variations between fuels. For example, at SOI 15CAD BTDC, D-EN590 exhibited the highest pressure of 204.23 bar, followed by MDO and HFO with 203.82 bar and 202.70 bar, respectively. This variation is attributable to the differences in viscosity, cetane number, and calorific value. The lower the cetane number in a fuel, the slower the combustion rate; hence, the cylinder pressures are slightly lower than those of other fuels. In addition, HFO has a lower calorific value than do MDO and D-EN590, which results in the release of less energy per unit time during combustion and thus lowers the overall in-cylinder pressure [26]. In terms of injection timing, the in-cylinder pressure increases with the advancement of injection timing. The in-cylinder pressure was highest at 15CAD, and this was followed by 10CAD and 5CAD BTDC. These results are due to the longer ignition delay, which results in more fuel and air blends, and thus more efficient combustion [27].

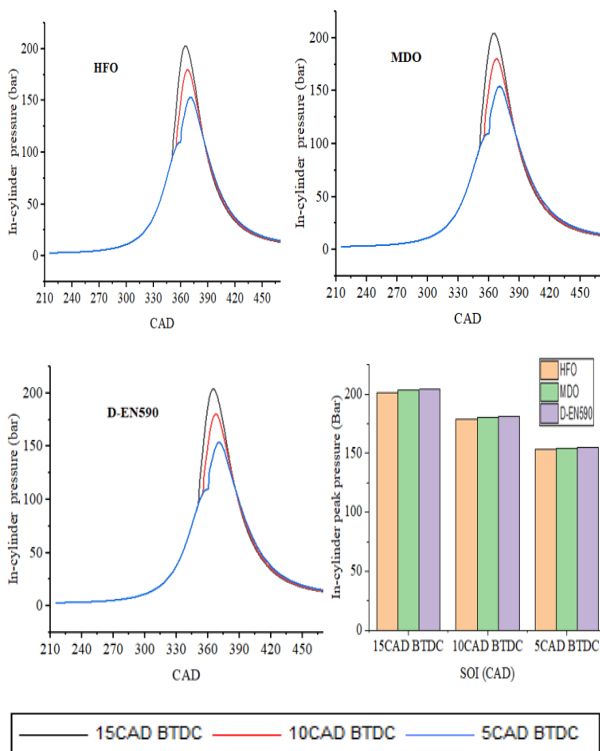


Figure 4: In-cylinder pressure variations for different fuels at different injection timings

3.2 In-cylinder mean temperature

Figure 5 shows the variations in the in-cylinder mean temperatures of the different fuels at different injection timings under the same fuel injection conditions. The in-cylinder temperatures of the various fuels exhibit small variations. At 15CAD BTDC, D-EN590, MDO, and HFO exhibit temperatures of 1899.82 K, 1896.68 K, and 1895.63 K, respectively. These results are attributed to the differences in the cetane numbers and atomization efficiencies of the fuels, which result from the differences in their viscosities. Specifically, lower viscosity leads to more efficient and complete combustion, even though the calorific values of the three fuels are in the same range. The advancement of the injection timing resulted in increased in-cylinder temperatures, owing to the increased fuel-air blended mixture in the combustion chamber [28]. However, the in-cylinder temperatures were slightly higher at 5CAD BTDC than at 10CAD BTDC, due to the shorter ignition delays at peak pressures and high fuel temperatures [29].

3.3 Nitrogen Oxide (NO) Emissions

Figure 6 illustrates the NO mass fractions of different fuels at different injection timing for the same amount of injected fuel.

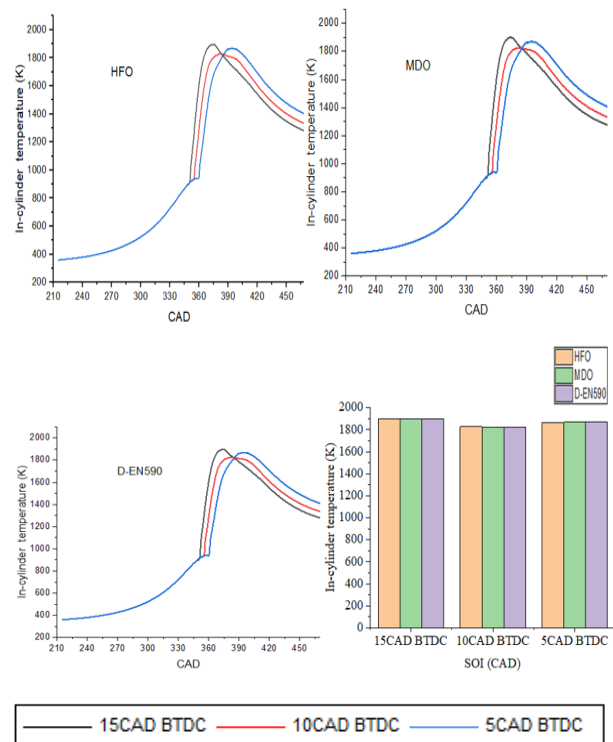


Figure 5: In-cylinder temperature variations for different fuels at different injection timings

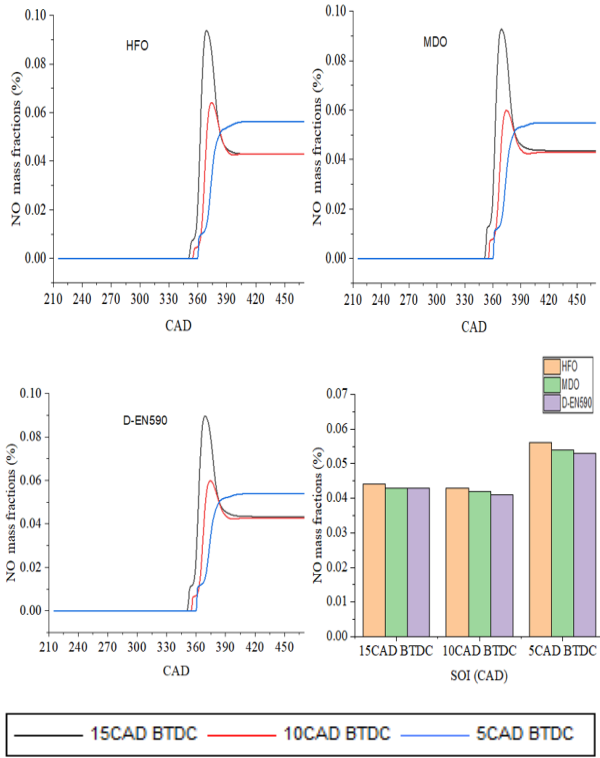


Figure 6: NO mass fraction variations for different fuels at different injection timings

Temperatures above 1800 K, oxygen content, and reaction time inside the engine cylinder had the most significant effects on NO production. HFO has a higher brake fuel oil consumption of 0.266 kg/kWh than do MDO and D-EN590, which have consumptions of 0.255 kg/kWh and 0.252 kg/kWh, respectively. Thus, the peak in-cylinder temperature reaction with atmospheric nitrogen in the combustion chamber combined with fuel NO produced, owing to the high nitrogen content in HFO, results in the ideal formation of higher NO_x in HFO, compared with MDO and D-EN590 [30]. In terms of injection timing, which has an influence on the ignition delay time and ignition energy produced where the temperature curves are highest, that is, 5CAD BTDC, NO_x formation increased [31].

3.4 Carbon Dioxide (CO₂) Emissions

Figure 7 shows the CO₂ mass fractions of different fuels at different injection timings using the same amount of total fuel injected. CO₂ emissions from the combustion chamber depend on several factors, such as the amount of carbon atoms in the fuel, amount of fuel injected, and combustion quality. CO₂ is formed when the produced CO is further oxidized, owing to the presence of a sufficient concentration of oxygen and in-cylinder temperature conditions [32]. HFO produced slightly higher carbon dioxide

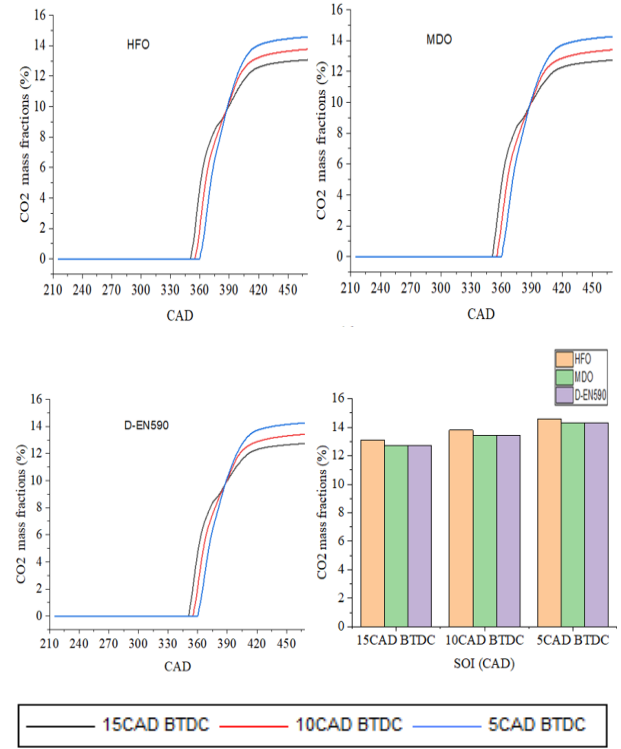


Figure 7: CO₂ mass fraction variations for different fuels at different injection timings

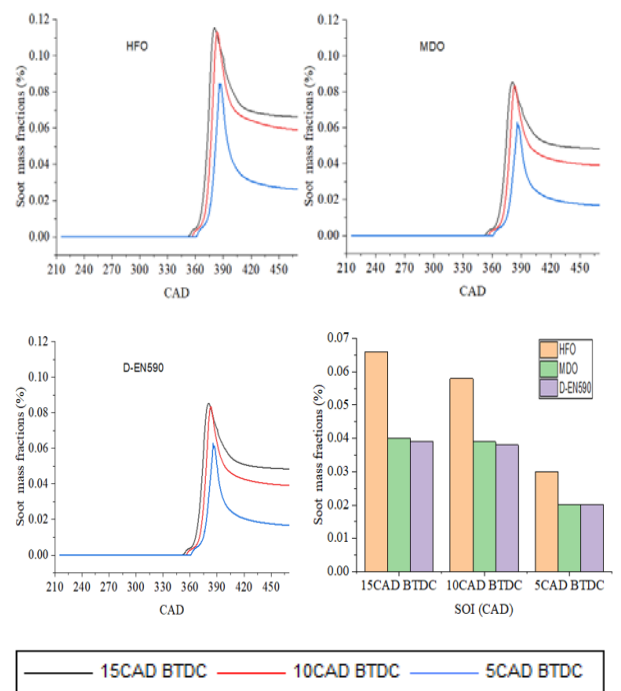


Figure 8: Soot mass fraction variations for different fuels at different injection timing

emissions, compared with MDO and D-EN590. These results are attributable to the high carbon content of HFO and its poor combustion efficiency [33]. Furthermore, the injection timing at the

top dead center is accompanied by high temperatures, which leads to high CO₂ formation, as high temperatures facilitate the oxidation of CO to CO₂ [34].

3.5 Soot Emissions

Figure 8 shows the soot mass fractions of different fuels at different injection timings for the same amount of injected fuel. The soot emissions from HFO were significantly higher than those from MDO and D-EN590. These results are attributable to the higher viscosity and poor volatility of the fuel, which led to poor fuel atomization and incomplete combustion [35]. In addition, the soot mass fractions were lower at injection timings with higher in-cylinder temperatures. The D-EN590 values at 15CAD, 10CAD and 5CAD BTDC were 0.040%, 0.039%, and 0.025%, respectively, because high temperatures enhance soot oxidation and thereby reduce the soot mass fraction [36].

3.6 HFO, MDO, and D-EN590

The use of various types of sulfur fuel oils, such as HFO, MDO, and D-EN590, has been shown to cause significant changes in the in-cylinder combustion process. These changes are mainly attributable to the properties of the fuels, including the density, sulfur content, and chemical composition. The density of HFO is comparatively higher than those of MDO and D-EN590 and therefore result in slow combustion rates and lower in-cylinder temperatures and pressures than those of the other two fuels, which leads to high soot mass fractions, as shown in the **Figure 8**. In addition, the high viscosity of HFO, compared with those of MDO and D-EN590, requires proper heating; otherwise, high viscosity leads to poor atomization, which deteriorates the combustion process, reduces the thermal efficiency of the engine, and compromises exhaust gas emissions. In conclusion, the lower sulfur content in both the MDO and D-EN590 fuels results in the lower production of SO_x, enhanced cleaner combustion, and alleviation of environmental crises. Therefore, the above analysis proves that the need to switch to low-sulfur fuels impacts not only SO_x emissions but also other in-cylinder combustion performance characteristics and exhaust gas emissions.

4. Conclusion

This study investigated the combined effects of fuel injection timing and the use of various types of sulfur fuel oils from HFO,

MDO, and D-EN590, which is an ultra-low-sulfur fuel oil, in enhancing the combustion performance and reducing the emission characteristics of diesel engines. The main conclusions are as follows.

- 1) The use of D-EN590 led to a slight increase in the in-cylinder pressure and temperature, which promoted enhanced combustion efficiency, compared with MDO and HFO.
- 2) The use of D-EN590 resulted in a greater decrease in soot emissions than in NO_x and CO₂ emissions. Compared with HFO, D-EN590 significantly reduced exhaust emissions. However, a few insignificant differences were observed between the MDO and D-EN590 groups.
- 3) Early fuel injection timing at 15CAD and 10CAD BTDC led to an increase in the ignition delay, and hence an increase in the percentage of premixed combustion, resulting in a higher peak in-cylinder pressure and temperature. Therefore, the optimal SOI based on in-cylinder pressure and temperature is between 15CAD and 10CAD BTDC.
- 4) Furthermore, for D-EN590 at 15CAD and 10CAD BTDC, the NO emissions are lower by 0.044% and 0.043%, respectively, whereas the NO emissions are higher by 0.055% at 5 CAD BTDC. These results indicate that 10CAD is the optimal SOI.
- 5) The 15CAD BTDC SOI of D-EN590 had the lowest CO₂ emissions of 12.70%, compared with 10CAD and 5CAD BTDC, which showed emissions of 13.43% and 14.28%, respectively. These results indicate that 15CAD is the optimal SOI.
- 6) The 5CAD BTDC showed a slightly high soot reduction, but considering other combustion characteristics, such as pressure and temperature, it does not offer an optimal SOI.

Therefore, based on the aforementioned conclusions on combustion characteristics and emissions, the need to switch to low-sulfur content fuels and effective fuel injection timing between 15CAD and 10CAD BTDC is essential for enhancing diesel engine combustion performance and reducing emissions.

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Author Contributions

Conceptualization, Methodology, Software, Validation, Formal Analysis, Visualization, Writing-Original Draft Preparation, Writing-Review & Editing, N. I. Ahmed; Formal Analysis, Final review, J. K. Kim; Conceptualization, Formal analysis, K. W. Chun; Methodology, Project Administration, J. H. Choi; Conceptualization, Supervision, Project Administration, Writing-Review & Editing, W. J. Lee.

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