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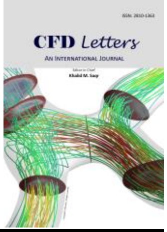

Numerical Investigation of Split Injection Strategies and Injector Nozzle Bore Influence on Combustion and Emissions

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ABSTRACT

The power generation mainly depends on fossil diesel fuel, the primary source of harmful emissions and global warming. Therefore, the researcher aims to explore alternative fuels that got greater attention in compression ignition engines. The commercial Diesel-RK software simulates the current numerical study of diesel engine direct injection with speed engine 1500 rpm, compression ratio 17.5, single cylinder, and naturally aspirated. In the basis of the parliamentary research, the literature did not investigate the influence of combustion and emission characteristics of compression ignition engines by using various double and trouble injection strategies along with different injector nozzle bore sizes. Also, the initiative was undertaken to study the effect of the different diesel-biodiesel blends ratio studied, SP20 (80% diesel+20% spirulina), Sp40(60% diesel+40% spirulina), and Sp100(0% diesel +100% spirulina) While the scope of the gap expanded to include a comparison of results with baseline diesel fuel. The results show that MPR increased by 4.2%, maximum gas temperature increased by 8.9%, ignition delay increased by 7.9%, maximum heat release rate decreased by 9.5%, NO_x decreased by 7.8%, CO_2 decreased by 3.9%, and particularly matter emissions decreased by 6.3% were compared to the double injection scheme , at 0.2 mm(INB).

1. Introduction

The researchers on alternative fuels for internal combustion (IC) engines have shown that despite the fuel replacement function, many non-fossil fuels also have good properties that lead to lower pollutants [1]. This is why it is necessary to bridge the knowledge gap in understanding the combustion process of different alternative fuels. This will enable further evaluation of the environmental impacts for future use. The fuel's chemical properties, such as; cetane number, volatility, and oxygen content [2], play a vital role in the combustion process and emissions. So, there are effective options to attain lower emissions and higher efficiency by taking advantage of the chemical and physical properties of multi-fuel. This has the potential to enhance combustion quality, performance, and emissions. Biodiesel as a mixture of different long-alkyl easter has been attracting attention as prominent alternative fuel due to greenhouse gas reduction, secure future energy needs,

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less carbon content, as well as about 10-12% oxygen molecular weight contained in the biodiesel composition molecules [3]. The oxygen content of the fuel plays a significant role in restricting the generation of smoke precursors by reducing the local equivalent temperature during the combustion process. A blend of oxygenate biodiesel with baseline diesel fuel significantly reduces soot emissions. This is a clear indicator of possible improvement in combustion quality. Biodiesel can also be described as having a high-octane number, reducing combustion speed and prolonging the ignition delay (ID) period [4]. However, the ignition delay (ID) period was considered an influential factor influencing fuel-air mixture development. Biodiesel has other factors, such as dilution, boiling point (less aromatic hydrocarbons) [5], and higher viscosity [6] which play paramount roles in applications requiring high biodiesel-diesel blending ratios.

Many research institutions globally have conducted several researchers on combustion mechanics, emissions, and diesel engine performance. The results showed that PM [7], hydrocarbon (HC) [3], and carbon monoxide (CO) [8] were reduced in the case of utilizing biodiesel as an alternative to conventional fossil fuels in internal combustion (IC) engines. Some of the disadvantages of biodiesel are the lower energy content, higher viscosity, high cloud, and high oxide of nitrogen (NO_x) emissions when compared with conventional diesel fuel in internal combustion (IC) engines [9]. Oxygen in biodiesel composition molecules has a key role in increasing NO_x emissions when burning in internal combustion (IC) engines up to 15% [10]. According to the zeldovich mechanism, oxides of nitrogen ($NO_{x)}$ formation appear prominently at higher local temperatures, which mainly depends on the local oxygen concentration within the ignition zone.

Generally, biodiesel products are extracted from animal fats, vegetable oils, recycled cooking gases, and wasted plastics. American Society defines biodiesel's chemical and physical properties for Testing and Materials (ASTM), the chemical composition of biodiesel consists of alkyl esters (ethyl or propyl, methyl) for long-chain fatty acids, which are derived from various animal fats as well as vegetable oils [11]. The essential biodiesel sources are either edible or non-edible oils [12]. In general, edible oils can be considered first-generation and non-edible oils are ascribed to second-generation oils. There is no desire to produce biodiesel from edible oils due to the expected shortage of global cereal stocks [13]. It should be clear to every researcher that they should focus on biodiesel sources that do not compete with food crops.

These raw materials are represented by non-edible oils such as sterculia foetida, jatropha curcas, karanja, pongamia, jojoba, cottonseed, algae and sea mango, Mahua, Neem [14]. Rajak *et al.,* [15] proposed an experimental study on the effects of the turpentine oil, jatropha, with their blends in diesel engine configuration. They reported brake thermal, NO_x, HC, and smoke emission reduced. Meanwhile, CO_2 levels increased with blends of biodiesel at full load operation. Rashed *et al.,* [16] An experimental investigation of diesel engine characteristics using moringa oil, jatropha, and palm as operation testing fuel. They also made a comparative study between them. According to their report, on all biodiesel fuels, the increase was evident in brake specific fuel consumption (BSFC), CO, and gas emissions, while there was a reduction in brake force.

Al-lwayzy *et al.,* [17] conducted an experimental study to evaluate diesel engines' performance and exhaust emissions features using spirulina microalgae as an alternative renewable fuel. They noticed a decrease in the CO_2, NO_x, gas exhaust temperature (GET), while they recorded a clear increase in O_2 and BSFC. In the account of the best alternative fuel when it comes from the biodiesel class, biodiesel extracted from microalgae has a standing potential substation for conventual fossil fuels due to its biodegradability and non-toxicity [18]. The hydrocarbon-rich microalgae have been discussed in terms of its utility as alternative fuels due to their large lipid storage and detoxification ability. Even across different microalgae species, at least 15-50% of their dry weight is stored as lipids and ready for extraction. In light of the ever-increasing concern about global warming and the fuel depletion crisis, as evidenced by a US Department of energy report [19], microalgae have gained attention and potential due to higher energy conversion efficiency in the past few decades. On the other hand, the highly quantitative analyses like water footprint, the nutrient balance to microalgae biofuels, and the production energy expenditures raise potential concerns about economic energy efficiency.

Recent reviews reported that the issue of economic and technological bottlenecks is being resolved using biofuels based on microalgae as an excellent alternative to petroleum. Realizing the potential of microalgae biodiesel as an alternative fuel suggests convenient and practical solutions to diesel engine behaviors while in operation. Besides, there are particular feasibility concerns with experimental and numerical studies for characterizing the behavior of this fuel. Hence it is worth of investigating these phenomena deeply. In accordance with previous works, the methodology presented in this paper can investigate the diesel engine behaviors when in operation with a high blending ratio of microalgae biodiesel. Thus, this study investigates the performance, combustion and emissions behaviors of compression ignition (CI) engines operated with spirulina microalgae-based diesel fuel for split injection scheme technique and injection pressure.

The fuel injector nozzle is one of the most essential parts of the diesel engine. The decomposition mechanism for spraying fuel through the injector nozzle is thought to be the theory of aerodynamic atomization. The geometry of the fuel injector nozzle and fuel flow characteristics strictly affects the development of fuel decomposition within the combustion chamber, performance, combustion process mechanics, and harmful emissions from compression ignition (CI) engines. The fuel injection systems in diesel engines aim to produce large amounts of atomized fuel with high pressure to enhance the shortage of fuel penetration, in turn promoting further evaporation in a very short time to achieve favorable conditions for the best combustion process [20]. In modern combustion engines, fuel is injected at high pressure, leading to the fuel cavity and adhered to the exit and inlet nozzle aperture. Concerning the discharge coefficient, the cavity reduces the flow efficiency, as well as the collapse of the cavity bubble, which mainly erodes the injector nozzle material [21].

Kumar *et al.,* [22] conducted an experimental study to investigate the effect of the nozzle orifice diameter on the combustion, performance, and emission characteristics of diesel engines fueled with biodiesel blends based on diesel fuel. They reported that the smaller nozzle orifice diameter leads to a shortening combustion duration due to the improved air-fuel quality, vaporization, and atomization. But the only disadvantage was the increased NO_x levels due to higher local temperature within the combustion zone.

Diesel-RK is one of the essential specialized software used for thermodynamic analysis of internal combustion (CI) engines by many researchers to investigate in depth the different characteristics such as: heat release rate (HRR), ignition delay (ID), carbon monoxide (CO), carbon dioxide (CO_2), oxide of nitrogen (NO_x) , combustion cylinder temperature and the peak of pressure. Diesel-RK has emerged as a very reliable and convenient numerical modelling tool for different fuels such as waste oil, animal fats, alcohol biodiesel, edible and non-edible, which operates in different conditions. Validation of the results obtained with experimental data that proved the numerical solver provides very close results is practiced. Therefore, Diesel-RK is an appropriate and confident tool given the results' accuracy and low time spent obtaining data.

The reasons mentioned earlier led the researchers in the internal combustion (IC) engine field to take the simulation method approach to analyze diesel engine characteristics. Al-Dawody *et al.,* [23] conducted an optimization study on the NO_x emission in diesel engines using biodiesel (Soybean methyl ester) as an alternative fuel.

The simulation was done by Diesel-RK software; the results were validated with an experimental study at identical operating conditions. They showed that NO_x emissions were reduced by up 50.26%.

Datta & Mandal [24] performed a numerical analysis to simulate internal combustion (IC) engine using alcohol as a supplement fuel in baseline diesel. The authors utilized Diesel-RK software as a tool to accomplish the analytical study. The preliminary operation conditions were injection timing at 23° before TDC, and engine speed 1500 rpm kept constant. Their results showed a noticeable reduction in almost all emissions. Also, the authors indicated that NO_x emission levels in biodiesel blends are higher than in baseline diesel fuel. Rajak *et al.,* [25] conducted an experimental and numerical simulation study to investigate the different characteristics of emissions from various categories of fuels. These fuels include non-edible and edible vegetable oils in diesel engine single-cylinder, while constant advance injection time and engine speed were maintained. The results obtained by Diesel-RK software showed reduced emissions for edible, non-edible, animal fats and grease oil by 3.19%, 2.8%, 1.8%, and 3.22%, respectively. From previous studies, Diesel-RK software can be evaluated as a reliable and convenient tool for the numerical analysis and investigation of a large variety of fuels operating in different conditions.

According to the previous studies in the literature, to comply with the global emission stringent, the diesel engine must be modified to overcome the environmental crises and fossil fuel depletion and boost the compression ignition (CI)engine in terms of combustion and emissions characteristics. Agarwal *et al.,* [26] reported that the smaller injector nozzle bore plays a vital role in reducing the emissions contents. In addition, split injection strategies were carried out as stated in [27], which conducted an experimental study to reduce the emission to attain the lowest emissions limit. Therefore, the current research takes advantage of the combined efficient ways of the injector nozzle bore, split injection strategy and use of the biodiesel fuel alternative, which were in limited literature and represented the gap of study.

2. Methodology

The present research performed a numerical investigation to study the integrated behavior of spirulina biodiesel blends (Diesel, Sp20, Sp40, Sp100), split injection strategy, and injector nozzle bore effect on the engine performance, combustion characteristics and compressed ignition (CI) engines emissions. The study then compared the outcomes with the baseline diesel. Figure1 shows the cases considered in the present study. A numerical simulation study of the diesel engine was carried out using the Diesel-RK. The numerical model is a perspective of a diesel engine developed for six cylinders, direct injection, four-stroke, and natural spread. Table 2 shows engine specifications in detail. While the determinants mentioned above have a significant influence on the engine mechanism behavior. Over this entire simulation, the compression ratio was 17.5:1, engine speed 1500 and early injection timing 23^{o} before TDC, also, the engine runs under a full load state of 3.7 kW. Adjusting nozzle orifice diameters was done within the range of 0.20mm to 0.28mm step 0.02mm along with two different strategies of injection scheme (double and triple). Due to the importance and feasibility of these parameters in studying the environmental performance of internal combustion (IC) engines, the investigation has been limited to these highly influential parameters.

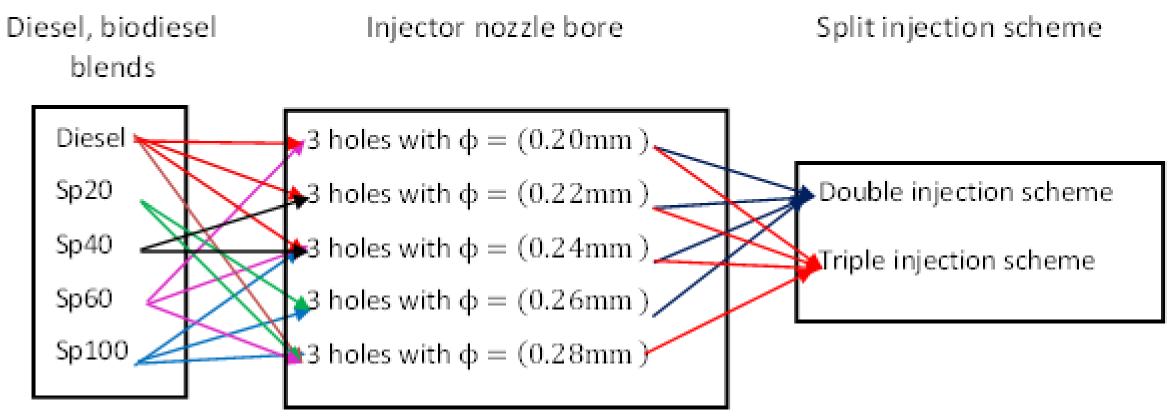


Fig. 1. Cases considered for the research analysis

2.1 Physical and Chemical Properties

The performance, combustion and emissions characteristics depended on the chemical properties of the test fuel. The diesel, spirulina biodiesel chemical and physical properties are presented as shown in Table 1, which were collected from a reliable previous research paper in the literature [15].

Table 1
Properties of pure diesel and spirulina biodiesel blends

	Diesel	Spirulina biodiesel blends			
Properties	D100	Sp20	Sp40	Sp60	Sp100
Density (kg/m3) at 15 ˚C	830	836.5	842.7	848.9	861
Viscosity (mm2/s) at 40 ˚C	2.9	3.26	3.67	4.2	5.26
Low heating value of fuel (MJ/kg	42.5	42.2	41.88	41.58	41.0
Cetane number	45-55	48.88	49.7	50.5	52.2
Flash point (°C)	76	86.4	96.8	107.3	128.2
C (%)	87.0	84.9	83.08	81.2	77.46
H (%)	12.6	12.51	12.4	12.36	12.21
O (%)	0.4	2.48	4.47	6.45	10.33

2.2 Diesel-RK Model

Diesel-RK software is a thermodynamic model used to visualize the computational study to analyze combustion, performance, and emissions characteristics based on the first thermodynamic law. Furthermore, the software has the ability to simulate advanced diesel engines with various fuel blends, as well as multi-factor optimization. The test fuel's properties are required to calculate the temperature, pressure, heat release rate and other parameters. These were calculated per crank angle degree or time step. Engine friction and heat release rate are considered with respect to pose quasi-experimental correlations derived from experimental study results.

Diesel-RK models can be classified into two groups: single-zone and multi-zone. A multi-zone model is computational fluid dynamics applied to simulate complex combustion systems and advanced diesel sprayers based on numerical calculation of momentum, mass, and energy equation. This was done to follow the flame front propagation through the combustion chamber. Furthermore, it is possible to visualize fuel spray and the interaction between near-wall flows due to adjacent sprays in the animation format. It equally considers the injection shape profile, like the split injection scheme; drop size; spray direction within the combustion chamber. Different diesel engine characteristics, such as ignition delay, combustion duration, carbon dioxide, carbon monoxide, etc., were critically analyzed for various operating conditions and different fuels.

2.2.1 Spray evolution model

The diffuse fuel injection through the combustion chamber during the spraying period used in the Diesel-RK model is shown in Figure 2. During a short period, the instantaneous position and speed of fuel propagation from the injector to the spray tip is described in Eq. (1).

$$(\frac{U}{U_o})^{3/2} = 1 - \frac{l}{l_m} \tag{1}$$

where l is the instant distance between elementary fuel mass (EFM) and the injector nozzle. $\mathrm{U} = \mathrm{dl}/\mathrm{d}_{\tau\mathrm{k}}$ where U is the instant velocity for elementary fuel mass (EFM).

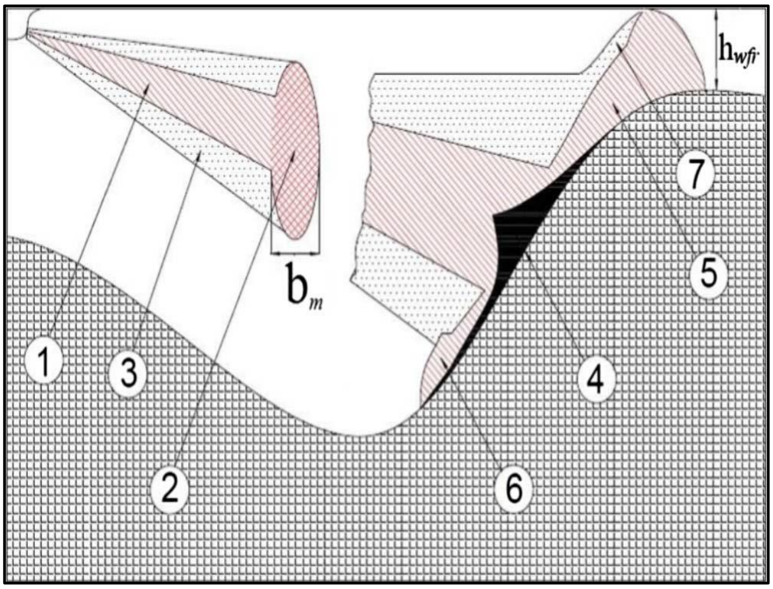


Fig. 2. Various zones of diesel spray [27]

2.2.2 Formation of NO_x model

Zeldovich mechanism chain is a chemical mechanism used to calculate NO_x emissions produced in the combustion chamber on every step and solve eighteen species. The overall system contains three material balances for the equilibrium equation and Dalton equation of partial pressure described by Eq. (2) to Eq. (7) [28].

$$O_2 \leftrightarrow 2O \tag{2}$$

$$N_2 + O \leftrightarrow NO + O \tag{3}$$

$$N + O_2 \leftrightarrow NO + O \tag{4}$$

$$\frac{d[NO]}{d\Theta} = \frac{P\times2.333\times10^{-7}.e^{-\frac{38020}{T_b}}[N_2\,]_e.[O]_e.\{1-([NO])/[NO]_e)^2\}}{R.T_b.\left(1+\frac{2365}{T_b}.e^{\frac{2365}{T_b}}.\frac{[NO]}{[NO]_e}\right)}.\frac{1}{\omega} \tag{5}$$

In cylinder of diesel engine the NO concentration given by :

$$r_{NO_c} = r_{NO} r_{bc} \tag{6}$$

Calculate the NO emission in g/kWh expressed as :

$$e_{NO} = \frac{30 \times r_{NO} \times M_{bg}}{L_C \times \eta_M} \tag{7}$$

2.2.3 Soot formation model

Soot is formed inside the diesel engine due to the large environmental pollution of unburnt hydrocarbons. Hatridge smoke was used in order to calculate the soot formation level , which it described using Eq. (8).To calculate the soot formation in combustion zone, the model is given by Eq. (9) [28] .

$$Hartridge = 100\{1 - 0.9545\exp(-2.4226)\,[C])\} \tag{8}$$

$$\left(\frac{d[C]}{dt}\right)_k = 0.004\frac{q_c}{V}\frac{dx}{dt} \tag{9}$$

2.3 Diesel -RK Tools Validation

In order to reliable the results extracted from Diesel-RK software model, the results were compared with the experimental results [28] as shown in Figure 3, Figure 4 and Figure 5, to validate the numerical software model. The boundary conditions proposed for the present numerical work are identical to the experimental work, as shown in Table 2. For the present study, the author proceeded with validation between numerical and experimental results for three parameters (in-cylinder pressure, heat release rate, and NO_x emissions) as shown in Figure 3, Figure 4 and Figure 5. The experimental test engine was fueled with baseline diesel, single cylinder, cooperation ratio (17.5) and 1500 rpm speed engine. The figure clearly shows that the results obtained from the current numerical model and the experimental of the specified test engine are in good agreement. Therefore, the proposed numerical tools are reliable and can be used to simulate diesel engines and analyses combustion and emissions characteristics.

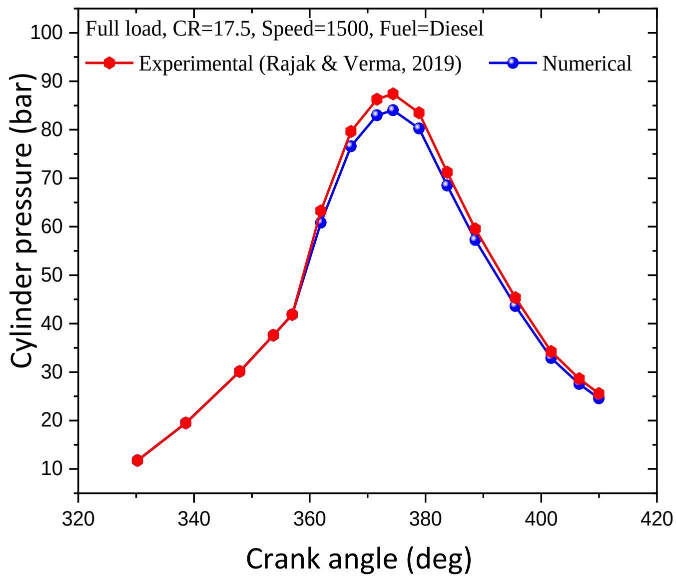


Fig. 3. Cylinder pressure variation with crank angle

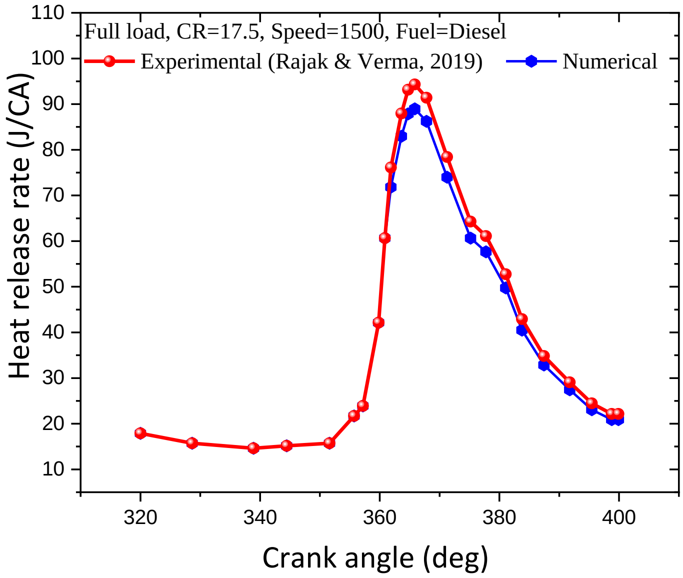


Fig. 4. Heat release rate variation versus crank angle

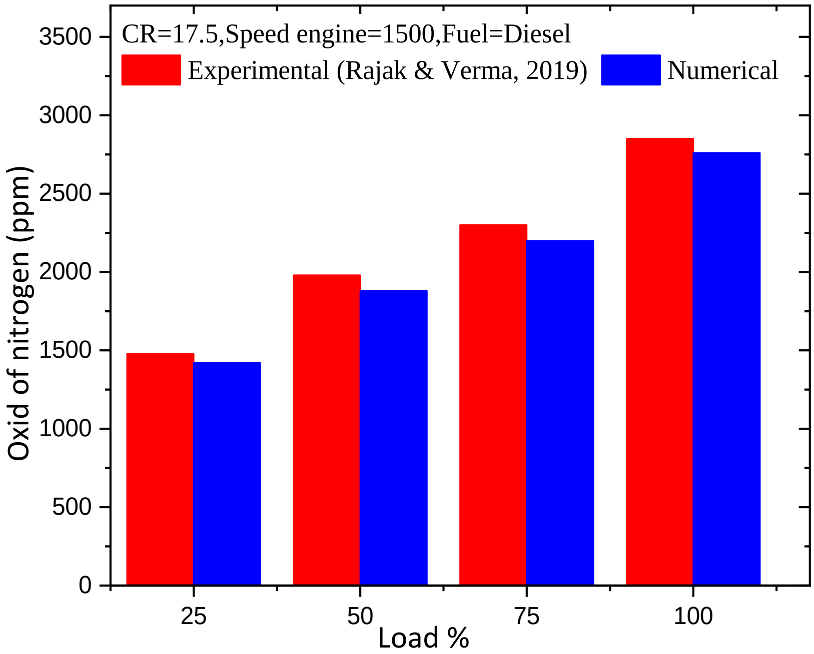


Fig. 5. NO_x emission variations versus engine load

Table 2
Initial boundary condition for numerical analyses

Parameter	Value
Bore	80 mm
Stroke	110 mm
Compression ration	17.5:1
Initial pressure	1.0 bar
Speed engine	1500 rpm
Initial temperature	300 K
Piston temperature	530 K
Liner temperature	420 K
Head temperature	500 K
Fuel injection pressure	220 bar
Fuel injection timing	23.5° before TDC
Inlet valve open	4.5° before TDC
Inlet valve closed	35.5° after TDC
Outlet valve open	35.5° after TDC
Outlet valve closed	4.5° before TDC
Fuel spray angle	70° degree
Maximum power	3.7 kW

The error deviation (ED) analyses between the experimental and numerical results study calculated based on the Eq. (10) as follows.

$$ED = \left(\frac{X_v - Y_v}{X_v}\right) \times 100 \quad (10)$$

Where the X_v value represents the higher-level value, while Y_v represent the lower-level value.

The results extracted from Diesel-RK software was qualitatively matching well for the three-parameter of experimental work, while the divergence ratio was in an acceptable range. The error deviation (ED)for the in-cylinder pressure, HRR, and NOx were obtained 3.72%, 4.24% and 2.44%, respectively, as shown in Table 3.

Table 3
Experimental and numerical results at full load condition for tool validation

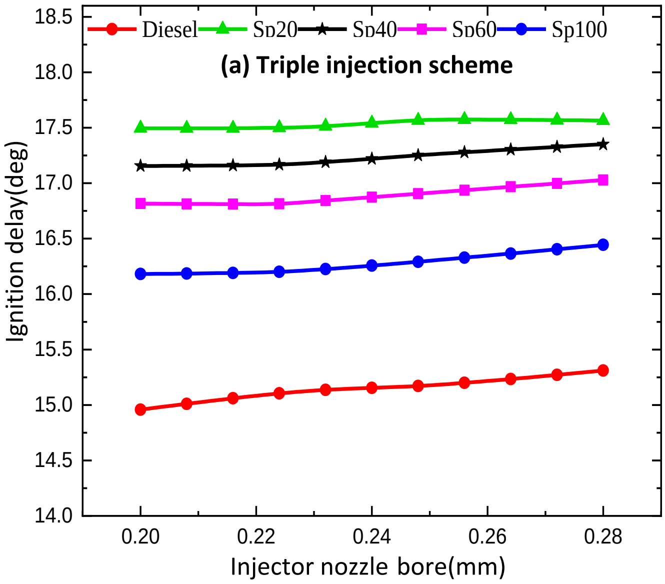
Parameter	Experiential	Numerical	ED/ (%)
In-cylinder pressure (bar)	88.7	85.4	3.72
heat release rate (J/CA)	95.2	91.5	4.24
NO_x (g/kWh)	2849	2780	2.44

3. Result and Discussion

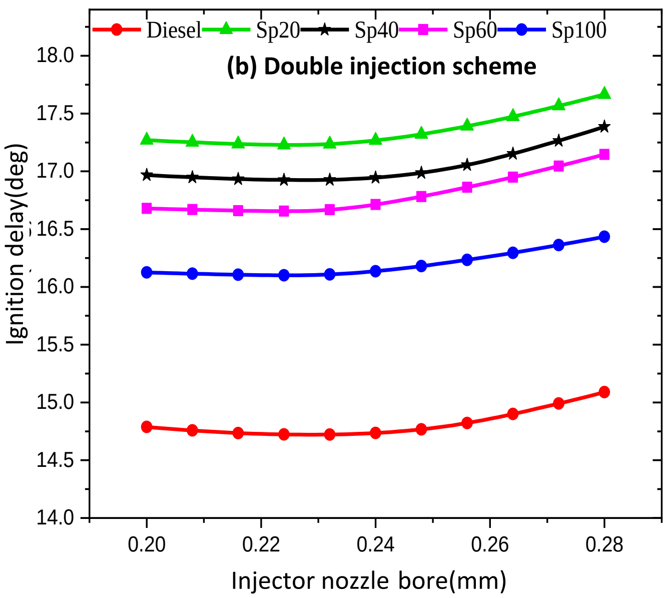
3.1 Ignition Delay

The period measured in crank angle degree between the beginning of fuel injection and the starting of igniting the fuel was defined as the ignition delay (ID). Defiantly, ignition delay (ID) was considered a main distractive parameter responsible for the start of the combustion process. Longer ignition delay causes more fuel to be injected, leading to accumulation within the premixed combustion phase. This contributes to an improved air-fuel mixture. The ignition delay (ID) depends on the cylinder pressure, temperature, and fuel chemistry. Ignition delay (ID) for blends of spirulina biodiesel exceeded that observed when diesel was the fuel. This result was consistent with the report of [28]. With a double injection scheme, the highest ignition delay value was 17.54^o deg for Sp20 at 25 mm (INB). The result indicated that the ignition delay of the main combustion phase was prolonged as the ratio of spirulina increased. This is because spirulina biodiesel viscosity and boiling temperature are higher than the baseline diesel fuel. At the same time, the volatility of diesel fuel is lower than that of spirulina biodiesel blends. These indicators show that the blended fuels with lower spirulina ratios are easier to break up and evaporate [29]. So, adding spirulina increases the ignition delay of the main injection compared with pure diesel fuel. This led to a leaner air-fuel mixture result and reduced the combustion temperature and NO_x emission level.

To investigate the influence of the split injection scheme, injector nozzles bore (INB) and types of fuel on ID, the average values were calculated and compared in Figure 6(a) and Figure 6(b). The result shows the relationship between ignition delay versus different injector nozzles bore (INB) with double and triple injection schema for diesel and blends of spirulina biodiesel. At 0.2 mm (INB) and double injection scheme, ignition delay was found as 14.95^o deg, 17.49^o deg, 17.15^o deg, 16.81^o deg and 16.18^o deg Figure 6(a). While it was 14.98^o deg, 17.67^o deg, 16.968^o deg, 16.97^o deg and 16.12^o deg for diesel corresponding to Sp20, Sp40, Sp60 and Sp100, respectively, with triple injection scheme Figure 6(b). The abovementioned results showed that ID with triple injection scheme was higher than with double injection scheme by 1.137 %, 1.292 %, 1.096 %, 0.821 % and 0.352 % for Diesel, Sp20, Sp40, Sp60 and Sp100, respectively, at 0.2 mm (INB). The Injection event is formed by the two and three identical injection pulses for double and triple injection schemes, respectively. One of the main reasons responsible for the prolonged ID duration is the 2nd and 3rd pulses. The ignition is mainly initiated by the fuel injected through 1st pulse at the end of the delay period. In contrast, lately, the 2nd and 3rd pulses of fuel injected impact the ignition process [30]. Thus, the ID increase corresponds to the increase in split injections. This result was consistent with the report of the researchers [31].



(a)



(b)

Fig. 6. Ignition delay versus injector nozzle bore (a) Triple injection scheme (b) Double injection scheme

3.2 Maximum Pressure Rate

The MPR in compression ignition engines depends mainly on the quantity of fuel burnt within a premixed phase. As well as, many factors like as, viscosity and cetane number play a significant influence in the MPR variations. The variations of the MPR for the diesel-biodiesel blend are shown in Figure 7, as seen in the paragraphs, all the blends containing diesel-biodiesel exhibited a lower MPR compared to baseline diesel fuel. This can be explained as, biodiesel having a higher ID period than baseline diesel so that a massive amount of accumulated fuel is in standup to initiate the ignition during the premixed phase. So, a longer ID period extends the start of combustion to TDC and the beginning of expansion stroke, resulting in reduced in-cylinder pressure [32]. In addition, the slight difference between spirulina biodiesel and baseline diesel fuel in properties and chemical composition gives approved biodiesel a realistic alternative fuel to diesel. Biodiesel has a higher oxygen content in the fuel molecules and considers an oxygenate fuel, while oxygen influences the combustion process in two ways. First, the higher oxygen content level in the biodiesel cooled the fuel mixture in the premixed phase, leading to a long ID period, which gives the air: fuel mixture the time to more mixing rate. Second, the air: fuel mixture provided the oxygen required to accelerate the oxidation process. Based on the theory mentioned above,

Now, the effect of the injector nozzle bore (INB) on the MPR is shown in Figure 7, it was obvious that decreasing trend of the MPR with increasing INB. The MPR was (76.53 bar at 0.20 mm INB), (75.90 bar at 0.22 mm INB), (74.86 bar at 0.24 mm INB), (73.32 bar at 0.26 mm INB) , and (71.83 bar at 0.28 mm INB) for bassline diesel fuel and double injection scheme. The size of fuel particles increases with increased INB, which leads to lousy vaporization transaction, poor atomization, slower mixing rate, and shorter penetration. All these conditions negatively impact the combustion process and reduce mean gas temperature (MGT) [22]. The effect of the split injection strategy on MPR is discussed in this section. As seen in Figure 7, the triple injection scheme has a higher MPR than the double injection scheme for all test fuels. This is attributed to the longer ignition delay (ID) period than the double injection scheme, which gives the advantage of improving and bringing the charge to better mixing [29]. This leads to an increase the combustion efficiency and higher MPR. Another explanation could be proposed: the extended ID period attributed to the triple injection scheme contributed to the combustion process near the TDC. So that the first steps of burnt gas expansion occur in the tiny combustion zone; hence, a greater MPR can be attained with the triple injection scheme.

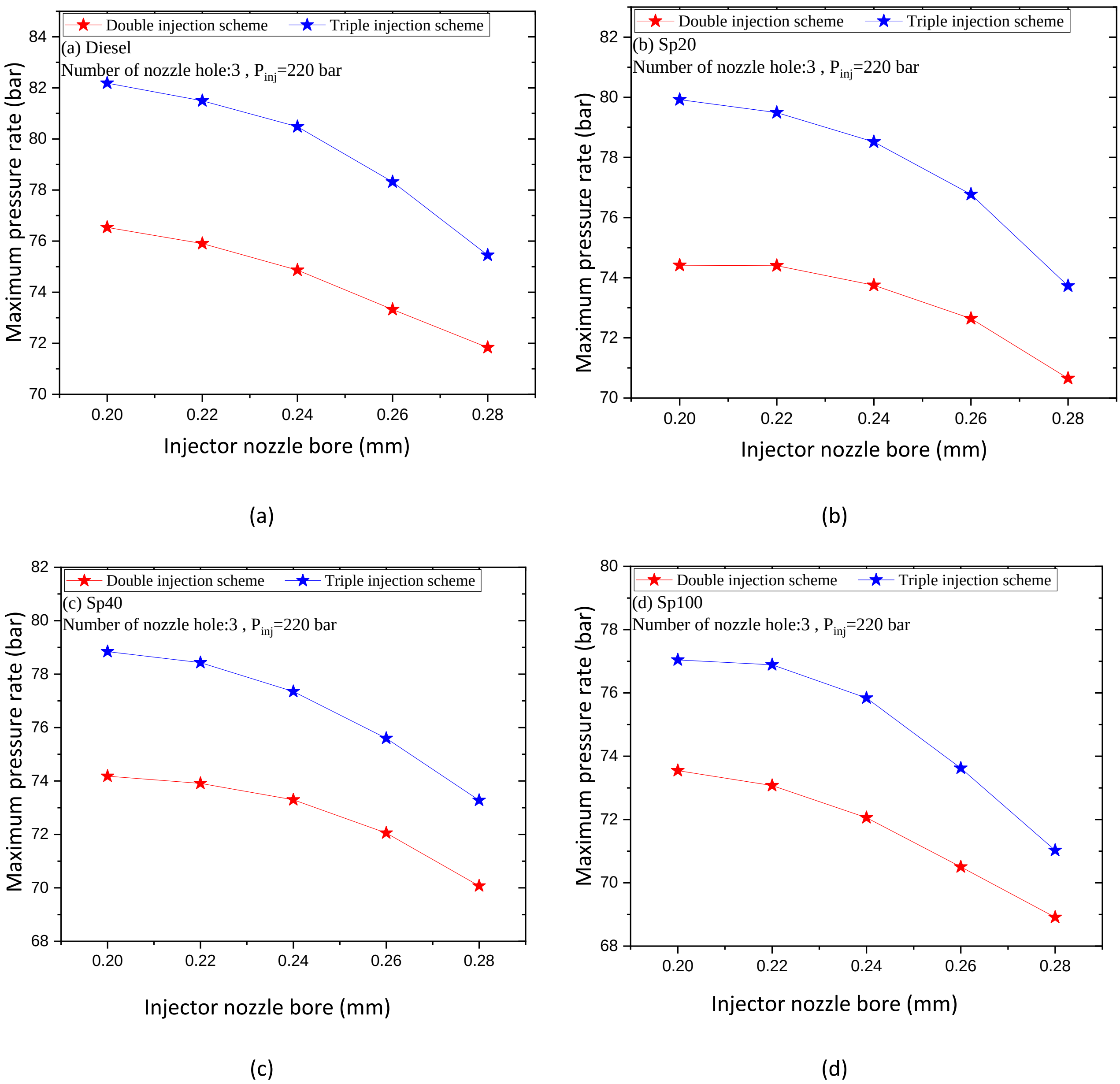
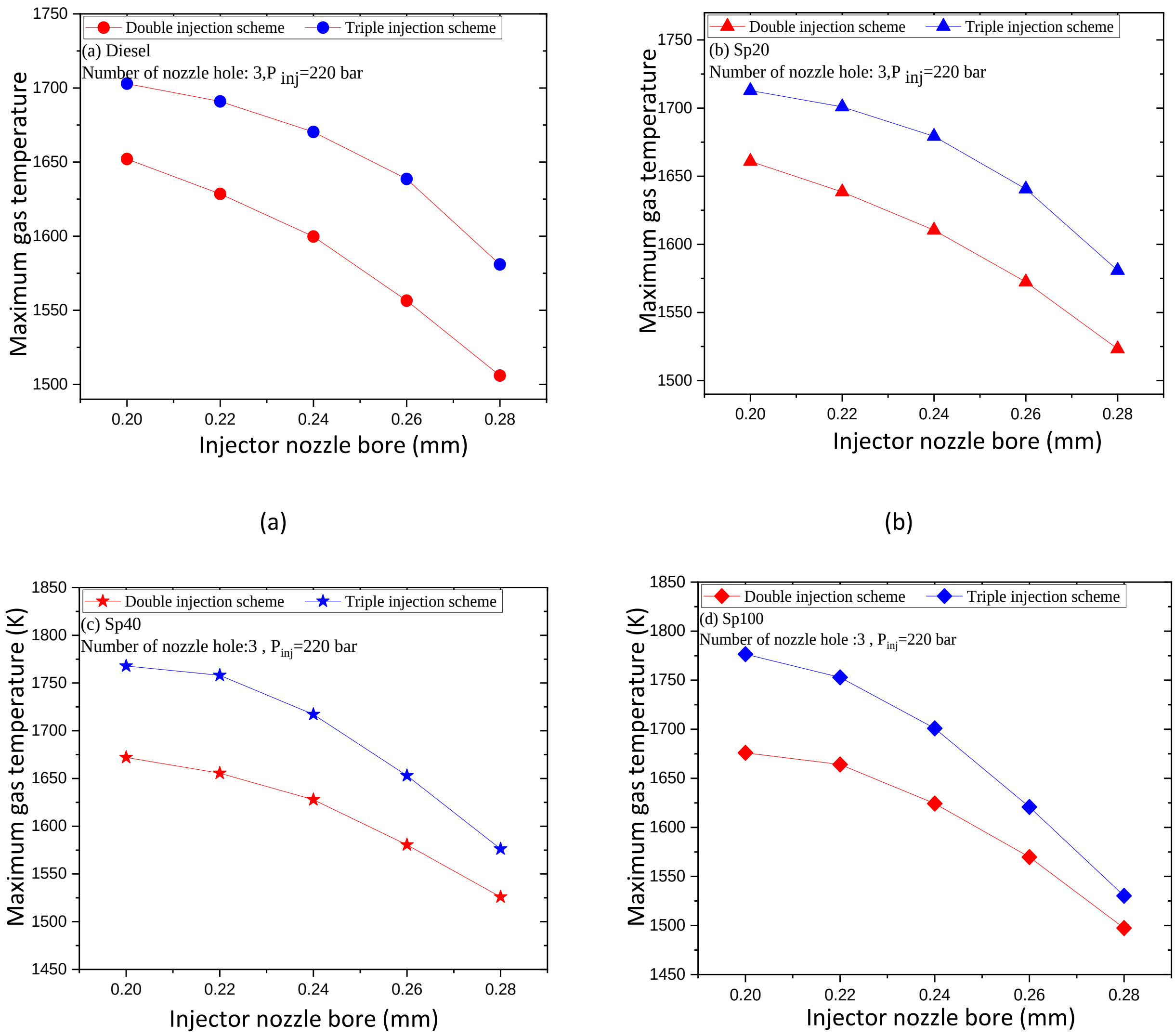


Fig. 7. Maximum pressure rate versus injector nozzle bore (a) Diesel (b) Sp20 (c) Sp40 (d) Sp100

3.3 Maximum Gas Temperature

The results show the maximum gas temperature (MGT) for the diesel-biodiesel blends affected by the split injection strategy and injector nozzle bore properly are shown in Figure 8. Appreciated for all the graphs, the MGT increased as the spirulina biodiesel ratio increased in the blends. The main differences between pure diesel fuel and biodiesel are related to the low heating value and C/H/O compositions. Biodiesel has a higher oxygen amount than diesel, which increases the ignition delay (ID) period. Moreover, biodiesel's cetane number (CN) is more elevated than diesel, which assumes ID should be shorter [33]. Still, the higher viscosity value of biodiesel gave inverse ID prediction. However, the effect of the higher CN value is more significant on the MGT value [33], inferred from the distinct differences shown in Figure 8. The higher value of the CN means a high amount of fuel produces the same energy, resulting in a massive quantity of fuel cumulated within the pre-ignition phase [29]. Hence, the combined effect of the two parameters, higher CN value and oxygen availability, contributes to decreased premixed phase duration. This is the main reason to help extend the burning of the air-fuel mixture till the last stage of the combustion phases within the compression stroke and provide more released heat. The CN and main fuel properties indexed with the different proportions of biodiesel are shown in Table. The same results were noticed in a study conducted by Yusoff *et al.,* [34], the author conducted an experimental study with different biodiesel fuel. The EGT for the different diesel-biodiesel blends slightly increased with increased biodiesel ratio in blends also varied distractively compared to the bassline diesel fuel. The EGT found (1670 K: Diesel), (1675 K: Sp20), (1680 K: Sp40) and (1690 K: Sp100), respectively, for the double injection scheme and at 0.2 mm (INB). Also, Figure 8 shows the variation of the MGT regarding various injector nozzle bore INB value, while each nozzle has 3 holes. It was observed that the decreased trend to MGT, along with increased INB for all the test fuel was due to poor atomization, which led to lousy air: fuel amalgamation resulted in an incomplete combustion process. The size of the atomized fuel particle increased with increased INB [29], inferred to be an unappropriated mixing condition.

Moreover, Figure 8. shows the spilt injection strategy's influence on the MGT; this approach can be conducted by dividing the single direct injection into equal two or three consecutive injections with constant dwell timing and the start of injection for one cycle of a combustion engine. Or in other words, the main injection splits into two and three equal injection pulses. One of the split injection strategy advantages is enabling mixture air: fuel to improve charge homogenate.



(a) (b)

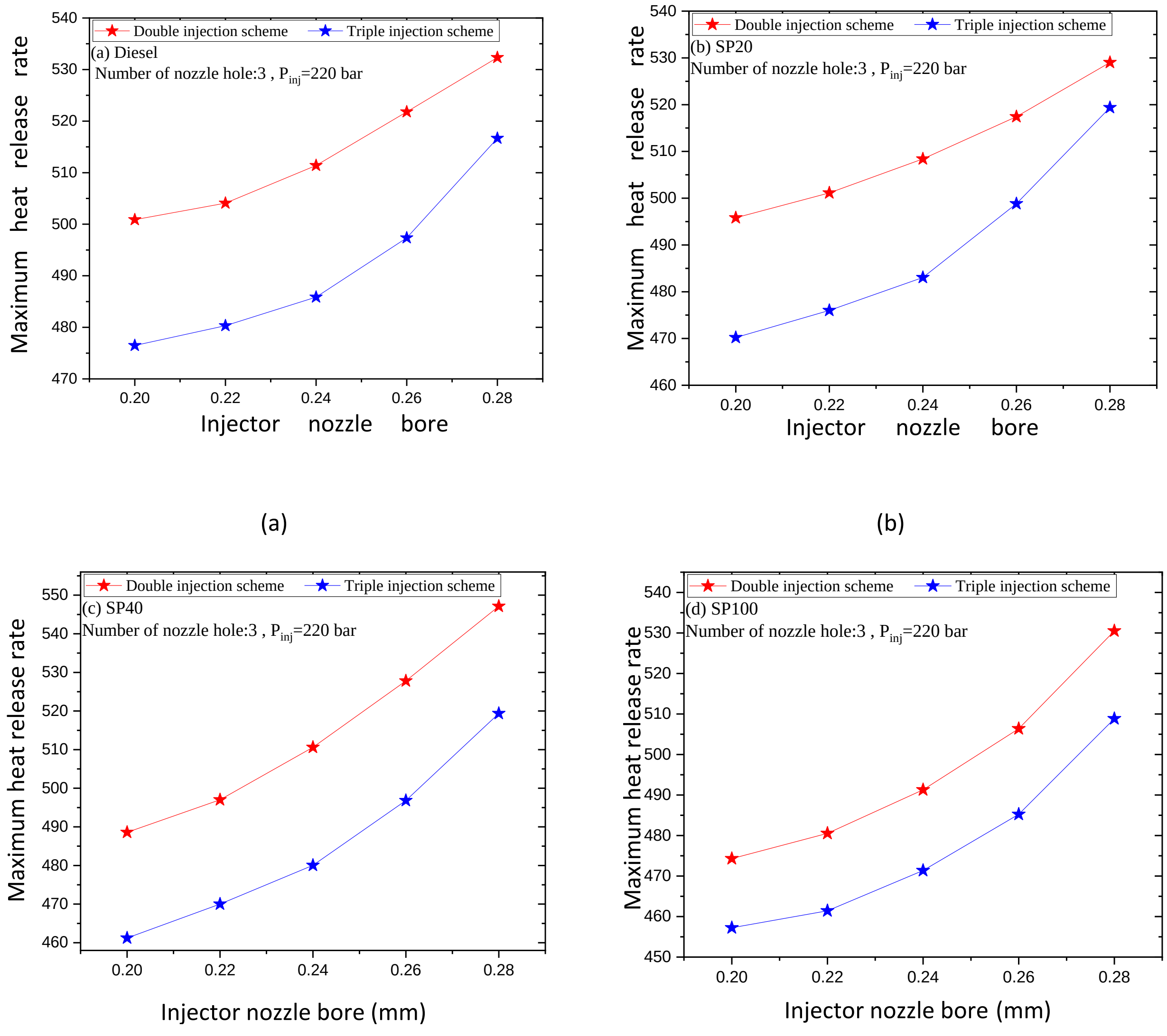
(c) (d)

Fig. 8. Maximum gas temperature versus injector nozzle bore (a) Diesel (b) Sp20 (c) Sp40 (d) Sp100

3.4 Maximum Heat Release Rate

The spirulina biodiesel fuel has a lower low heating value (LHV) and higher cetane number (CN) than baseline diesel fuel, as indexed in Table 1. So, the expectation is that an increased spirulina ratio in the blends leads to reducing the heating value and increasing the CN of the diesel-biodiesel combinations. The early start of combustion is favorable for the higher cetane number, in the other hands, LHV favors the low heat release. So, the blends that contain high ratios of biodiesel exhibited a lower heat release rate (MHRR) [34]. Figure 9 shows that the MHRR was (507 J/S: Diesel), (495 J/S: Sp20), (487 J/S : Sp40) , (473 J/S : Sp100), for the double injection scheme and 0.20mm INB.

Moreover, spirulina biodiesel has a higher viscosity than diesel fuel, decreasing the over-mixing of the air: fuel mixture near the pilot fuel injector, which produces inferior combustion and reduces MHRR [29] . Besides, the lower MHRR can be related to the higher oxygen content in the biodiesel molecules. In contrast, increased biodiesel fuel in the blends leads to decreased calorific value compared to baseline diesel fuel. Moreover, Figure 9 shows the split injection scheme influence in the MHRR. As seen from graphs that MHRR for the triple injection scheme less than double injection scheme. The split injection strategy depends on dividing the main fuel injection into two or three identical injection pulses. Besides, the combustion process impacted by the malty injection strategy.



(a) (b)

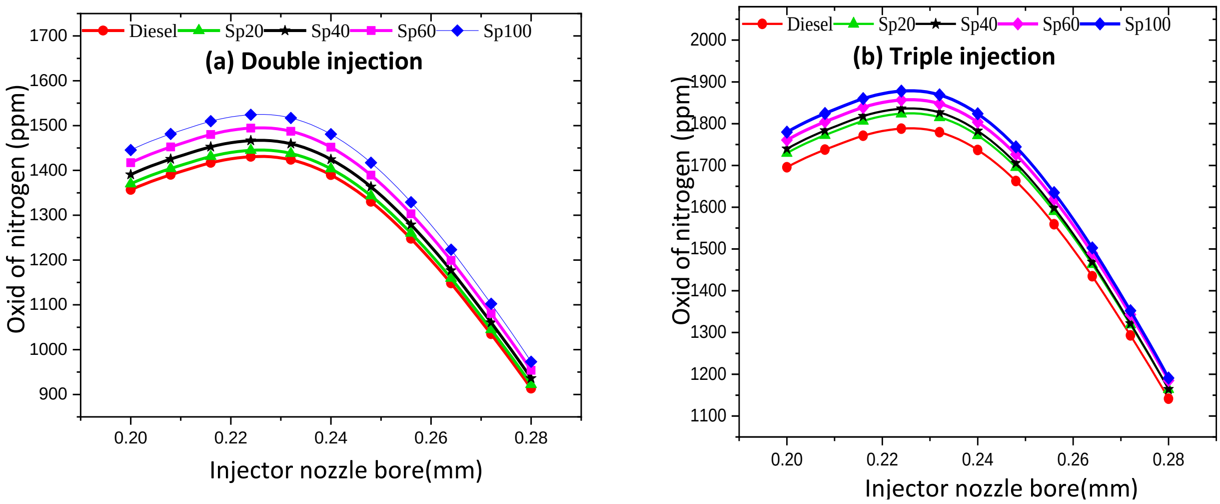
(c) (d)

Fig. 9. Maximum heat release rate versus injector nozzle bore (a) Diesel (b) SP20 (c) SP40 (d) SP100

3.5 NO_x

The effect of biodiesel blends, different split injection strategies (double and triple injection scheme), and injector nozzles bore on the oxide of nitrogen (NO_x) emissions quantity are shows in Figure 10(a) and Figure 10(b). NO_x emissions formation agrees with the zelodvioh mechanism [35], which is a strong justification for numerous studies using biodiesel as an alternative fuel. NO_x produced within exhaust comprised nitric oxide (NO) and nitrogen dioxide (NO_2). The NO_x emissions depend mainly on the oxygen concentration in the fuel, dwelling reaction time, in-cylinder temperature, viscosity, air-fuel ratio, combustion, and cetane number [36]. Figure 10 (a) & (b) show the effect of diesel, spirulina and its blends, split injection scheme, and injector nozzles bore on the NO_x emissions. The results revealed that NO_x for spirulina blends was higher than baseline diesel fuel due to the oxygen content in spirulina blends. This resulted in improved combustion efficiency, causing a complete combustion reaction. This is consistent with previous studies like [37-39]. The NO_x for Sp20, Sp 40, Sp 60, Sp100 was higher than that obtained using diesel as fuel by 1.66%, 1.712%, 6.99% and 14.96%, respectively. This was at 0.2mm (INB) with a double injection scheme as shown in Figure 10(a).

Also, at 0.2 mm (INB) and triple injection scheme, the NO_x emissions for Sp20, Sp40, Sp60 and Sp100% were found to be higher than when diesel was used by 4.01%, 5.49%, 8.16% and 11.69%, respectively as shown in Figure 10(b). This could be explained by a higher oxygen content associated with spirulina biodiesel and its blends used as an alternative fuel for diesel in combustion engines [36]. The spirulina biodiesel fuel was considered an oxygenating fuel with about 12% higher molecular oxygen than diesel. This fuel also has a linear air-fuel mixing rate reliant on the oxygen content in the fuel molecules structure. Moreover, dwelling reaction time is attributed to ignition delay, which corresponds to the blends of biodiesel more than diesel. Increasing the NO_x emissions in exhaust favors the intermolecular oxygen and ignition delay (ID), which are more operative parameters compared with low heating value and evaporation latent heat of fuel [39].



(a) (b)

Fig. 10. Emission of NO_x versus injector nozzle bore (a) Double injection (b) Triple injection

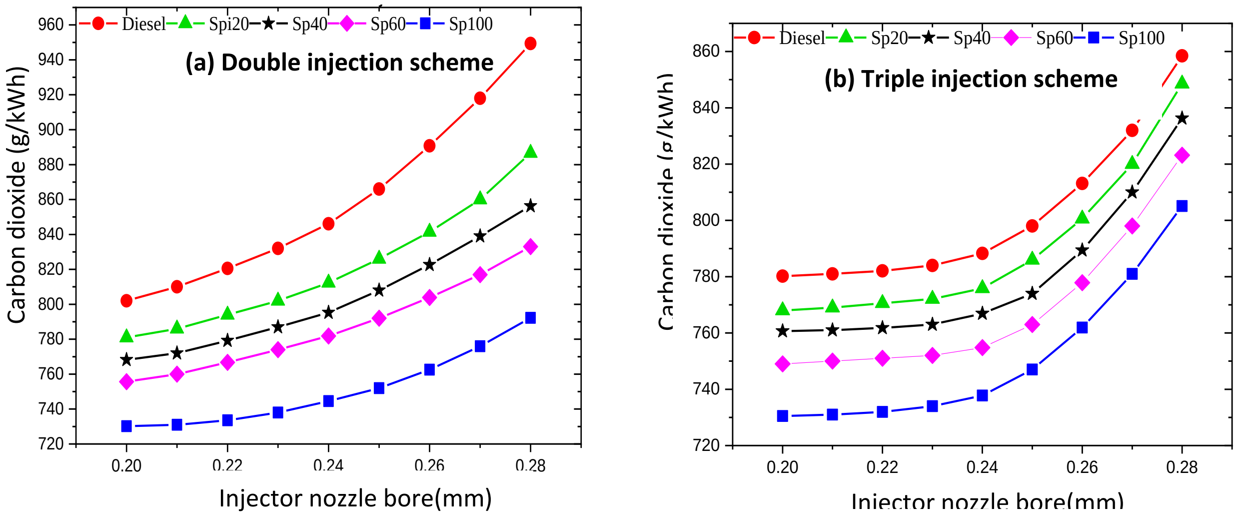
Also, the injector nozzle bore has a distractive effect on the NO_x concentration, Figure 10(a) and Figure 10(b) show that an obvious reduction in NO_x emission corresponds to an increase in the injector nozzle bore. For Sp20% the NO_x emissions were found for 0.22 mm (INB) to be 2.13% higher than for 0.2 mm (INB), while for 0.24 mm (INB) ,0.26 mm (INB) and 0.28 mm (INB) reduced by 5%, 10% ,15.74% and 23.39%, respectively than NO_x of 0.2mm (INB) with double injection scheme as shown in Figure 10(b). When the triple injection scheme was implemented, the NO_x emission was found for 0.24 mm (INB) ,0.26 mm (INB) and 0.28 mm (INB) to be reduced by 8.9% ,15.65% and 38.07%, respectively, than NO_x of 0.2mm (INB) as shown in Figure 10. This was due to the sharp reduction of in-cylinder gas temperature in the combustion chamber, which plays an important role [22]. The split injection scheme plays a vital role in the account of NO_x emissions. An increase in split injection schemes caused a prolonged ignition delay (ID) period and combustion duration. This leads to a higher in-cylinder gas temperature which is imperative for promoting the Zeldovich NO_x formation mechanism. Compared to the double injection scheme, the rate of NO_x in the triple injection scheme is higher [40].

3.6 Carbon Dioxide

From the power and transportation sectors, carbon dioxide (CO_2) is a crucial concern and must be curtailed for its effect on the environment and drive to green technologies. The structure of combustion wave propagation in diesel engines consists of fuel, air, and a quantum of heat. These are required to generate the ignition of the compressed air-fuel mixture [41]. CO_2 is one of the products of the combustion process, which gives a clear indication of the completion of fuel combustion. While NO_x, CO, HC, and smoke are the products of incomplete combustion [42]. Hence, it could be said that there is a trade-off relationship between CO_2 and NO_x emissions. The composition of carbon dioxide in the exhaust depends on the fuel's calorific value, density and oxygen content [29]. One of the effective ways to reduce CO_2 is utilizing biodiesel and its blends as an alternative oxygenated fuel [43]. Figure 11(a) and Figure 11(b) show the effect of diesel, biodiesel, and its blends, spilt scheme injection and an injector nozzle bore on the CO_2 concentration. On average, the result showed that the rate of CO_2 emission for spirulina biodiesel blends was lower when compared with using diesel fuel. These results are similar to previous studies [18]. The CO_2 for Sp20, Sp40, Sp60 and Sp100 was lower than that of diesel by 3.75%, 4.40%, 6.13% and 9.68%, respectively, at 0.20mm (INB) and double injection scheme as shown in Figure 11(a). Also, at 0.2 mm (INB) and triple injection scheme, the CO_2 concentration for Sp20, Sp40, Sp60 and Sp100 was lower than diesel by 4.00% ,4.12%, 5.49% and 8.71%, respectively; Figure 11(b). The biodiesel blends are attributed to lower CO_2 emissions compared to pure diesel. Biodiesel significantly reduces CO_2 emitted due to the high oxygen content of biodiesel and its blend that played a vital role in combustion products. The abundance of oxygen improves the combustion process, reducing carbon dioxide levels in the combustion products. A similar result trend was reported by Datta & Mandal [44].

The injection nozzle bore strictly affects the air mixing rate of fuel. An increase in INB was reported leading to poor atomization of the mixture that affects the quality of combustion and increases the rate of carbon dioxide production [22]. Figure 11(a) Figure 11(b) shows that obvious rise in CO_2 emission corresponding to increase in injector nozzles bore. For Sp100 the CO_2 concentration was found to be higher for 0.22 mm (INB) ,0.24 mm (INB),0.26 mm (INB) and 0.28 mm (INB) by 1.40 %, 3.66 %, 6.36% and 9.76%, respectively. This is against the CO_2 of 0.2 mm (INB) with double injection scheme as shown in Figure 11(a). While, for triple schema injection, the CO_2 emissions were found to be higher by 0.19 %, 0.10%, 2.55 % and 8.51%, for 0.24 mm (INB),0.26 mm (INB) and 0.28 mm (INB) respectively when compared with the CO_2 of 0.2mm (INB) as shown in Figure 11(b).

The effect of the split injection strategy on the CO_2 emission reflected a trend of CO_2 emissions for different injection nozzles bore (INB). This was when various proportions of diesel, biodiesel and its blends were used for triple and double schema injection. Corresponding to the split injection schema are the pulses of air-fuel injected lately and retarded the combustion phase progressively in the compression stroke and moves towards TDC leading to rise in gas temperature. This effect promotes the combustion process and increases gas temperature resulting in increased NO_x and decreased CO_2 as a trade-off [29]. Besides, pulses of air-fuel mixture injected consume a major oxygen quantity required leading to the lack of CO oxidation reaction and reduction in the CO_2 emission in triple.



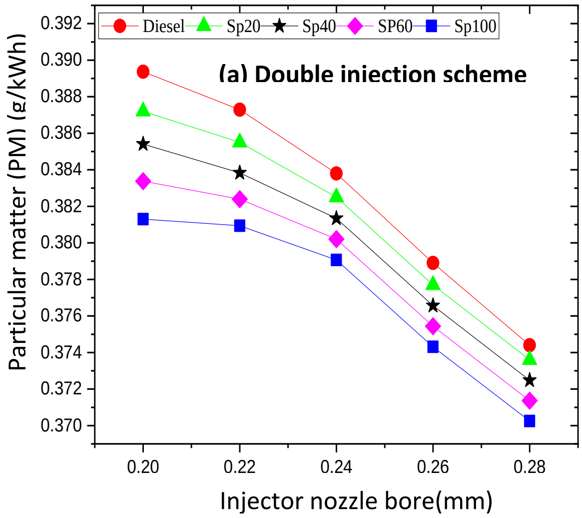
(a) (b)

Fig. 11. Emission of carbon dioxide versus injector nozzle bore (a) Double injection (b) Triple injection scheme

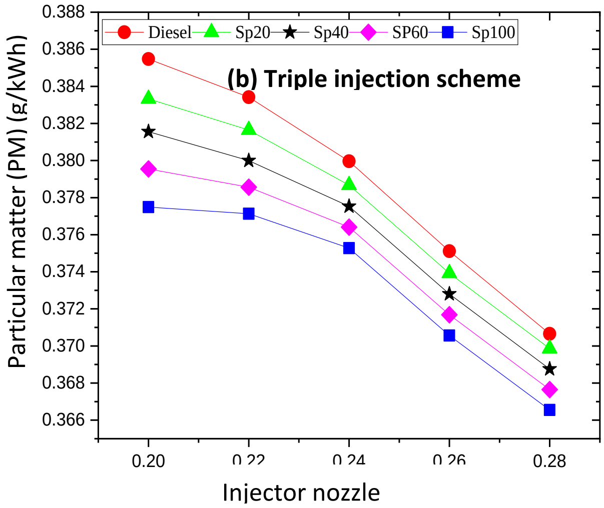
3.7 Particular Matter (PM) Emission

The particular matter (PM) is one of the essential emissions that combustion production of compression ignition (CI) engines. PM emitted from combustion mainly depends on the carbon chain length of the test fuel. Biodiesel blends have a long carbon chain length compared to baseline diesel fuel. Increasing the oxygen concentration within test fuel reduces PM emitted from the combustion [45]. Variations of the PM with different split injection schemes and various sizes of the injector nozzle bore for diesel-biodiesel blends are shown in Figure 12. In this manner, PM was lower in Sp20, Sp40, Sp60, and Sp100 by 2.4%, 3.9%, 4.2%, and 5.7% compared to the diesel for the double scheme injection and (0.20 mm) injector nozzle bore. The same trend has been shown for the triple scheme injection.

The injector nozzle bore effect in the PM emissions shown in Figure 12. The diameter of the injector nozzle significantly influences the fuel atomization development, combustion process and emissions in compression ignition (CI)engines. The fuel injection system is designed to make the air-fuel flow with a high amount of atomization to generate better fuel penetration, to enhance the evaporation during the transaction fuel process in a finite time purpose to improve the combustion process [46]. For very developed CI engine, the fuel injected with high pressure generates cavitation around the injector nozzle's inlet and exit hole. The fuel spray inclination appears wider due to the rising fuel spray formation and boosting the atomization. Improving the fuel sprayed leads to completing all the combustion phases, reducing emissions and lowering the specific fuel consumption [20]. In addition, there is a method of raising the CI engines efficiency by increasing the combustion process duration so that more time is available to complete the oxidation process and reduce the emissions level [41]. In this manner, and as observed from Figure 12. decreased the injector nozzle bore results in a reduced tendency for PM production.



(a)



(b)

Fig. 12. Particular matter (PM) emission versus injector nozzle bore (a) Double injection scheme (b) Triple injection scheme

4. Conclusion

The influence of the splits scheme strategy, various injector nozzle bore and alternative spirulina biodiesel fuel on the combustion and emissions characteristics for diesel engines direct injection. The current numerical study inferences are listed as follows.

I. The promotion of diesel engines by utilizing spirulina biodiesel fuel the maximum gas temperature (MGT), maximum heat rises rate (MHRR), CO_2, and particular matter (PM) while the best agreement with the Sp100.
II. The results showed a decreasing trend of the maximum pressure rate (MPR) with increasing injector nozzle bore (INB). The MPR was (76.53 bar at 0.20 mm INB), (75.90 bar at 0.22 mm INB), (74.86 bar at 0.24 mm INB), (73.32 bar at 0.26 mm INB), and (71.83 bar at 0.28 mm INB) for bassline diesel fuel and double injection scheme.

III. MPG increased by 4.2%, MGT is increased by 8.9%, ID increased by7.9%, MHRR decreased by 9.5%, NO_x decreased by 7.8%, CO_2 decreased by 3.9%, and PM decreased by 6.3% were compared to the double injection scheme, at 0.2 mm (INB).
IV. There is a reduction trend of MPG, MGT, NO_x and PM emissions with an increase in INB. But the increasing trend was evident with MHRR, CO_2 emissions.
V. The ignition delay (ID) with triple injection scheme was higher than with double injection scheme by 1.137 %, 1.292 %, 1.096 %, 0.821 % and 0.352 % for Diesel, Sp20, Sp40, Sp60 and Sp100, respectively, at 0.2 mm (INB).

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