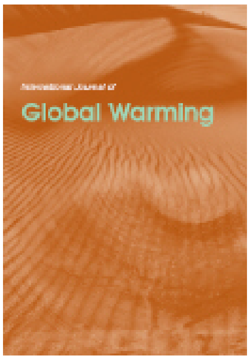

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NO and performance characteristics of a CI engine operated on emulsified fuel

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Abstract: Emulsified fuels are among the alternative solutions nowadays when global warming and emissions are on the hot topic. This research presents the experimental and computational results of an engine running on emulsified fuel containing 10% water. The investigated engine in experiments is a naturally aspirated, single-cylinder diesel engine. In-cylinder pressure, power output, specific fuel consumption and NO emission data have been obtained from experiments and this data have been used to fit some coefficients of the two-zone combustion model. After fitting the coefficients, the effect of design parameters, bore, stroke, inlet pressure and temperature, compression ratio, equivalence ratio, residual gas fraction, cylinder wall temperature, and start of injection time, on engine performance have been investigated. As a result, the largest reductions in NOx emissions were 91.28% and 88.21% at 15% and 15°CA greater crank angles than the original values for residual gas fraction and ignition time, respectively.

Keywords: engine design; emulsified fuel; water-diesel emulsion; NO emissions; combustion model; engine combustion; model-based calibration; compression ratio; equivalence ratio; residual gas fraction; injection time; inlet pressure; inlet temperature; cylinder wall temperature.

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1 Introduction

Today, alternative power systems and emission-reducing solutions to traditional internal combustion engines are discussed due to increasing environmental concerns. Alternative power systems are predicted that it will take a long time for them to become widespread. Therefore, it is important to optimise the design parameters of internal combustion engines accordance with alternative fuels for emission reduction.

Stroke and bore parameters have a fundamental effect on engine performance as they are related to engine dimensions. In addition to these parameters, valve dimensions and combustion chamber geometry, which are related to heat transfer and effect on engine performance, are among the important engine design parameters (Hoag et al., 2016). Another engine design parameter, average piston speed (or engine speed), is an important parameter for high specific power requirements.

But an upper limit is imposed on average piston speed when friction and durability are considered. This imposes an upper limit on the stroke value indirectly (Filipi and Assanis, 2000). Compression ratio parameter, which is ratio of top and bottom dead volumes, can be increased to improve power output and thermal efficiency of the engine (Khayum et al., 2021). However, NO emissions increase with increasing compression ratio (Cung et al., 2021). Because of this reason, performance requirements and engine materials come to the fore in determining the maximum value of compression ratio (Prasad et al., 2021).

The combustion rate, which has a direct effect on engine performance and emissions, basically depends on the waste gas and equivalence ratio. Residual gases vary with charging and scavenging pressures, compression ratio, piston speed and valve timing. As the residual gas increases, engine performance and emissions decrease (Heywood, 2018). An increase in combustion speed can be achieved by increasing the equivalence ratio a certain level (Zheng et al., 2019). Aklouche et al. (2017) observed that increasing the equivalence ratio from 0.35 to 0.7 reduced HC, CO and NOx emissions by 77%, 58% and 24%, respectively.

Jayashankara and Ganesan (2010) developed a computational fluid dynamics model to examine the effects of fuel injection timing on engine performance and emissions. They found that if the injection timing was delayed, the in-cylinder pressure, temperature, heat release rate, cumulative heat release and NOx emissions increased. Moreover, Mohiuddin at al. (2021) stated that delayed injection timing from the reference value led to an increase in specific fuel consumption and the increase in NOx emissions was realised at significant levels (Xu et al., 2019).

In the literature, there are many applications implemented to reduce NOx emissions (Dhahad and Fayad, 2020; Mehregan and Moghiman, 2020; Wang et al., 2020; Ayhan et al., 2020). One of these methods is to water in diesel emulsion fuel (WDEF) (Samec, 2002; Samec et al., 2000; Hoseini and Sobati, 2021; Senthur et al., 2021). The purpose of this method is to slow down the NOx formation mechanism by reducing the flame temperature in diesel engines with the help of the water in the fuel (Hoseini and Sobati, 2021). Studies have reported that the formation of NOx in WDEF emulsion fuel is significantly reduced. Khanjani and Sobati (2021) reduced NOx emissions up to 25.53% thanks to the emulsified fuel created with diesel, biodiesel and water. Also, Senthur et al. (2021) expressed that emulsion fuels containing 5%-10%-15% water reduced NOx emissions 125.23 ppm, 145.61 ppm and 161.43 ppm compared to the reference diesel fuel. Engine performance parameters such as fuel thermal efficiency, brake thermal power and specific fuel consumption are of vital importance as well as the environmentalism of the fuel. Many researchers have concluded that there is a slight decrease in power and torque in the emulsion fuel due to the low heating values (Jhalani et al., 2019). For this reason, it is important to determine the optimal amount of water. Okumuş et al. (2021, 2020) determined the optimum ratio of water used in emulsified fuel to 10%, depending on the performance parameters and NOx emissions.

In this research, the effect of the water diesel emulsion fuel on the performance and emission parameters of a single cylinder diesel engine have been investigated with experimental and computational methods. The investigated in experimental methods engine is a naturally aspirated, single-cylinder diesel engine with a direct injection system. For the computational method, the coefficients of the two-zone combustion model have been used and fitted with the data obtained from the experiment. Thus, the effects of engine design parameters on performances parameters of the engine operating with this fuel have been investigated by using the two-zone combustion model. The design parameters considered are bore, stroke, compression ratio, residual gas fraction, equivalence ratio and start of injection time while engine performance parameters are power output (kW), specific fuel consumption (gr/kWh), thermal efficiency (%) and NO (ppm) emission.

2 Experimental setup

The experiments have been carried out with 3LD510 diesel engine, which is a Antor, four-stroke, naturally aspirated direct injection, water cooled and single cylinder. The engine has a cylinder volume of 510 cm^3, bore length of 85 mm and stroke length of 90 mm. The rated power and speed of engine are 9 kW and 3,000 rpm, respectively.

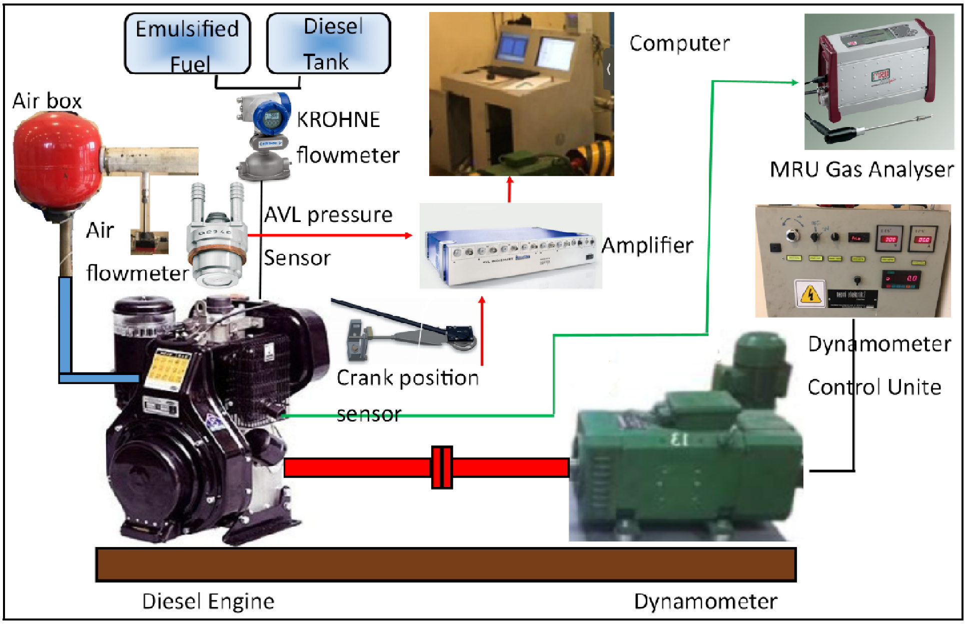
In the tests, the control of engine speed and load has been obtained with a KEMSAN brand direct current dynamometer. In order to measure the engine torque, it has been measured with an ESIT brand, S type load cell with a precision of 0.1 N. The KROHNE Optimass 3300 Coriolis mass flow meter has been used to measure the fuel. The accuracy of the flow meter used is 0.1 kg/h. Since the test engine used is a single cylinder, an air box ten times larger than the cylinder volume has been placed in front of the flow meter used to minimise fluctuations in intake air measurement. The inlet air mass flow rate measurement has been made with a Bass Instruments brand thermal mass flow meter.

In-cylinder pressure data have been collected with the help of the AVL QC34D piezoelectric pressure gauge placed on the cylinder head. The micro voltages obtained from the sensor were powered by the Indismart Gigabit amplifier. AVL 365 C Crank position sensor has been used to measure the crank angle. The information obtained from the crank position sensor and amplifier have been combined and visualised with AVL Indicom software.

The engine has been run for sufficient time at each measurement to achieve a steady state conditions. The experiments have been repeated three times to check the reproducibility of all results. The experimental setup designed and setup to carry out this study is shown in Figure 1.

In the experiments, 88% diesel, 10% water and 2% surfactant emulsion have been used as a fuel which was determined as the optimum ratio in previous studies of Okumuş et al. (2020). Emulsions are mixtures of two or more liquids that do not mix under normal conditions. In stable diesel-water emulsions, water molecules should be dispersed in diesel fuel as homogeneously as possible. If there is more than one layer in the emulsion obtained, it is not suitable for use in diesel engines. There are many variables such as water concentration, surfactant concentration, surfactant type used, mixing speed, mixing method and mixing time that affect the stability of diesel-water emulsions. In order to obtain the emulsified fuel to be used in the experiments, it was first decided to use the surfactant. Two different surfactants, Span 60 and Tween 80, were used in the emulsified fuel used in the experiments. The ratios of surfactants were adjusted so that the HLB value of the surfactants was 10 and the percentage in the emulsion was 2. In order for the emulsion to remain stable, the water particles should be reduced to as small size as possible. For this, a multi-blade mixer was used to prepare the emulsion. While forming the emulsion, a high speed mixer can be preferred to ensure homogenisation in a short time. However, the speed of the mixer is too high, causing the water molecules to be thrown out of the emulsion. For this, 4,000 rpm was chosen as the optimum stirring speed.

Figure 1 Schematic diagram of the experimental setup (see online version for colours)



3 Theoretical model

Computational models have an important place for the internal combustion engine analysis. There are many reasons why these models are widely used by engineers and researchers. While it is necessary to setup different experiments to measure different parameters in the analyses performed in the classical experimental environment, as in this study, multiple parameters can be calculated in the same environment and at once by using the computational model. It provides users with the highest performance in a shorter time period with less energy. This also brings financial savings to the users.

The two-zone combustion model gives more precise results than the ideal cycle models (Gonca, 2017a). In the two-zone combustion model, the combustion chamber is divided into two as burnt gas and unburned gas zone. Moreover, the coefficient of in the two-zone combustion model has been fitted according to experimental results for emulsified fuels in this research. The energy conservation equation could be expressed in the following equation:

$$m\frac{du}{d\theta}+u\frac{dm}{d\theta}=-\frac{dQ_b}{d\theta}-\frac{dQ_u}{d\theta}-P\frac{dV}{d\theta}-\frac{dm_1}{d\theta}h_1 \quad (1)$$

The heat transfer from the burnt zone to the unburned zone is expressed by the following pair of equations:

$$Q_b = h_{tr} A_b T_{bw} \quad (2)$$

$$Q_u = h_{tr} A_u T_{uw} \quad (3)$$

Stroke volume varies during the cycle depending on the crank angle. The variation of the stroke volume depending on the crank angle is expressed as follows:

$$\frac{dV}{d\theta}=\frac{\pi}{8}B^2S\sin\theta\left[1+\varepsilon\frac{\cos^{-1}\theta}{(1-\varepsilon^2\sin^2\theta)^{\frac{1}{2}}}\right] \tag{4}$$

The internal energy equation to be used to solve the differential equations is as follows:

$$\frac{du}{d\theta}=C_p-\frac{Pv}{T}\left(\frac{\partial\ln v}{\partial\ln T}\right)_P\frac{dT}{d\theta}-v\left[\frac{\partial\ln v}{\partial\ln T}+\frac{\partial\ln v}{\partial\ln P}\right]_T\frac{dP}{d\theta} \tag{5}$$

The mathematical expression of the burnt mass leaking from the segments is as follows:

$$\frac{dm_1}{d\theta}=\frac{Cm}{\omega} \tag{6}$$

Air and injected fuel masses are used for the mass balance in the cylinder and the equation is expressed in differential form as follows:

$$\frac{dm}{d\theta}=\frac{dm_a}{d\theta}+\frac{dm_{fi}}{d\theta} \tag{7}$$

Variation of pressure with crank angle, burnt and unburned gas temperatures, work, heat leaks and losses are expressed as follows:

$$\frac{dP}{d\theta}=\frac{\begin{aligned}&\left\{\frac{1}{m}\left(\frac{dV}{d\theta}+\frac{VC}{\omega}\right)\right.\\&+h_{tr}\frac{A_{cyl}}{\omega m}\left(\frac{v_b}{C_{p.b}}\frac{\partial\ln v_b}{\partial\ln T_b}\sqrt{x}\frac{T_b-T_w}{T_b}+\frac{v_u}{C_{p.u}}\frac{\partial\ln v_u}{\partial\ln T_u}(1-\sqrt{x})\frac{T_u-T_w}{T_u}\right)\\&\left.-(v_b-v_u)\frac{dx}{d\theta}-v_b\frac{\partial\ln v_b}{\partial\ln T_b}\frac{(h_b-h_u)}{C_{p.b}T_b}\left[\frac{dx}{d\theta}-\frac{(x-x^2)C}{\omega}\right]\right\}\end{aligned}}{x\left[\frac{v_b^2}{C_{p.b}T_b}\left(\frac{\partial\ln v_b}{\partial\ln T_b}\right)^2+\frac{v_b}{P}\frac{\partial\ln v_b}{\partial\ln T_b}\right]+(1-x)\left[\frac{v_u^2}{C_{p.u}T_u}\left(\frac{\partial\ln v_u}{\partial\ln T_u}\right)^2+\frac{v_u}{P}\frac{\partial\ln v_u}{\partial\ln T_u}\right]} \tag{8}$$

Hohenberg expressed the heat transfer coefficient as follows (Dabbaghi et al., 2021):

$$h_{tr}=C_1V^{-0.06}P^{0.8}\left(xT_b+(1-x)T_u\right)^{-0.4}\left(\overline{S}_P+C_2\right)^{0.8} \tag{9}$$

The dual Wiebe function was used to determine the heat released from combustion. The expression of the dual Wiebe function depending on the crank angle is expressed below (Yasar et al., 2008):

$$x=a_v\left[Q_{pre}\left(1-e^{-a_v\left(\frac{\theta}{\theta_{pre}}\right)^{m_{pre}+1}}\right)\right]+Q_{dif}\left(1-e^{-a_v\left(\frac{\theta}{\theta_{dif}}\right)^{m_{dif}+1}}\right) \tag{10}$$

In order not to deviate from the focus of this study, the general form of the equation systems is given. It is possible to find more details of the model in Dabbaghi et al. (2021) and Yasar et al. (2008).

4 Results and discussion

The experimental and computational cylinder pressure diagram of the engine running with 10% emulsified fuel is shown in Figure 2. It can be seen that the two curves are close to each other and the computational model has been validated (Okumuş et al., 2020). Therefore, the effects of engine design parameters on engine performance parameters can be examined by via the computational model. In Table 1, the lower and upper limits of the engine design parameters have been shown relative to their original values.

Figure 2 Comparison of experimental and theoretical results of cylinder pressures (see online version for colours)

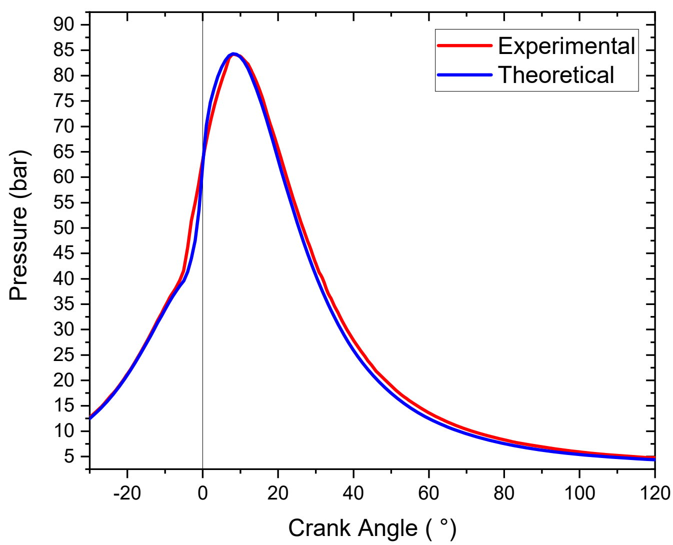
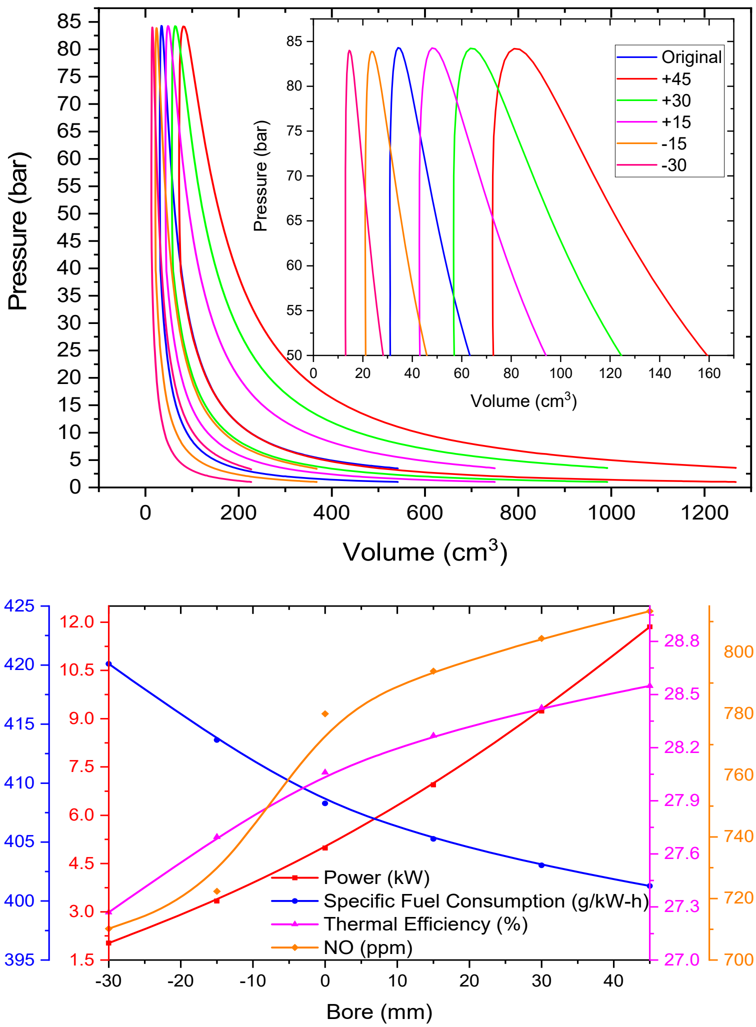


Table 1 Limits of parameters changed in the model

Parameter	*Lower bound*	*Upper bound*	*Unit*
Bore	–30	+45	mm
Stroke	–45	+45	mm
T_i	–100	+50	K
P_i	–20	+20	kPa
T_{wall}	–100	+200	K
Φ	–0.4	+0.4	(-)
θ_{si}	–25	+15	°CA
CR	–7.5	+7,5	(-)
RGF	–5	15	%

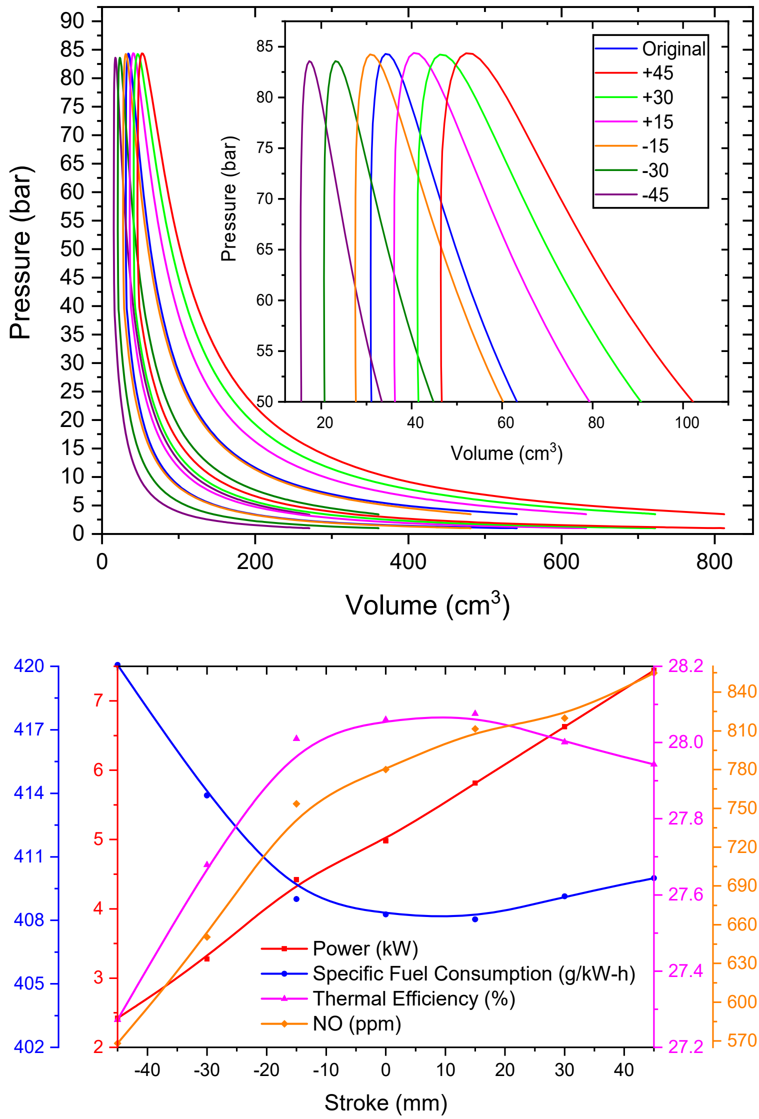
Figure 3 illustrate the impact of cylinder bore modification on engine performance and the formation of NO emission in a diesel engine powered by emulsified diesel fuel. Engine power increased significantly with the increase in cylinder bore, while the increase in BTE and NO emissions was limited. Also, a slightly decrease was observed in the SFC. Increasing the cylinder diameter by 45 mm from 85 mm, which is the actual value, the enhance in engine power, BTE and NO emissions occurred by 138%, 1.75% and 4.27%, while the SFC decreased by 1.71%.

Figure 3 The effects of cylinder bore (see online version for colours)



The reason for the change in engine power is that the bore parameter is directly related to the main dimensions of the engine. The change in the bore of the cylinder increased the amount of fuel entering the combustion chamber since the equivalence ratio was kept constant in the model. The fact that the increase in BTE remained limited with respect to the engine power slightly decreased the BSFC. NO emissions varied according to the amount of burnt fuel and in-cylinder temperatures.

Figure 4 The effects of stroke (see online version for colours)



In Figure 4, the impact of changing the engine stroke length from 45 mm positive to 45 mm negative were examined on engine performance and NO formation. The stroke parameter appears to have strong effects on engine power. If the stroke length was reduced by 45 mm, the power decreased by 51.40%, while if the stroke length was increased by 45 mm, the power increased by 49.37%. Since the change in stroke also changes the amount of fuel in each cycle depending on the constant equivalence rate, the SFC is virtually unchanged. The maximum change observed in the SFC was observed at

–45 mm, which is 2.88% higher than the standard engine. The shortening of the stroke length has been helpful for reducing the NO formation. A decrease in NO emissions of up to 27.15% was observed with the reduction of stroke length, while an increase in the stroke length increased NO emissions and a maximum increase of 9.58% was found. The main source of NO emissions is nitrogen and oxygen from the air. Depending on the stroke, change in the amount of air entering affected the NO emissions. In addition, since stroke changes the amount of fuel entering, changes in NO emissions have been determined depending on the temperatures. In addition, the change in the amount of fuel entering the combustion chamber has also affected the NO emissions in the direction of the change in cylinder temperatures (Gonca, 2017b).

Figure 5 The effects of inlet air temperature (see online version for colours)

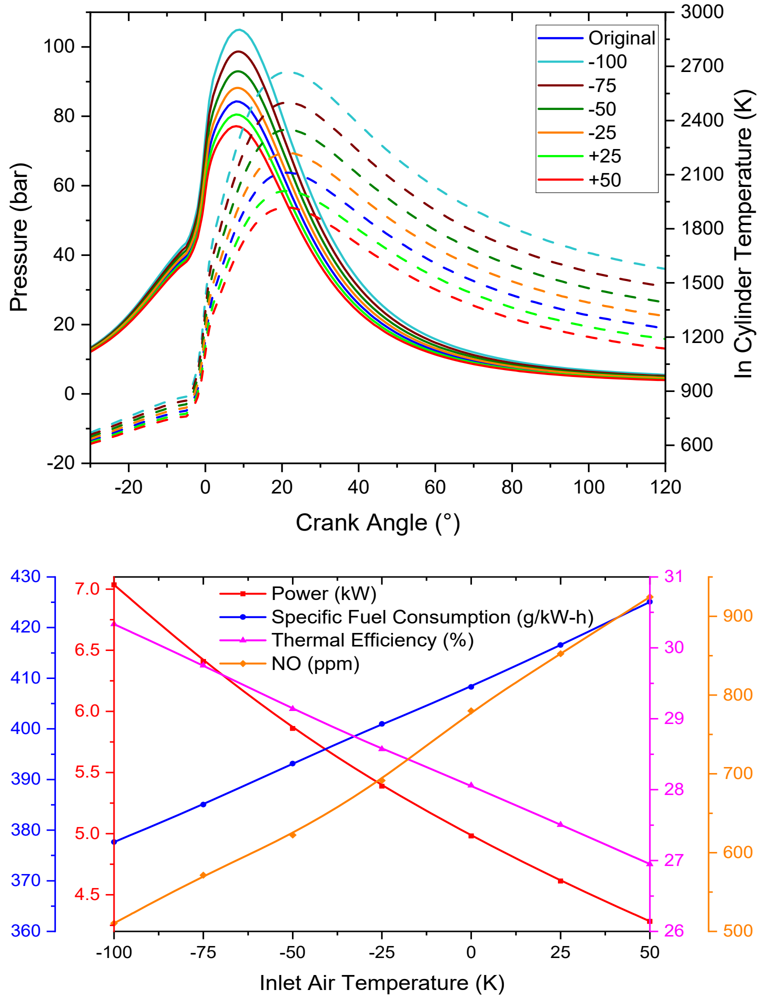


Figure 5 illustrate the impact of inlet air temperature on engine performance and NO emission formation. The effects of cooling the inlet air temperature up to 100 K from standard values or heating it up to 50 K are discussed. Reducing the temperature of the intake air increased maximum in-cylinder pressures and exhaust temperatures. Increasing the inlet temperature by 50 K caused a decrease in power by 14.03%, while BTE decreased by 3.95%. On the other hand, BSFC and NO emissions decreased by 4.11% and 18.52%, respectively. In the case of cooling the inlet air by 50 K, the power and BTE increased by 17.64% and 3.84%, while BSFC and NO emissions decreased by 3.70% and 20.19%, respectively. Increasing inlet air temperature negatively affects volumetric efficiency. The decrease in volumetric efficiency causes these results to occur (Abassi et al., 2010).

Figure 6 The effects of inlet air pressure (see online version for colours)

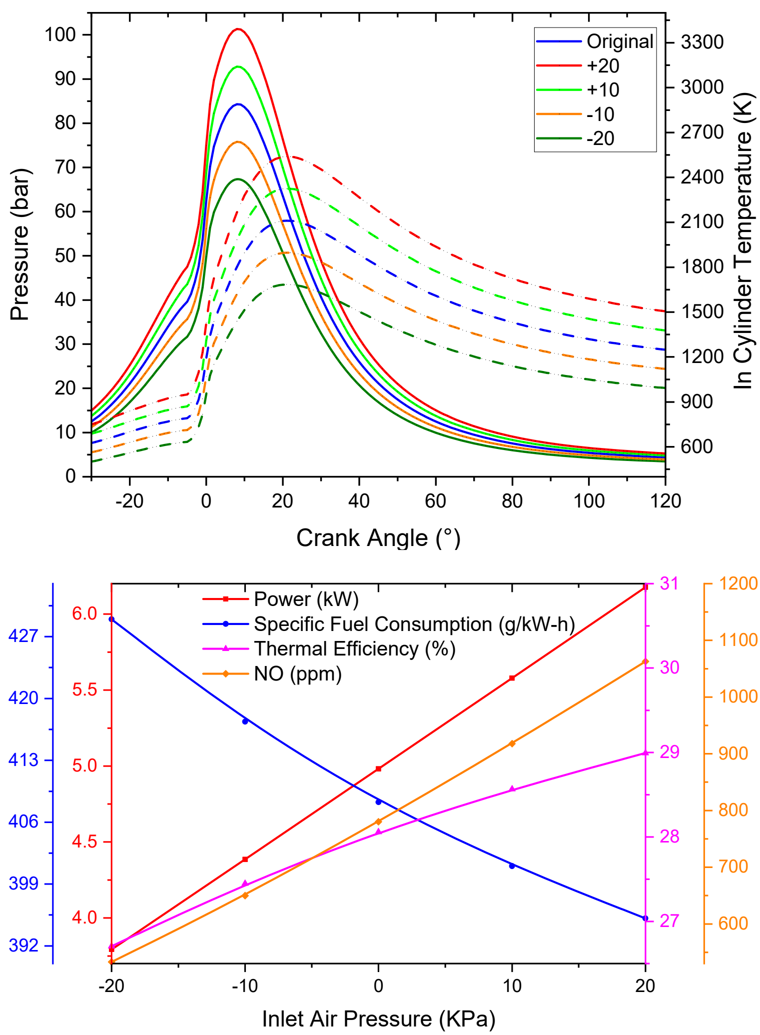


Figure 6 illustrate the impact of inlet air pressure on engine performance and NO emission formation. Unlike the temperature, the increase in the pressure of the inlet air increases the volumetric efficiency. With the increasing inlet air pressure, the maximum pressures in the cylinder and the indicated power values have increased. The positive change in the inlet pressure of 20 kPa decreased the BSFC output of the engine by 3.23%. However, engine power, BTE and NO formation decreased by 23.99%, 3.33% and 36.24%, respectively (Han et al., 2018).

Figure 7 The effects of cylinder wall temperature (see online version for colours)

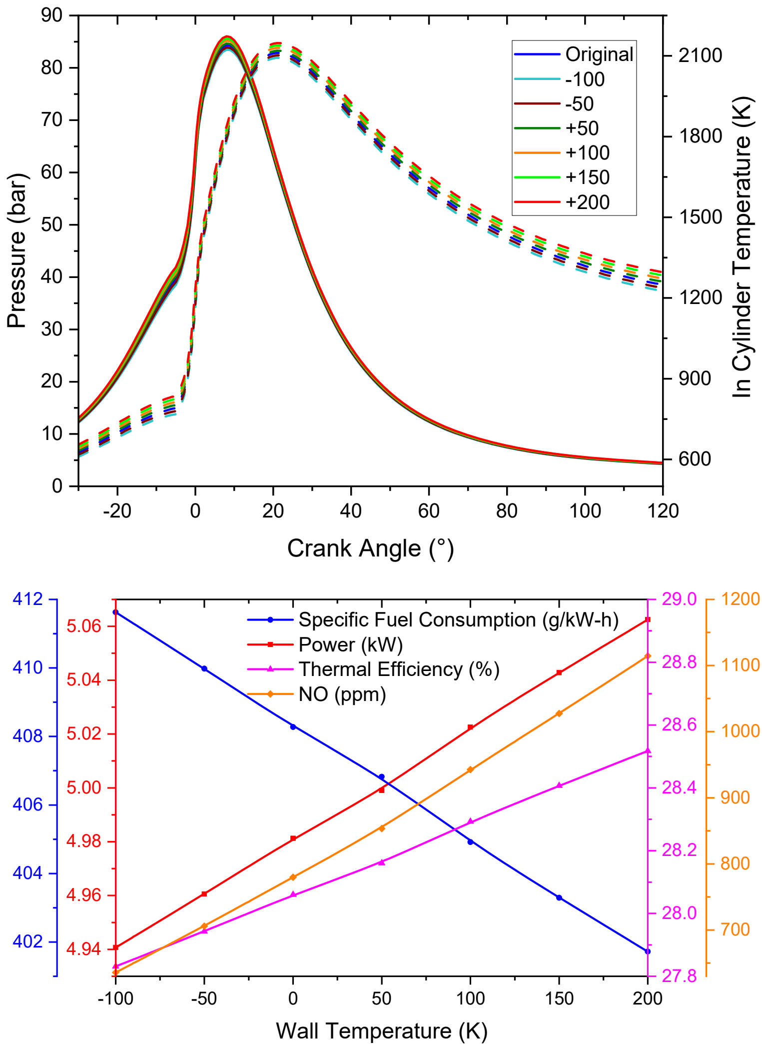


Figure 7 illustrate the impact of cylinder wall temperature on power, SFC, BTE and NO emission. The effect of cylinder wall temperature on power, SFC and BTE is very small. The main effect of cylinder wall temperature was on NO emission. Increasing the cylinder wall temperature by 50 K affects the NO emission by 9.4%, and when the increase reaches 200 K, this ratio becomes 42.87%. Augmenting cylinder wall temperature increases in cylinder gas temperature (Wang and Fan, 2021). In addition, this situation negatively affects the heat transfer from the working fluid to the cylinder wall (Bolla et al., 2020). All these components cause higher NO formation.

Figure 8 illustrate the impact of the equivalence ratio on the engine performance and NO emission formation of the diesel engine running with water-diesel emulsified fuels. At the point where the equivalence ratio is equal to 1, the maximum power, maximum BTE and minimum SFC value are obtained from the engine (Kayadelen, 2017). At this point, the presence of maximum pressures positively affects the engine performance. However, NO emissions are also maximum at this point due to increased in-cylinder temperatures (Karyeyen, 2018). If the equivalence ratio deviates from one point in the positive or negative direction, it reduces NO emissions as well as engine performance.

Figure 8 The effects of equivalence ratio (see online version for colours)

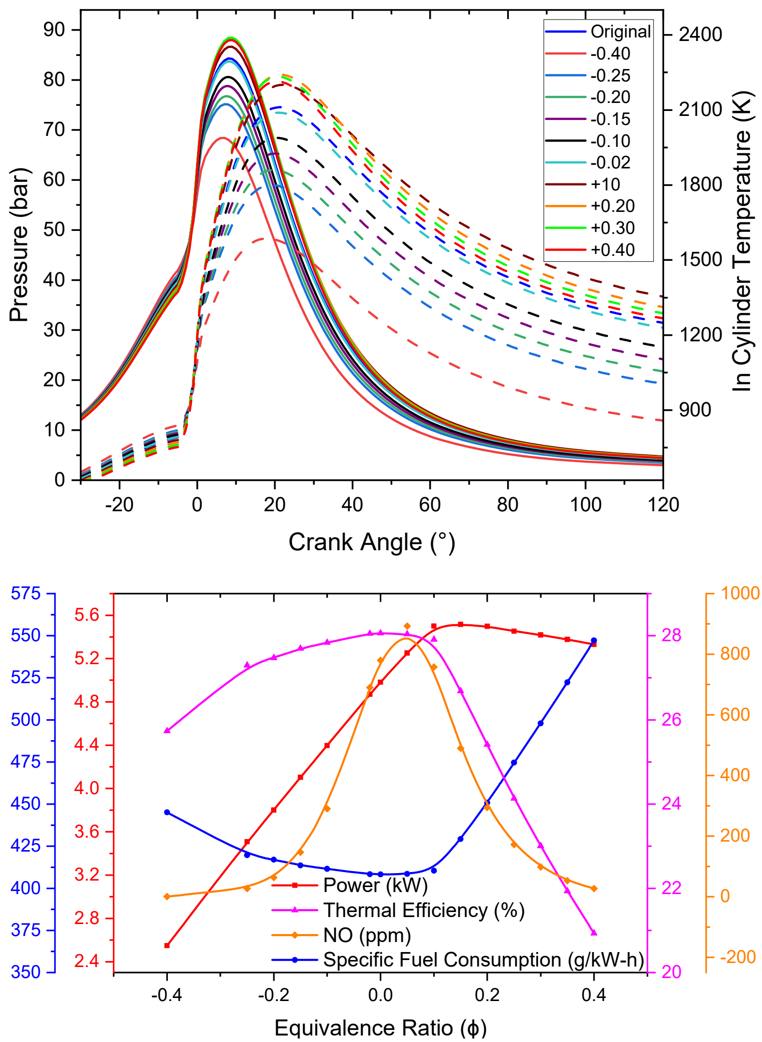


Figure 9 illustrate the impact of the start of injection time on the performance and NO formation of the diesel engine operating water-diesel emulsified fuels. As seen in Figure 9, there is an optimum point for the injection time to provide performance and NO emissions. Maximum power, maximum BTE and minimum BSFC values are obtained from the current start of injection time of the engine. However, if the start of injection time is delayed by 5°CA, it is possible that NO emissions could decrease by 41.36%. Delaying the spraying timing by 5°CA causes losses in power, BSFC and BTE by 0.96%, 0.97% and 0.96%, respectively. However, these losses are acceptable given the considerable reduction in NO emissions.

Figure 9 The effects of start of injection time (see online version for colours)

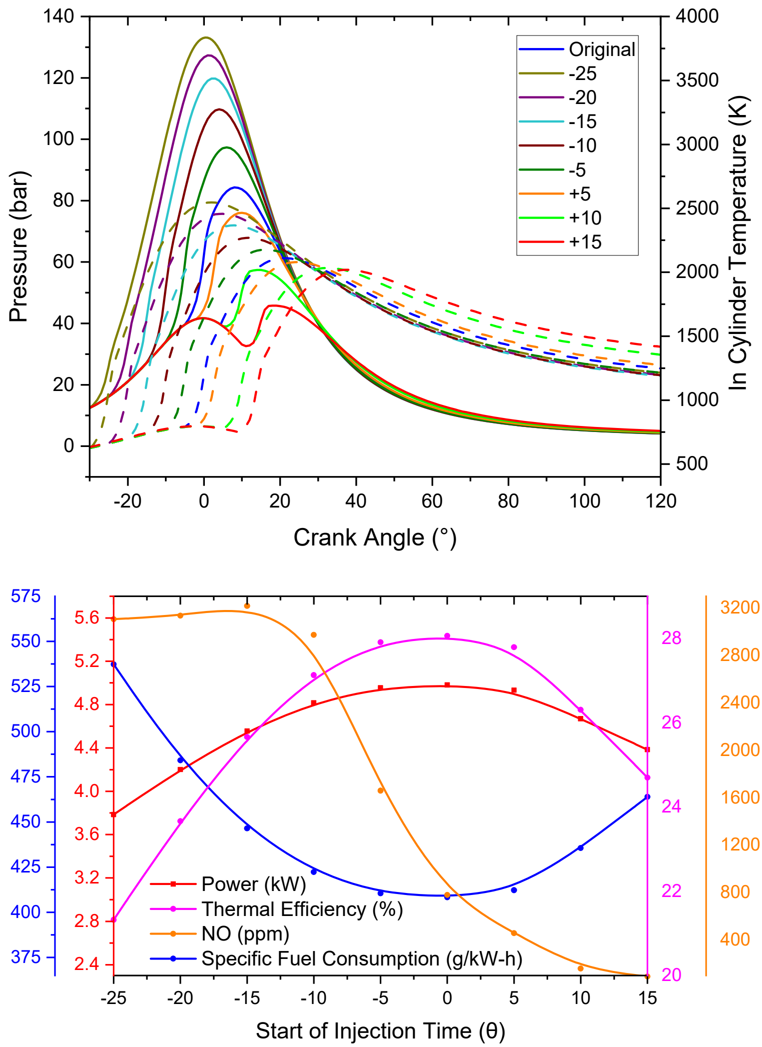


Figure 10 illustrate the impact of the compression ratio on the engine performance and NO emission formation of the diesel engine running with water-diesel emulsified fuels. The effects of the compression ratio have been investigated by changing from the standard value up to 7.5 in the negative and positive direction. By increasing the compression ratio, up to 3.63% gain in engine power can be achieved. However, this situation increased the NO emissions up to 4.15%. NO emissions basically depend on three main parameters: the in-cylinder peak temperatures, the duration of the high temperature the cylinder gases are exposed to, and the oxygen concentration (Anufriev, 2021). Although the peak temperatures decreased, the exposure time of the gases to high temperature increased after TDC. This situation slightly increased the NO emissions, gases (Kökkülünk et al., 2014).

Figure 10 The effects of compression ratio (see online version for colours)

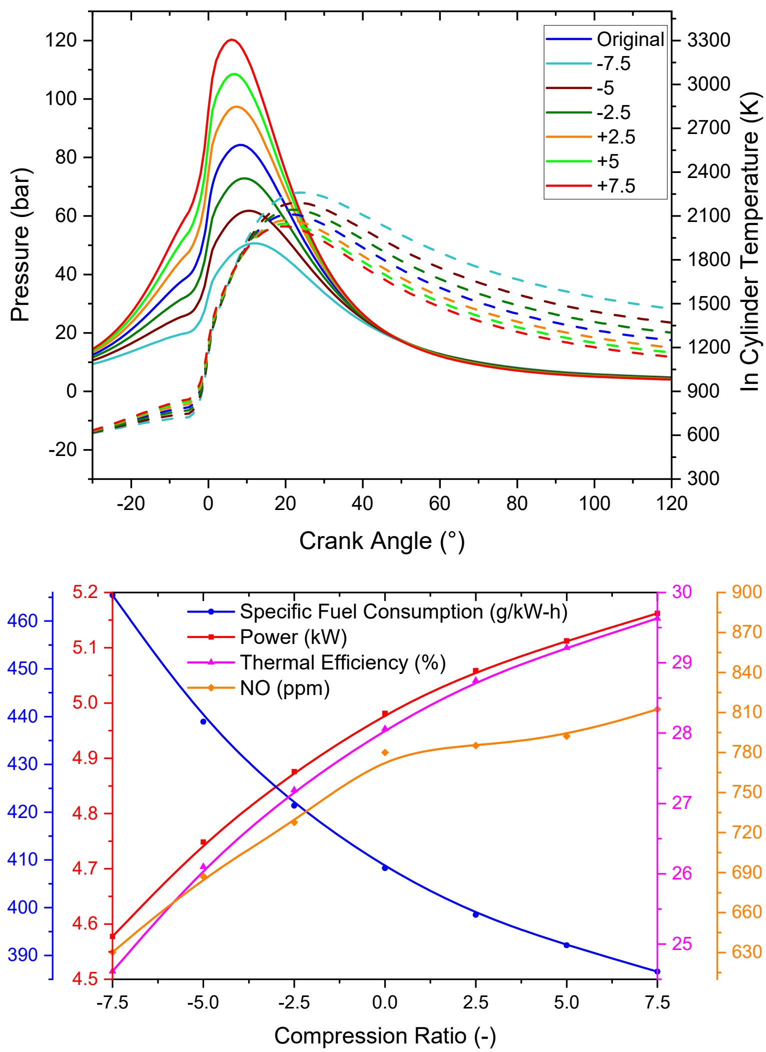
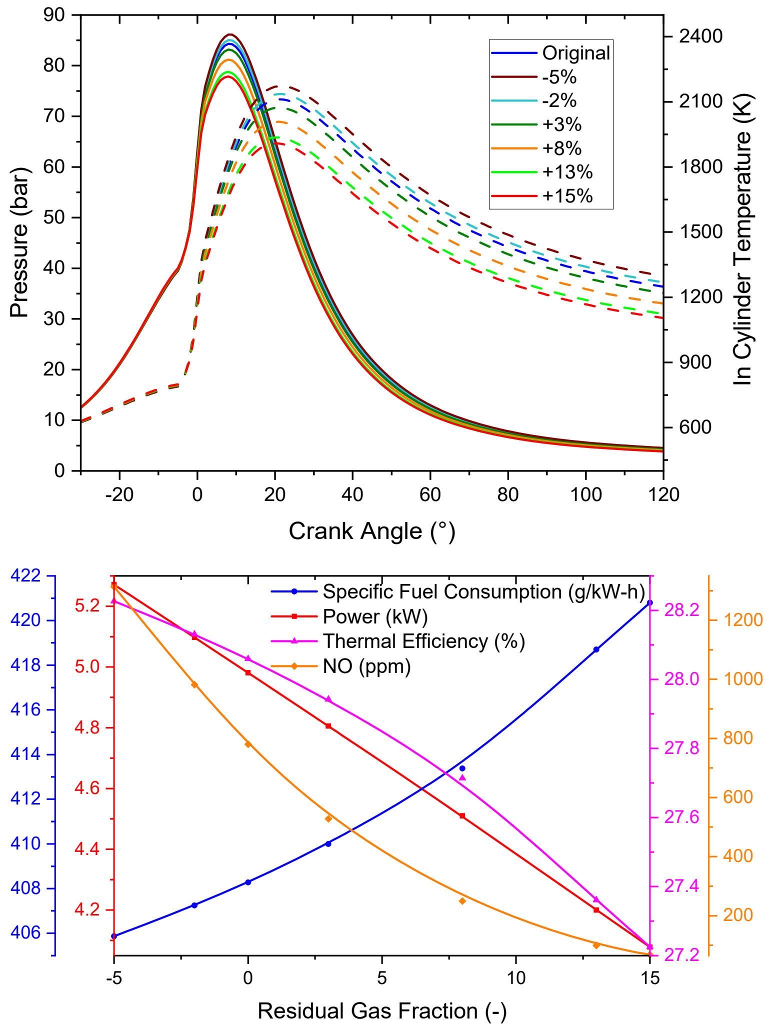


Figure 11 illustrate the impact of the compression ratio on the engine performance and NO emission formation of the diesel engine running with water-diesel emulsified fuels. In order to examine the effects of the residual gas fraction, 5% in the negative direction and 15% in the positive direction were removed from the current value. If the residual gas fraction is increased by 15%, there is a loss of 18.13% in engine power. On the other hand, NO emissions decreased by 91.31%. Increasing the residual gas fraction tends to lower in-cylinder pressures and temperatures (Khoa et al., 2020). For this reason, with the increase in the waste gas fraction, engine performance parameters deteriorated, while NO emissions improved.

Figure 11 The effects of residual gas fraction (see online version for colours)



5 Conclusions

In this research, the performance parameters of an emulsified diesel engine have been investigated with engine design and operating parameters using experimental and computational methodology. In the experimental study, the data of in-cylinder pressure SFC, BTE and NO emission have been obtained from the engine using emulsified fuel by operating at a constant 1,500 rpm. A two-zone combustion model has been developed by fitting the coefficients with the experimental data for the engine running emulsified fuel. In this way, the effects of engine design and operating parameters, which are cylinder diameter, stroke length, intake air pressure and temperature, equivalence ratio, injection start, compression ratio, residual gas fraction, on engine performance parameters have been obtained parametrically. The results obtained are shown below:

- Bore and stroke are two highly effective parameters on power. Power output is significantly affected by the change in these parameters. The change in SFC, BTE and NO has been relatively limited. For each 5 mm increase in cylinder bore, engine power, BTE and NO emissions increased by an average of 43%, 0.92% and 2.78%, while SFC decreased by 0.92%. Similarly, each 5 mm increase in stroke resulted in an average 21%, 0.41% and 7.20% increase in engine power, BTE and NO emissions, while SFC decreased by 0.41%.
- The decrease in the inlet air temperature positively affected the performance data and caused an increase in NO emissions. The air inlet temperature was examined in 25 K increments. Each step in here, power and efficiency decreased by 7.94% and 1.95%, while NO emission increased by 10.42% on average.
- The increase in inlet air pressure showed an inverse relationship with temperature due to volumetric efficiency. The reduction in pressure worsened performance outputs and reduced NO formation. With each 10 kPa increase in inlet pressure, an average increase of 12.98%, 2.08% and 18.86% was observed in power, efficiency and NO, respectively.
- The striking effect of cylinder wall temperature was on NO emissions. NO emissions have increased due to the increase in cylinder wall temperature. On the other hand, serious changes were not observed in the in-cylinder pressure and performance values. With each 50 K increase in inlet pressure, an average increase of 0.41%, 0.41% and 9.81% was observed in power, efficiency and NO, respectively.
- If the equivalence ratio is 1, maximum performance and maximum NO formation have been achieved. At values below and above 1, NO emissions decrease while performance values deteriorate. What was striking here was that the 0.02 decrease in the equivalence ratio reduced NO emissions by 11.52%, while the power was lost by only 2.23%.
- Changing the compression ratio stands out as one of the most effective methods to enhance engine performance and control NO emissions. While the engine performance improves by increasing the compression ratio, an increase is observed in NO emissions depending on the flame temperature. For every 2.5 increase in compression ratio, engine power and NO emissions increased by an average of 4.56%, 4.36% and 2.78%, while SFC decreased by 3.03%.
- The increase in the waste gas fraction is a situation that negatively affects the combustion efficiency. Accordingly, the increase in the waste gas fraction causes the performance to deteriorate and contributes to the reduction of NO emissions. If the residual gas fraction is increased by 15%, there is a loss of 18.13% in engine power. On the other hand, NO emissions decreased by 91.31%.

The effects of the design and operation parameters of the engine operating with 10% water-diesel emulsified fuel were investigated. This study was important in terms of visual presentation in order to determine the optimum value of each parameter, separately. In the future, a multi-objective optimisation study in which all parameters are evaluated under a single framework by determining the objectives, the limits of the parameters and the alternatives may be interesting.

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Nomenclature

SFC	Specific fuel consumption	ppm	Particle per million
BTE	Brake thermal efficiency	Q	Rate of heat transfer
B	Bore	RGF	Residual gas fraction
CFD	Computational fluid dynamics	S	Stroke
CO	Carbon monokside	SOI	Start of injection
CR	Compression ratio	TDC	Top dead centre
ER	Equivalance ratio	T_i	Intake temperature
HC	Hydrocarbon	T_{wall}	Cylinder wall temperature
HLB	Hydrophilic-lipophilic balance	V	Volume
NO	Nitrogen monoxide	WDEF	Water in diesel emulsion fuel
NOx	Nitrogen oxides	Φ	Equivalence ratio
P_i	Intake pressure	Θ_{si}	Start of injection