

Research Article

Simulation of the Effect of Ethanol Blending Ratio on Combustion and Performance of Diesel-Ethanol Dual-Fuel Engine at Different Loads

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Abstract

The use of ethanol in diesel engines as an alternative to fossil fuels and air pollution prevention due to vehicle exhaust emissions has been investigated. It can be seen that for a more practical application of the performance study of diesel-ethanol dual fuel engine, the effect of ethanol on the performance at 100% load as well as at different load conditions should be considered. The effect of ethanol content on combustion and performance characteristics under different load conditions of diesel-ethanol dual-fuel engine was investigated using engine simulation tool. The engine model was constructed using the pre-combustion-diffusion combustion two phase combustion model of GT-POWER, and the simulation results of the main performance indices were compared with the test data. The effect of ethanol blending ratio on the performance of diesel-ethanol blend engine with varying ethanol content (0-20% ethanol) under different loading conditions (25-100% load) was investigated. As the load decreases, the effect of ethanol blending ratio on the brake power becomes weak. With the load decreasing to less than medium (50%), the specific fuel consumption increases rapidly. Also, the indicated thermal efficiency (ITE) and NO_x emission per unit power, usually has an optimal value at medium load. It was shown that the ethanol content under different loading conditions should be set up reasonably. From the practicality and reliability of the simulation results, it is expected that the future will be used for the performance prediction and the optimal design of the diesel-ethanol dual-fuel engine at different loads using the two phase combustion model of GT-Power.

Keywords

Ethanol Content, Different Load, Gt-Power, Two Phase Combustion, Compression Ignition Engine

1. Introduction

In recent years, several aspects have been investigated to use alternative fuels such as bio-alcohol in compression ignition engines, from fossil fuel depletion and environmental protection [1-5]. Bio-alcohols such as ethanol and methanol are considered as relatively effective alternative fuels and potential solutions to reduce exhaust emissions. Many stud-

ies have revealed advantages and limitations in performance and emission of exhaust gases for diesel-ethanol blends [6-14]. Performance tests of a compression-ignition engine have revealed the possibility of reducing the CO, NO_x and soot of exhaust gases under normally specified engine operating conditions, compared with diesel fuel alone. It was

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found that the latent heat of evaporation of ethanol, calorific value, cetane number (CN), etc. caused undesirable changes in combustion and performance, which may have several limitations.

Many studies have analyzed the relationship between auto-ignition and combustion, performance in the diesel-ethanol dual fuel engine. The studies [15-20] has shown that the heat of vaporization in a compressor engine is an important characteristic of fuel, which affects the initial ignition process of combustible mixtures, and affects ignition delay, engine performance and emission characteristics. The increase in alcohol content increases the evaporation heat of the blend and affects the delay of the combustion process. In diesel engines, it was found that this resulted in ignition delay and "stiffening" engine control by affecting the pressure rise rate. Oliveira [21] studied the effect of diesel-biodiesel-ethanol blend (diesel with 7% biodiesel, 5, 10, and 15% ethanol) on the performance, emission and combustion characteristics of diesel power generators. The results showed that the maximum pressure and heat release rate inside the cylinder decreased under low load conditions and increased at high loads with the use of ethanol. Also, with increasing ethanol content, the ignition delay increases, the combustion duration decreases, and the exhaust gas temperature decreases. Li D [22] analyzed that high oxygen content of ethanol increases the air-to-fuel ratio of the blend fuel and changes the combustion efficiency. The low cetane number of ethanol-diesel blends increases the ignition delay and heat release rate, which is related to the ethanol content in the blends. The low cetane number of ethanol is disadvantageous because it decreases the cetane number of the blend and increases the ignition delay. Studies on the possibility of adding ethanol to diesel fuel have evaluated the auto-ignition characteristics and clearly revealed that with increasing ethanol, the auto-ignition characteristics of the fuel deteriorated. Hubert Kuszewski et al. [23] considered ignition delay and combustion delay mainly to evaluate the spontaneous ignition characteristics and determined the effect of gas medium temperature on ignition delay and combustion delay period. In [24-27], auto-ignition was considered as a particularly important parameter among the parameters related to combustion and performance of a dual fuel engine, and a study was carried out to improve it.

In addition, the premixed combustion, performance and exhaust emission relationships of ethanol in a compression ignition engine were analyzed. As indicated in several studies [28], the shorter ignition delay period in diesel engine decreases the amount of fuel accumulated in the combustion chamber and thus the reaction intensity during premixed combustion decreases. Kidoguchi Y [29] reported that the increase of CN of fuel through engine tests had the effect of decreasing the heat release rate during premixed combustion with the maximum cylinder pressure. Moreover, with the increase of CN, the concentration of NO_x in the exhaust gas decreases obviously, especially in the higher load operating

range of the engine. On the other hand, it was observed that the THC (total hydrocarbon) concentration in the exhaust gas increased obviously in the case of lower engine load and lowest CN value. Taghizadeh-Alisaraei A study [30] found that the addition of ethanol in diesel fuel caused the engine performance indexes and vibration, especially irregularities of engine operation, due to the increase of ignition delay period. On the other hand, from the Rakopoulos DC study [31], it is found that an increase in ethanol content in diesel fuel leads to an increase in the maximum combustion temperature, resulting in an increase in NO_x emissions during premixed combustion. Le Ning et al. [32] found that more fuel was accumulated before the start of combustion, thus the premixed combustion process was more intensive, resulting in higher heat release rates. It was also analyzed that with increasing peak pressure values, a maximum temperature increase occurs, resulting in an increase in NO_x emissions. Murcak A [33] and Jayashankara B [34] analyzed that injection timing is an important engine operating parameter that affects combustion and performance of diesel engines and that injection timing improves premixed combustion duration and reduces fuel consumption. Hubert Kuszewski et al. [35] investigated the effect of the addition of 2-ethylhexyl nitrate (2-EHN) on the auto-ignition performance of ethanol-diesel blends (EDB). Liu et al. [36] showed that with increasing 2-EHN content, the ignition delay period is shortened, resulting in a decrease in the heat release rate in the premixed combustion phase. J.M. Herreros et al. [37] found that the addition of additive DGE (diethylene glycol diethyl ether) to ethanol-diesel blends significantly improved the blend stability and the blend auto-ignition characteristics, achieving the simultaneous reduction of soot and NO_x. It was also found that the addition of ethanol in the case of additive could improve NO_x emission, with a decrease in premixed combustion fraction with a decrease in soot emission due to an increase in oxygen content. TUTAK, W. and Jamrozik, A.[38] have analyzed the heat release rate, combustion stages and combustion stability.

The effect of ethanol content on performance and exhaust emissions under low load conditions was also investigated through engine simulations and experiments. Zhiqiang Wang et al. [39] combined the program CHEMKIN and CONVERGE to analyze the effect of ethanol blending ratio on engine performance and emission characteristics under three load conditions of 100, 50 and 25% of diesel-ethanol dual-fuel engine. Praptijantoa A [40] documented the effect of ethanol content on the performance and exhaust emissions of a fuel blend of diesel and alcohol using the engine simulation analysis tool AVL Boost. In the simulation, a one-zone combustion model of AVL Boost was used, the mixing mode between ethanol and diesel fuel was E0, E2.5, E5, E7.5, and E10, and the engine performance was simulated under the load conditions of 0, 10, 20, 30, 40, 50, and 60 Nm. In [41], 5 vol% (E5), 10 vol% (E10) and 15 vol% (E15) ethanol-diesel blends were studied at low, medium and high loads. The study showed that higher fuel oxygen content increased ignition

delay, which slowed the combustion phase and decreased the maximum cylinder pressure. Vin çius B [42] et al. conducted experimental studies on ethanol dual-fuel combustion in a large diesel engine operating at 1200 rpm and 25% load, with the aim of reducing combustion losses while reducing NO_x and soot emissions, and considered as important limitation at low load. Jinlin Han et al. [43] evaluated the ethanol content and load dependence on engine performance and emissions. It was found that the combustion and thermal efficiency decreased at low load and the thermal efficiency increased at medium load condition when ethanol was added. The main objective of this study is to investigate the maximum ethanol replacement ratio at different loads when the dual fuel engine operates in injection timing of the original diesel engine.

The analysis of the effect of ethanol content on performance under different loading conditions is important for performance calculation and prediction of ethanol content to achieve the desired working performances. The study of the effect of ethanol blending ratio on combustion and performance of diesel-ethanol dual fuel under different load conditions is still not sufficient, especially the study of the relationship between combustion characteristics parameters and ethanol content through simulation analysis is also not sufficient. GT-Power is recognized as the best commercial software for the prediction and calibration of combustion and performance, exhaust emissions of internal combustion engines. In this paper, the effect of ethanol blending ratio on the performance of diesel-ethanol dual fuel engine under different load conditions was investigated using engine simulation tool GT-POWER. The main objective of the paper is to analyze through simulation analysis the effect of ethanol blending ratio on engine performance, combustion and emission characteristics under different load conditions of diesel-ethanol blend engine. The paper describes the modeling of a dual fuel engine using the two-stage combustion model of GT-Power, model validation through test data, analysis of the variation characteristics of various performance and combustion parameters, and further analysis for further research to improve the performance.

2. Methods

2.1. Fuel Preparation

In this simulation, a homogeneous diesel-ethanol blend with no emulsifier or stabilizer is considered. The volume fraction of ethanol in the diesel-ethanol fuel blend is 5, 10, 15 and 20%. The physical properties of diesel and ethanol are shown in Table 1.

Table 1. Fuel specification [12, 25].

Fuel specification	unit	diesel	ethanol
Molecular formula	-	C ₁₄ H ₃₀	C ₂ H ₅ OH
Molecular weight	g	198.4	46.068
Cetane number	-	51	8
Research octane number	-	15-25	129
Boiling point	K	453-643	351
Liquid density	kg/m ³	840	789
Lower heating value	MJ/kg	42.5	26.9
Heat of evaporation	kJ/kg	243	918
Self-ignition temperature	K	503	698
Stoichiometric air-fuel ratio	-	14.6	9.06
Viscosity (at 25°C)	mPa s	2.419	1.078
Carbon content	%	85	52.2
Hydrogen content	%	15	13
Oxygen content	%	0	34.8

2.2. Simulation

The simulation was conducted on a marine medium-speed four-stroke diesel engine as in study [39]. The specifications of the simulated engine are shown in Table 2.

Table 2. Engine specifications and boundary conditions.

Type	Value	Type	Value
Bore × stroke (mm)	190/210	Compression ratio	14:1
Number of cylinders	4	Head temperature (K)	553
Brake power (kW)	220	Piston temperature (K)	423
Nozzle radius (mm)	0.26	Wall temperature (K)	433
Fuel injection holes	8	Temperature at IVC (K)	341
Cylinder Pressure (MPa)	2.05	Pressure at IVC (bar)	1.97

Type	Value	Type	Value
Connecting rod (mm)	410		

The engine model using GT-POWER is shown in Figure 1.

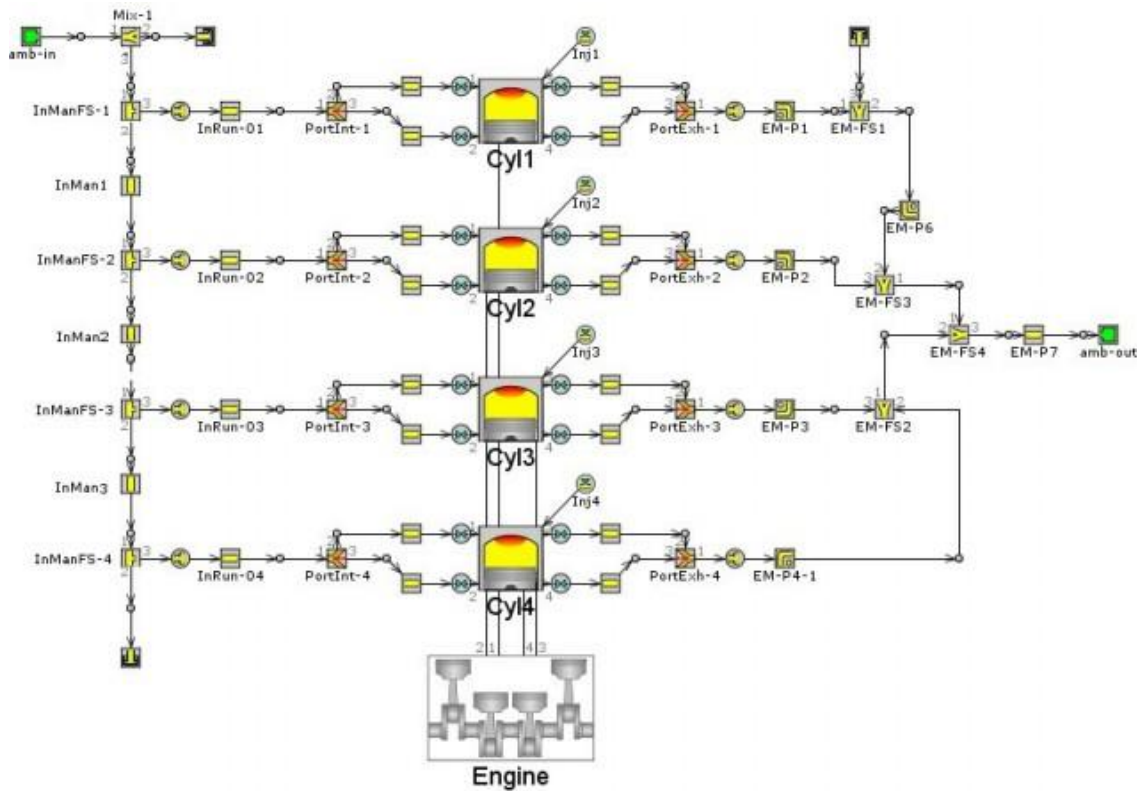


Figure 1. Engine model in GT-POWER.

The fluid flow in the tubes was modeled in one dimension by a discharge coefficient defined as the initial data.

2.2.1. Convective Heat Transfer Model

The calculation of convective heat transfer in cylinders can be done in various ways, using the 'Woschni' method [44, 45].

$$Q = hA(T_g - T_w) \quad (1)$$

Where Q is convective heat transfer flow, A is Heat Transfer Area, h is convective heat transfer coefficient, T_w is Temperature of the cylinder wall, T_g is gas temperature inside the cylinder.

2.2.2. Combustion Model - Two Phase Combustion Model

Analyzing the indicator diagram, it is reasonable to consider the two-zone-two phase combustion process [44].

The combustion process is considered to consist of the following steps:

- 1) The duration of ignition delay required for fuel from a high-pressure fuel pump to be injected into the cylinder and to rise to the auto-ignition temperature, ϕ_i .
- 2) The premixed combustion duration during which fuel heated to the ignition temperature undergoes rapid combustion through oxidation reaction with air, ϕ_p .
- 3) The relatively slow combustion lasts a long time, i.e. the diffusion combustion duration ϕ_d (post-combustion reaction).

Hence it is important to study the heat release characteristics according to the main engine parameters in order to perform numerical simulations of the engine operation.

The combustion rate of diesel engine can be calculated by the following equation:

$$\frac{dx}{d\phi} = 6.908 \frac{x_p}{\phi_p} (m_p + 1) \left(\frac{\phi - \phi_{ig}}{\phi_p} \right)^{m_p} \exp \left[-6.908 \left(\frac{\phi - \phi_{ig}}{\phi_p} \right)^{m_p + 1} \right] + 6.908 \frac{x_d}{\phi_d} (m_d + 1) \left(\frac{\phi - \phi_{ig}}{\phi_d} \right)^{m_d} \exp \left[-6.908 \left(\frac{\phi - \phi_{ig}}{\phi_d} \right)^{m_d + 1} \right] \quad (2)$$

where ϕ_p, ϕ_d are premixed-combustion duration, diffusion combustion duration, x_p, x_d are the mass fraction of burned fuel in Premixed-combustion and diffusion combustion, m_p, m_d are Wiebe Exponent in Premixed-combustion and diffusion combustion, ϕ is Instantaneous Crank Angle, ϕ_{ig} is Start angle of combustion. The above equation is derived from the synthesis of two Wiebe functions, assuming that Premixed-combustion and diffusion combustion occur simultaneously. The figure shows the multi-stage Wiebe combustion input window in GT.

2.2.3. Ignition Delay Duration

The usual Arrhenius equation is used for the calculation of ignition delay [44].

The duration of ignition delay is calculated by the following Arrhenius equation:

$$\tau_{ID} = Ap^{-n} \exp(E/RT) \quad (3)$$

where E-is Activation Energy of the fuel, R is universal gas constant, p is gas pressure, T is gas temperature, A, n are constant.

2.2.4. Engine Friction Model

The friction model of the engine was used by Chen-Flynn model [46].

The frictional loss pressure by the Chen-Flynn model is calculated as follows:

$$FMEP = C + (PF \cdot P_{max}) + (MPSF \cdot \text{Speed}_{mp}) + (MPSSF \cdot \text{Speed}_{mp}^2) \quad (4)$$

where FMEP is Friction Mean Effective Pressure, P_{max} is Maximum Cylinder pressure, Speed_{mp} is Mean Piston Speed, C is Constant part of FMEP, PF is Peak Cylinder Pressure Factor, MPSF is Mean Piston Speed Factor, MPSSF is Mean Piston Speed Squared Factor.

2.2.5. Exhaust Gas Model

The virtual engine simulation tool GT-POWER is capable of modeling exhaust materials [45, 46]. The individual exhaust gases, such as NOx, CO, CO₂, etc., are calculated using chemical kinetics and are designed to be modified by giving correction factors taking into account various operating conditions. The NO calculation is based on the extended Zeldovich mechanism. k_1 , k_2 , and k_3 are the rate constants that are used to calculate the reaction rates of the three equations below.

N₂ oxidation rate equation: $O + N_2 = NO + N$

N oxidation rate equation: $N + O_2 = NO + O$

OH reduction rate equation: $N + OH = NO + H$

3. Results and Discussion

3.1. Analysis and Comparison of Engine Performance

3.1.1. Cylinder Pressure Curve

The analysis and comparison of engine performance are carried out with the emphasis on the analysis of the effect of ethanol blending ratio on combustion characteristics and performance under different load conditions. The results of the indicator diagram obtained in the analysis reflect the combustion process, such as heat release rate, ignition delay, combustion duration, etc., and can be used for the evaluation of the performance of the test engine. Figure 2 shows the pressure profile in the engine chamber versus the crank angle variation with ethanol content at 100% engine load. The pressure curves in Figure 2 show that with increasing ethanol content, the auto-ignition delay effect occurs, which is mainly responsible for the addition of low cetane number ethanol. With the observation of ignition delay, the combustion process is accompanied by a sudden increase of heat release rate and pressure rise. The pressure profile data obtained in the simulation are in agreement with the test results of [39]. The maximum pressure value for diesel-ethanol blends (D80E20)

at 100% engine load is 127 bar, which is satisfactory with an error of less than 5% compared to the test value of [39].

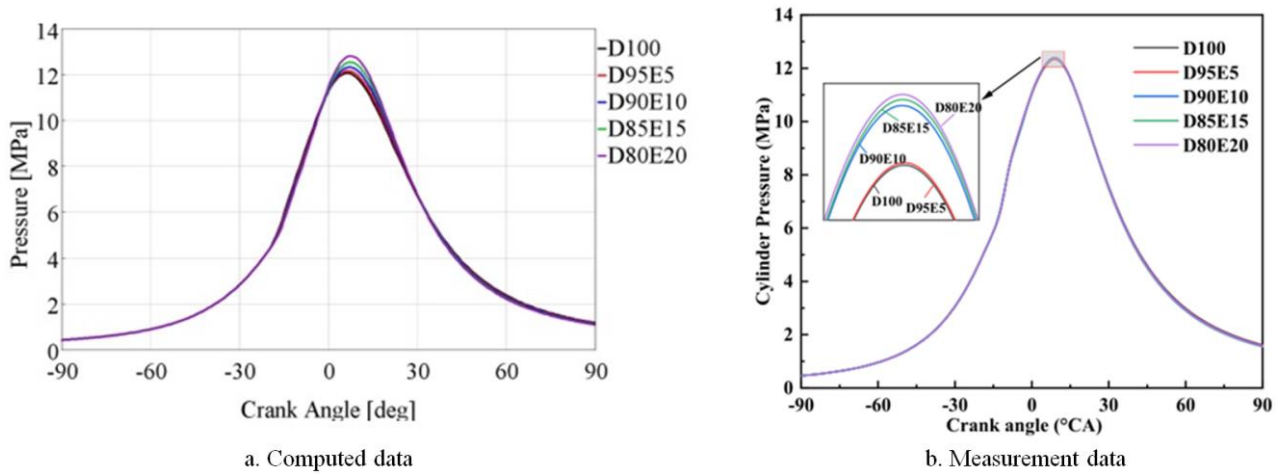


Figure 2. Pressure in the engine combustion chamber versus the crank angle with ethanol content at 100% engine load.

3.1.2. Cylinder Temperature Curve

Figure 3 shows the temperature profiles in the engine combustion chamber versus the crank angle variation with ethanol content at 100% engine load. The increase in ethanol content in diesel-ethanol blends causes an increase in combustion rate and an increase in maximum combustion tem-

perature. The simulations show that the maximum combustion temperature increases by 1.06%, 2.1%, 3.13% and 4.09% compared to diesel fuel when the ethanol content increases by 5%, 10%, 15% and 20%. The temperature profiles obtained in the simulation are in a trend agreement with the test results of [39].

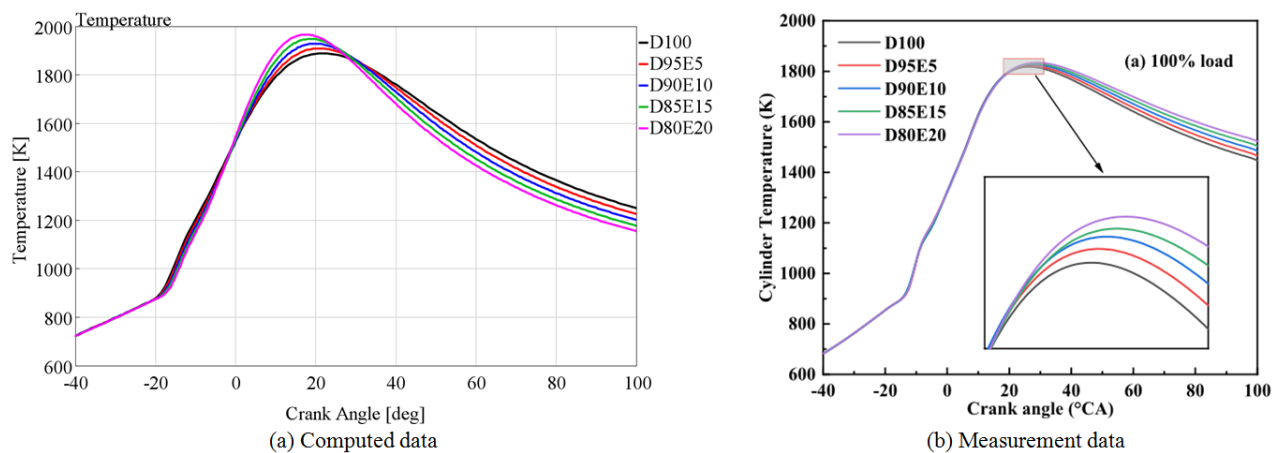


Figure 3. Heat release during combustion at 100% engine load.

3.1.3. Heat Release Rate

Figure 4 shows the heat release rate (HRR) during combustion of diesel-ethanol blend at 100% engine load. The simulation results show that the addition of ethanol to diesel fuel leads to prolonged ignition delay and increased heat release rate. The ignition delay increase moves to the main

combustion stage, the diffusion combustion stage, causing an increase in the heat release rate. The increase in ignition delay is caused by the cooling effect due to the addition of ethanol with a high latent heat of vaporization and a low cetane number. The highest values of HRR were obtained for D80E20 (see Figure 4).

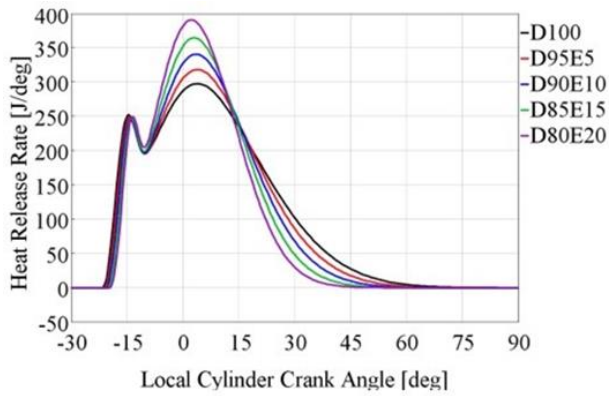
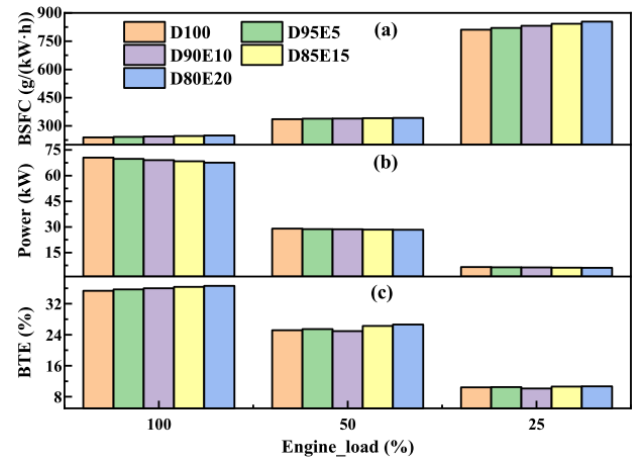
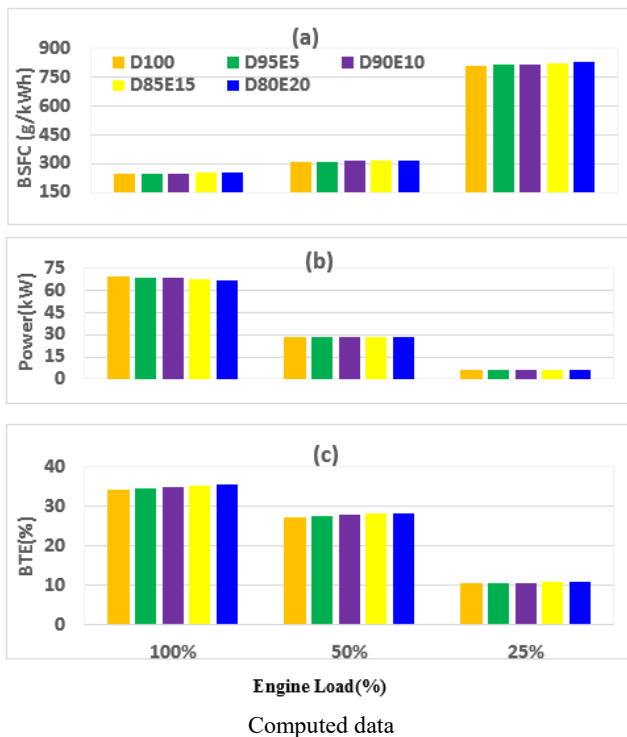


Figure 4. Heat release rate (HRR) during combustion of diesel-ethanol blends.

3.1.4. Brake Specific Fuel Consumption

In this study, one of the main characteristics of engine operation using diesel-ethanol blend was analyzed for fuel consumption (BSFC). Figure 5a shows the effect of ethanol content on the fuel consumption (BSFC) at different engine loads.

In diesel-ethanol blends, the BSFC showed an increasing trend with decreasing engine load and increasing ethanol content. When the ethanol content increases by 5% at 25% engine load, the BSFC becomes 808.6, 811.1, 814.9, 820.7, and 820.8 g/(kW · h). The fuel consumption values obtained in the simulation are in agreement with the test results of [39].



Measurement data [39]

Figure 5. Effects of diesel/ethanol blended fuels with different blending ratios on (a) BSFC, (b) power and (c) BTE under different loading conditions.

3.1.5. Brake Power

Figure 5b shows the effect of ethanol content on the brake power at different engine loads. In diesel-ethanol blends, the brake power showed a decreasing trend with decreasing engine load and increasing ethanol content. At 100% load, the brake power decreases by 0.64, 1.46, 2.48, and 3.73% compared to diesel when the ethanol content increases by 5%. The simulated brake power data are in agreement with the performance test results of [39] with an error of 5%.

3.1.6. Brake Thermal Efficiency

Figure 5c shows the effect of ethanol content on the brake thermal efficiency (BTE) at different engine loads. The BTE showed an increasing trend with increasing ethanol content in diesel-ethanol blend. Ethanol contains oxygen, which improves the combustion characteristics of diesel-ethanol blends, thereby increasing the BTE.

On the other hand, the BTE decreases when the engine load decreases. At the same speed, the mechanical loss power is constant and the reduction of engine load causes a relatively increase in the proportion of mechanical losses in the total indicated energy, resulting in a decrease in the BTE. The simulated brake power data are in agreement with the performance test results of [39].

3.2. Effect of Ethanol Content on Engine Performance Under Different Loading Conditions

3.2.1. Effect of Ethanol Content on Engine Brake Power

We simulated the change of the indicator efficiency with the change of engine load and ethanol content. The engine

load varied from 25 to 100% and ethanol content from 0 to 20%. The performance and emission characteristics of ethanol addition under different loading conditions with constant engine load were analyzed.

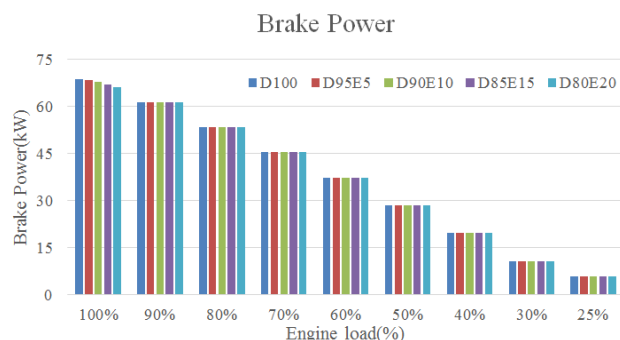


Figure 6. Effects of diesel/ethanol blended fuels with different blending ratios on brake power under different loading conditions.

As can be seen from simulations and experiments, the engine power decreases with increasing ethanol content. However, as the engine load decreases, the effect of engine power variation with ethanol content tends to decrease further.

3.2.2. Analysis of the Effect of Ethanol Content on Indicated Thermal Efficiency (ITE), Brake Thermal Efficiency and Specific Fuel Consumption

The ITE is an important index to evaluate the thermodynamic performance of an engine. Figure 7 shows the variation of the ITE with the change of engine load and ethanol content. The ITE first increases with decreasing engine load and reaches a maximum value at intermediate load (50-60% load) and then decreases. The lower engine load leads to a lower fuel delivery and thus a lower fuel injection time. This leads to the shortest combustion time and the lower char combustion fraction, which increases the combustion efficiency. Thus, the ITE at a constant fuel feed rate reaches a maximum.

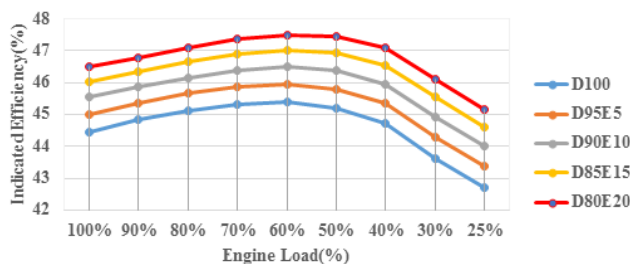


Figure 7. Effects of diesel/ethanol blended fuels with different blending ratios on ITE under different loading conditions.

However, the characteristics of the ITE change with etha-

nol content as the engine load decreases, which tend to increase and decrease, therefore, it is found that there exists a loading condition where the ITE is maximized. As can be seen from simulations and experiments, the BTE usually increases with increasing ethanol content. However, as the engine load decreases, the engine efficiency change with ethanol content tends to be almost absent.

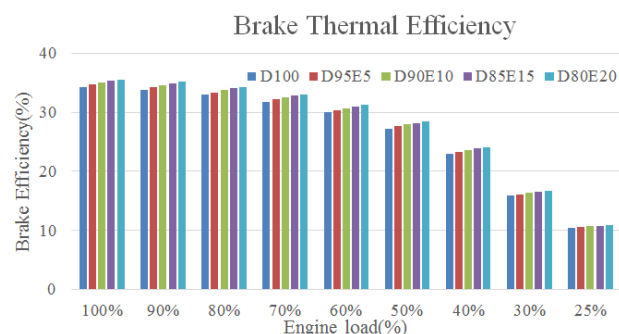


Figure 8. Effects of diesel/ethanol blended fuels with different blending ratios on BTE under different loading conditions.

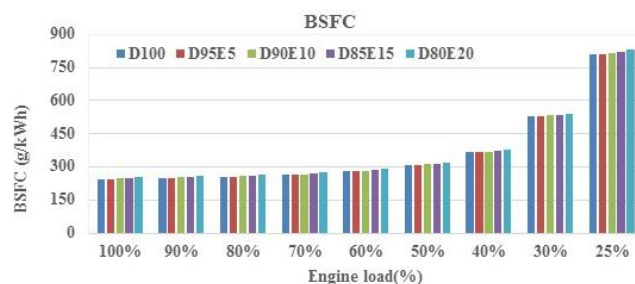


Figure 9. Effects of diesel/ethanol blended fuels with different blending ratios on specific fuel consumption under different loading conditions.

As the engine load decreases, the specific fuel consumption increases. With the decrease to the load of the medium (50%) level, the specific fuel consumption increases slightly compared to the 100% load case, but with the decrease of the load to the medium (50%) level, the specific fuel consumption increases rapidly.

3.3. Analysis of NOx Emission

NOx emission shall be low during engine operation, especially NOx emission shall be low while providing the required power. Hence, we focus on the NOx emission per unit power with the change of engine load and ethanol content. The effect of different blending ratios of diesel-ethanol blends on NOx emissions under different load conditions is shown in Figure 10. The analysis results show that the engine load decreases as the engine load decreases, as shown in [39]. Furthermore, the NOx emission of the engine increases as the ethanol ratio in the blend fuel increases. Because ethanol has low density, low

viscosity and high oxygen content characteristics, it improves combustion in the engine cylinder and produces more NOx in the local high temperature region.

Also, the cetane number of diesel/ethanol blend is lower than 1100, which results in prolonged ignition delay, increased in-cylinder fuel injection, and premixed combustion, thereby increasing NOx emissions.

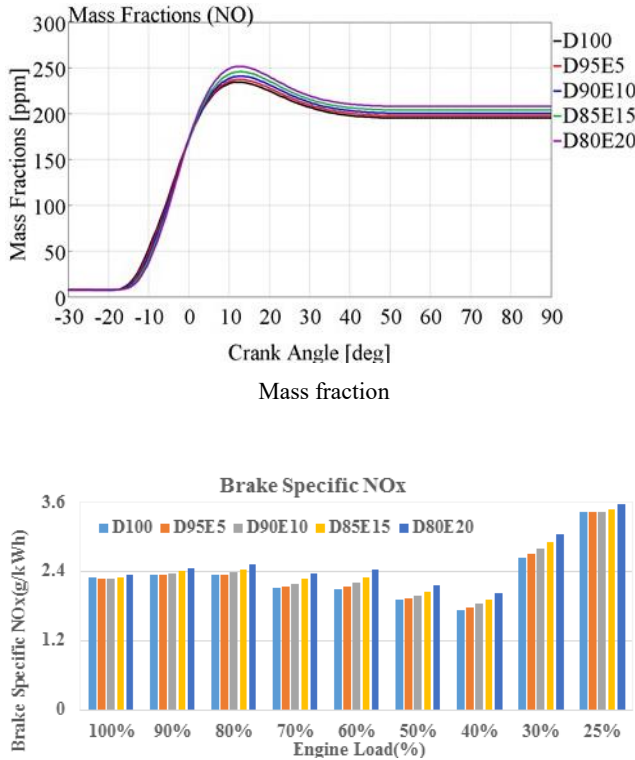


Figure 10. Effects of diesel/ethanol blended fuels with different blending ratios on NOx emission under different loading conditions.

The NOx consumption decreases first when the engine load decreases, reaches a minimum value at 40-50% load, and then increases. The lower engine load lowers the maximum combustion temperature, which reduces the NOx emission and also decreases the power output.

However, the NOx emission reduction rate is greater than the power reduction rate, so the NOx consumption is reduced. When the engine load is less than a certain value, the power output decreases rapidly and the NOx emission increases. The study shows that there exists a minimum engine load condition for NOx consumption.

3.4. Analysis of the Combustion Characteristics

The variation of premixed combustion fraction with ethanol content is shown in Figure 11.

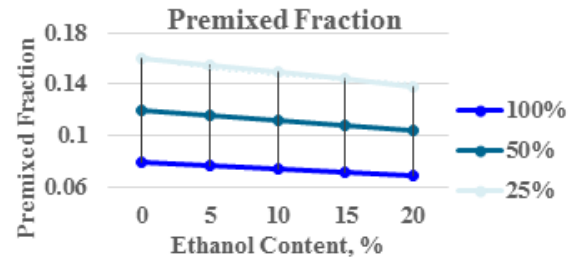


Figure 11. Premixed combustion fraction with ethanol content.

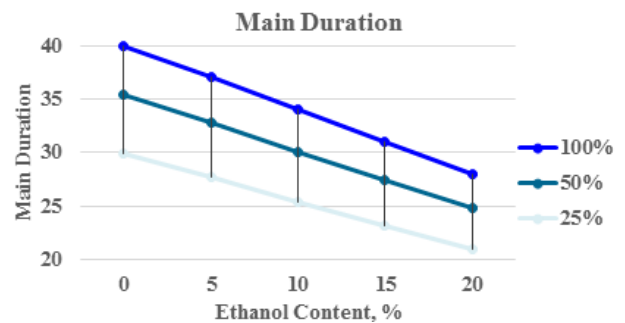
The simulation results show that the premixed combustion fraction decreases with increasing ethanol content.

It can be seen that the premixed combustion ratio decreases from 8% to 6.9% when the ethanol content increases at 100% load.

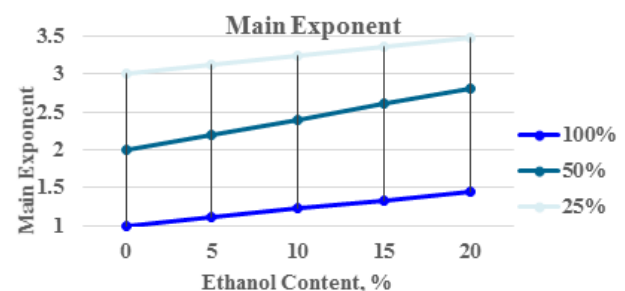
The same trend is observed for 50% and 25% load regimes.

Due to the low premixed combustion ratio, the effect of premixed combustion on engine characteristics is not significant, and hence the diffusion combustion process is considered as the main consideration.

The variation of the diffusion combustion duration (or main duration) and diffusion combustion exponent (main exponent) with ethanol content is shown in Figure 12. It can be seen that the increase of alcohol content and the resulting increase of reactive oxygen species increases the temperature of the medium in the reaction zone at the beginning of the combustion process, while at the same time increasing the number of ignition origin points.



(a) Main duration



(b) Main exponent

Figure 12. Main combustion duration and combustion exponent with ethanol content.

In particular, diesel-ethanol blends are injected, initially by the latent heat of vaporization and by the autoignition temperature of high cetane number diesel, which is first auto-ignited, forming the ignition source. Alcohol is also burned, and the combustion of alcohol is considered to be the same as that of alcohol in the spark-ignition engine. The ignition source formed by diesel fuel acts as an ignition spark and the alcohol-air mixture, which is injected, heat exchanged, evaporated, and mixed with air, is combusted as in the spark ignition engine.

In turbocharged engines, higher cylinder pressure and temperature at the start of injection lead to lower ignition delay, lower amount of fuel injected during that period, and lower heat exchange time, which leads to a decrease in the proportion of combustible alcohol fuel.

The diffusion combustion duration (or main duration) decreases and the diffusion combustion exponent (main exponent) increases.

The main duration decreases and the main exponent increases with the increase of the fully evaporated fraction of combustible alcohol combustion during diffusion combustion.

Thus, as the combustion intensity increases, pressure and temperature rise rapidly, which leads to an increase in NO_x emission, this trend is independent of load and is consistent with the performance test results of [25, 39].

4. Conclusions

The effect of ethanol blending ratio on the performance of diesel-ethanol dual-fuel engine under different loads (25-100%) was investigated using engine simulation tool GT-POWER.

The following important conclusions were obtained:

- 1) As the load decreases, the effect of ethanol blending ratio on the effective power becomes insignificant.
- 2) With the load decreasing to less than medium (50%) the specific fuel consumption increases rapidly.
- 3) Indicated thermal efficiency (ITE) and NO_x emissions per unit power output are usually optimal at medium load.
- 4) It can be seen that the premixed combustion fraction increases with decreasing load. At 100% load conditions, the diffusion combustion duration (main duration) decreases and the diffusion combustion exponent (main exponent) increases with increasing ethanol content. Under lower load conditions (with the same ethanol content), the main duration decreases and the main exponent increases.

From the practicality and reliability of the results, it is expected that the future will be used for the performance prediction and the optimal design of the diesel-ethanol dual-fuel engine at different loads using the two-stage combustion model of pre-combustion-diffusion combustion of GT-Power.

Abbreviations

CN	Cetane Number
HC	Hydrocarbons
BD	Burn Duration
BTE	Brake Thermal Efficiency
HR	Heat Release
HRR	Heat Release Rate
ITE	Indicated Thermal Efficiency
ID	Ignition Delay
IMEP	Indicated Mean Effective Pressure
TDC	Top Dead Centre

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Conflicts of Interest

The authors declare no conflicts of interest.

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