Study on combustion properties of diesel engine fueled with diesel-ethanol blend using two phase combustion model of GT-SUITE

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Abstract— Worldwide, there is a growing interest in the use of diesel-ethanol blends in compression ignition engines as a way to reduce vehicle emissions and use alternative fuels. The previous studies with the focus on the reduction of emission and the improvement of performance for compression ignition engine using ethanol shows that the combustion characteristics should not be missed. In this paper, the effect of ethanol content on combustion properties of diesel engine fueled with diesel-ethanol blends using two phase combustion model (combustion model of premixed -diffusion combustion) of the virtual engine simulation tool GT-SUITE was investigated. For the purpose of the study, the engine simulation tool was used to compare the Computed data with the benchmark data for the variation of ethanol content (0-40%) and to analyze the characteristics of auto-ignition, premixed combustion and diffusion combustion. The simulation results of combustion characteristics using diesel-ethanol blend showed that the auto-ignition delay increased and the premixed combustion ratio increased. The results of the study showed that the combustion model of premixed combustion -diffusion combustion can be used to perform combustion characteristics and performance analysis in a compression ignition engine using ethanol-diesel blend and to estimate the ethanol content to achieve the desired working performances.

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Keywords— **GT-SUITE; ethanol content; two phase combustion; auto-ignition; premixed combustion; diffusion combustion**

I. INTRODUCTION

Due to the reduction of fossil fuels and the emission of harmful gases, there were an increased interest in searching for alternative fuels with which can replace diesel fuel partially, or completely.

Recent years have seen the research on opportunities to use alternative fuels based on bio-alcohols such as ethanol, methanol and butanol in a compression ignition engine. [1-5] The use of ethanol-based fuels as diesel fuels can reduce engine emissions as they have lower carbon and sulfur contents, and higher oxygen content compared to fossil fuels. It was experimentally verified that the use of diesel-ethanol blends in a compression ignition engine could reduce the emissions of CO, soot and NO x.

In this paper, through simulations based on the dual combustion model of pre-combustion-diffusion combustion, the effect of ethanol content variation on the combustion characteristics of auto-ignition, pre-mixed combustion and diffusion combustion is analyzed and the performance is compared with that of a standard diesel fuel engine.

II. BACKGROUND AND PREVIOUS WORK

Many studies indicated advantages and limitations about the characteristics of diesel-ethanol blends and performance and emission characteristics in terms of their use. [6-19] Performance tests of a compression ignition engine revealed the possibility to reduce the CO, NO x and soot in the exhaust gas under the generally specified engine operating conditions, compared with the case of using only the diesel fuel.

Oliveira et al. [20] studied the effect of diesel-biodiesel-ethanol blend (diesel with 7% biodiesel, 5, 10, and 15% ethanol) on the performance, emissions and combustion characteristics of diesel power generator. The results showed that the in-cylinder peak pressure and heat release rate were decreased at low loads and increased at high loads with the use of ethanol. Increasing ethanol concentration caused the increased ignition delay, the decreased combustion duration and the reduced exhaust gas temperature. Praptijantoa A [21] used the engine simulation analysis tool AVL Boost to document the effect of ethanol content on the performance and exhaust emissions of the fuel blend of diesel and ethanol. The one-zone combustion model of AVL Boost was used, the mixing mode between ethanol and diesel 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.

The addition of ethanol to diesel engines has a great influence on the variation of the physicochemical parameters of diesel fuel, and the results of studies have been reported. [22-30]Of the many physicochemical parameters, particular attention should be paid to the auto-ignition propensity and associated ignition delay duration. It was found that the latent heat of evaporation of ethanol, calorific value, cetane number, etc. caused negative changes in combustion characteristics and performance, which may have different limitations. It has been reported [31-34] 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 calorific value of pure ethanol is about 60% of diesel fuel, but the heat of evaporation is over three times higher (840 kJ/kg) for ethanol compared to diesel fuel (260 kJ/kg). The increase in alcohol content increases the evaporation heat of the blend and affects the delay of the combustion process. In compression ignition engines, this leads to the ignition delay, reduced the rate of pressure rise, thus limiting ‘‘hard” engine operation. [33, 34] Also, increasing ethanol concentration increases ignition delay, decreases combustion duration and decreases exhaust gas temperature. It was also analyzed that high oxygen content of ethanol increases the air-fuel ratio of the blend fuel and changes the combustion efficiency. [35]

The low certain number of ethanol-diesel blends increases the ignition delay and heat release rate, which is related to the ethanol content in the blends. [36] The low cetane number (CN) of ethanol decreases the cetane number of the blend and is disadvantageous because of the prolonged ignition delay.

Studies on the possibility of adding ethanol to diesel fuel have clearly revealed that with increasing ethanol, the auto-ignition characteristics of the fuel deteriorate. In [37], the ignition delay and combustion delay were mainly considered to evaluate the auto-ignition characteristics, and the effect of gas medium temperature on ignition delay and combustion delay duration were determined. Of the many physicochemical parameters, particular attention should be paid to the auto-ignition property. [28-31]

In addition, the analysis of the exhaust emissions of ethanol in a compression ignition engine shows that the premixed combustion characteristics are of significant concern. In [32], engine tests revealed that the increase of CN of fuel has 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 NOX 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. As indicated in many studies, the shorter ignition delay time in diesel engines reduces the amount of fuel accumulated in the combustion chamber, thus reducing the reaction intensity during premixed combustion. [33,34] This can also be said to be the case for lower pressure rise rates. This reduces engine noise, reduces the intensity of heat transfer from the working medium, and causes a reduction effect on NOX emission from the exhaust gas. This characteristic is generally known for conventional fuels, especially diesel fuels.

Some studies carried out on a compression ignition engine using diesel-ethanol blends show similar characteristics in general. The study [35] shows that due to the extension of ignition delay time, the increase of ethanol content in diesel fuel causes irregularity of engine operation and increases vibration.

On the other hand, from the study [36], 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 emission during premixed combustion. It was found by Gordon et al. [37] that more fuel was accumulated before the start of combustion, so the premixed combustion process was more intensive, resulting in higher heat release rates. It was also analyzed that with the increase peak pressure values, a maximum temperature increase occurs, resulting in an increase in NO x emissions. Murcak A [38] found that injection timing is an important engine operating parameter that affects combustion and performance of diesel engines. The injection timing was analyzed to improve premixed combustion duration and reduce fuel consumption [39]. Murcak A [40] determined the effect of injection timing on engine performance in a single cylinder diesel engine with 5%, 10%, and 20% contents of ethanol by volume. in diesel fuel. Here, an adjustable cam mechanism was used in the test engine to change the injection time. J.M. Herreros et al. [41] found that the addition of additive DGE 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 the NO x emission, with the decrease of premixed combustion fraction with the decrease of emission due to the increase of oxygen content. Hubert Kuszewski et al. [42] investigated the effect of the addition of 2-ethylhexyl nitrate (2-EHN) on the auto-ignition performance of ethanol-diesel blends (EDB). Based on the pressure processes, the maximum pressure rise as well as the average and maximum pressure rise rate in the combustion chamber were analyzed. Liu et al. [43] showed that with increasing 2-EHN content, the ignition delay time decreases, resulting in a decrease in the heat release rate in the premixed combustion phase.

III. METHODS

A. *FUEL PREPARATION*

The analysis of the combustion step characteristics and the effect of ethanol content on the performance of the combustion stage and the use of diesel-ethanol blend in the engine is important for the calculation of the performance and the prediction of ethanol content to meet the required working performances. The effect of ethanol content on the stage-wise combustion characteristics of auto-ignition, premixed combustion and diffusion combustion in a diesel-ethanol hybrid engine has been simulated and compared with the benchmark data has not been reported yet.

In this paper, through simulations based on the dual combustion model of pre-combustion-diffusion combustion, the effect of ethanol content variation on the combustion characteristics of auto-ignition, pre-mixed combustion and diffusion combustion is analyzed and the performance is compared with that of a standard diesel fuel engine. In this simulation, a homogeneous diesel-ethanol blend with no emulsifier or stabilizer is considered. The volume fraction of ethanol in the blend is 10, 15, 20, 25, 30, 35, and 40%. The physical properties of diesel and ethanol are shown in Table 1.

Table 1. Fuel specification [10,29]

Fuel specification	unit	diesel	ethanol
Molecular formula	-	$C_{14}H_{30}$	C_2H_5OH
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℃)	mPa s	2.419	1.078
Carbon content	%	85	52.2
Hydrogen content	%	15	13
Oxygen content	%	0	34.8

B. Simulation

The simulation used a 1 CA90 cylinder four-stroke diesel engine operating at a rotational speed of 1500 rpm manufactured by Androlya Mott (Poland). [29,30] The simulation also set the optimum angle of fuel injection to 17°ATDC if the optimum angle of fuel injection is operated at a constant rotational speed of 1500 rpm, as in [29].

The specifications of the simulated engine are shown in Table 2.

Table 2. Engine main specifications

Parameter	
Type of engine	Four stroke compression ignition
Number of cylinders	1
Displacement volume(cm^3)	573
Engine rotational speed(rpm)	1500
Bore×Stroke(mm×mm)	90×90
Compression ratio	17:1
Injection pressure(MPa)	21
Injection timing(deg)	17bTDC
Rated power(kW)	7

GT-SUITE 2016 was used for the simulation.

GT-SUITE2016 can be used for the study of two- or four-stroke engines and spark ignition engines or compression ignition engines.

The engine model using GT-SUITE is shown in Fig. 1.

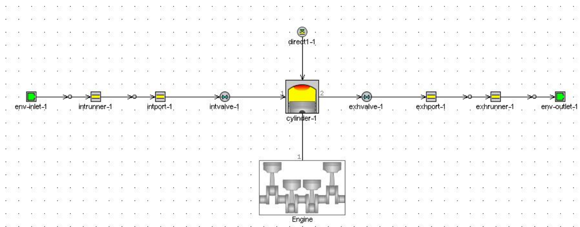


Fig. 1 Engine model in GT-SUITE

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

C. 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.

D. Combustion model - Two phase combustion model

Analyzing the in-cylinder pressure trace, 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.

②The 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:

$$\frac{dx}{d\varphi}=6.908\frac{x_p}{\varphi_p}(m_p+1)\left(\frac{\varphi-\varphi_{ig}}{\varphi_p}\right)^{m_p}\exp\left[-6.908\left(\frac{\varphi-\varphi_{ig}}{\varphi_p}\right)^{m_p+1}\right]$$
$$+6.908\frac{x_d}{\varphi_d}(m_d+1)\left(\frac{\varphi-\varphi_{ig}}{\varphi_d}\right)^{m_d}\exp\left[-6.908\left(\frac{\varphi-\varphi_{ig}}{\varphi_d}\right)^{m_d+1}\right] \quad (2)$$

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 Vibe functions, assuming that premixed-combustion and diffusion combustion occur simultaneously.

The figure shows the multi-stage Vibe combustion input window in GT.

E. Ignition delay

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.

F. 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:

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

Where FMEP is Friction Mean Effective Pressure, Pmax is Maximum Cylinder pressure, Speedmp 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

G. Exhaust gas model

The virtual engine simulation tool GT-SUITE is capable of modeling exhaust materials. [45]

The individual exhaust gases, such as NO x, CO, CO 2, etc., are calculated using chemical kinetics and are designed to be modified by giving correction factors taking into account various operating conditions.

The reaction mechanism and calculation of the individual exhaust material used in GT are presented in [44].

Ⅳ Results

A. Engine performance analysis and comparison

Cylinder pressure curve

Engine performance analysis and comparative analysis were carried out on the basis of the analysis of the combustion characteristics of the auto-ignition, pre-mixed combustion and diffusion combustion of fuels according to a variation of ethanol content.

The results of the in-cylinder pressure trace 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.

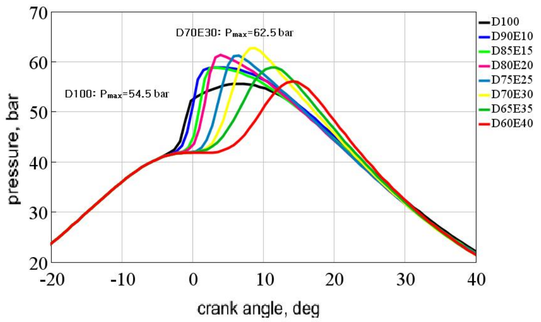
Fig. 2 shows pressure profiles in the engine cylinder versus changes in crank angles for ethanol percent contents in blends with diesel fuel.

The in-cylinder pressure traces in Fig. 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.

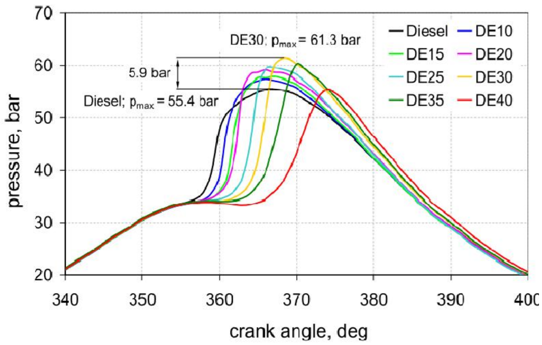
With the observation of ignition delay, the combustion process is accompanied by a sudden increase of heat release rate and pressure rise.

The pressure profile data obtained in the simulation are in agreement with the test results of [29,30].

The maximum pressure value for diesel-ethanol blends is 62.5 bar for DE30, which is satisfactory with an error of less than 5% compared to the test value of [29].



a) Computed data

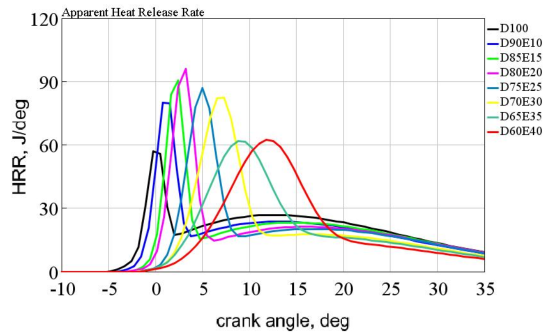


b)Measurement data

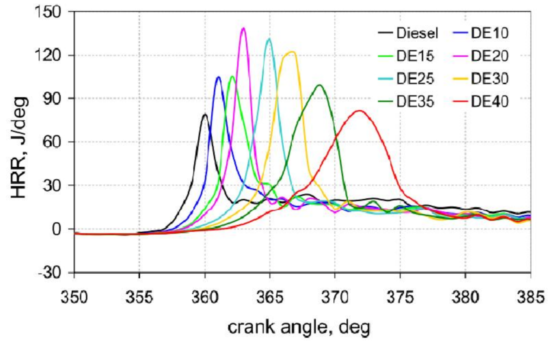
Fig. 2. In-cylinder pressure traces of the engine for combustion of diesel-ethanol

Heat release rate

Fig. 3 shows the heat release rate (HRR) during combustion of diesel-ethanol blends.



a) Computed data



b)Measurement data

Fig. 3. Heat release rate (HRR) during combustion of diesel–ethanol blends

The simulation results show that the addition of ethanol to diesel fuel lead to the prolongation of ignition delay and the increase of heat release rate.

Increasing ignition delay moves to the diffusion combustion stage, the diffusion combustion stage, leading to 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 DE20 (see Fig. 3).

Increasing the ethanol content by over 30% leads to a decrease in the heat release rate.

It can be seen that the increase in ethanol content with high oxygen content is the main reason.

Heat release

Fig. 4 shows the emission heat curve Q norm versus crank angle.

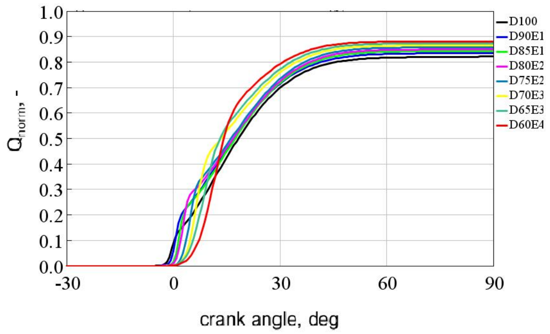
Besides the heat release rate (HRR), the important part of the analysis of combustion characteristics is the heat release curve during combustion.

The heat release curve can be used to determine two characteristic phases of the combustion process in a diesel engine: ignition delay (ID) and combustion duration (BD).

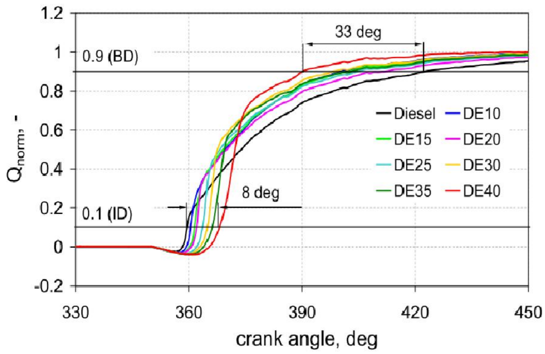
The combustion characteristics of the fuel in a compression ignition engine are characterized by a relatively short ignition delay duration and a relatively long combustion duration.

The increase in ethanol content in diesel-ethanol blends causes an increase in ignition delay caused by high evaporation heat of ethanol (cooling effect) and compression ignition temperature.

The heat release curves obtained in the simulations are in agreement with the performance test results of [29,30].



a) Computed data



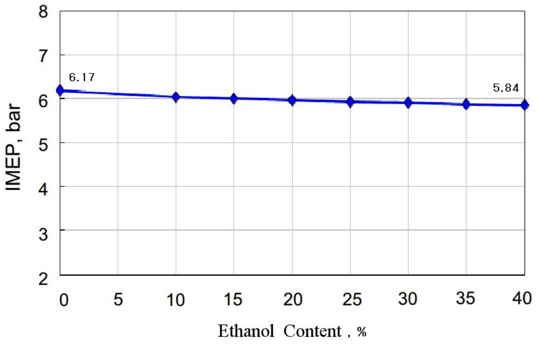
b)Measurement data

Fig. 4 Emission heat curve with crank angle Qnorm

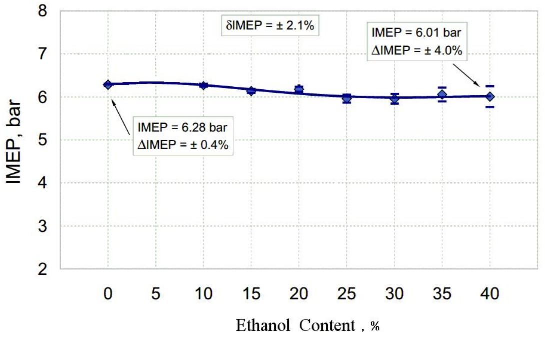
Mean effective pressure and indicator efficiency

In this study, one of the main characteristics of engine operation using diesel-ethanol blend was analyzed, the mean effective pressure (IMEP) and the indicated efficiency.

Fig. 5 shows the effect of ethanol content on the mean effective pressure (IMEP).



a) Computed data



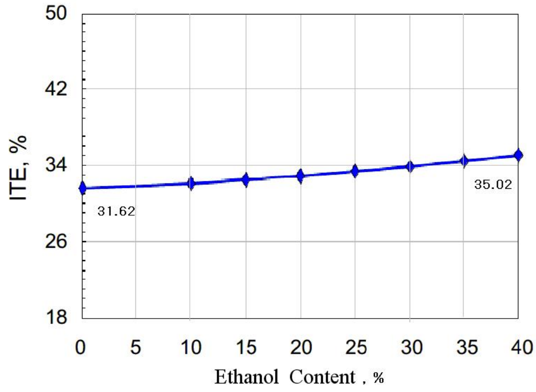
b) Measurement data

Fig.5 Effect of ethanol content on average effective pressure (IMEP)

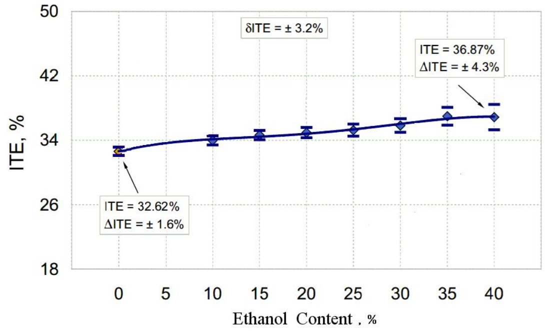
The IMEP shows a decreasing tendency with increasing ethanol content in diesel-ethanol blends.

The blend with 10-30% ethanol content had higher values of indicated mean effective pressure (IMEP).

Fig. 6 shows the effect of ethanol content on the indicated efficiency (ITE) of the engine using diesel-ethanol blend.



a) Computed data



b) Measurement data

Fig.6 Effect of ethanol content on engine pointing efficiency (ITE)

The simulation results show that with an increase of ethanol content, ITE increased from 31.6% to 35% (up to 10.7%).

The heat release curves obtained in the simulations are in agreement with the performance test results of [29,30].in error of 5 percentages.

B. Effect of ethanol content on characteristics of auto-iginition, premixed combustion, and diffusion combustion

Auto-iginition characteristics with ethanol content

With simulation information of characteristics on the Fig 2-5, we can determine change of ignition delay according to the ethanol content. Fig 9 shows the change of ignition delay according to the ethanol content.

With simulation information of characteristics on the Fig 2-5, we can determine change of ignition delay according to the ethanol content. Fig 7 shows the change of ignition delay according to the ethanol content.

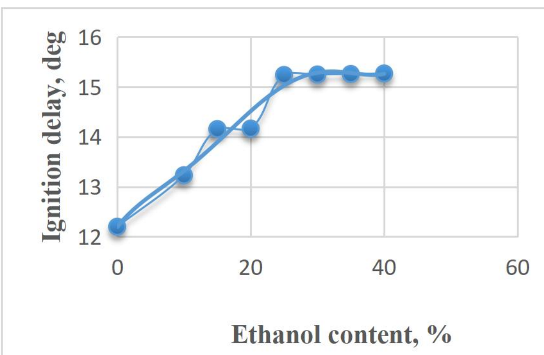


Fig 7. Ignition delay on the variation of crank angle with ethanol content

As increasing the ethanol content in the simulation result, the ignition delay increases. If ethanol content increases to 30% ,the ignition delay increases from 12.3 to 15.5, about 20%. Also if the ethanol content is over 30% , the little ignition delay changes .

Premixed combustion characteristics with ethanol content

The variation of premixed combustion fraction with ethanol content is shown in Fig. 8. The simulation results show that the premixed combustion fraction increases with increasing ethanol content.

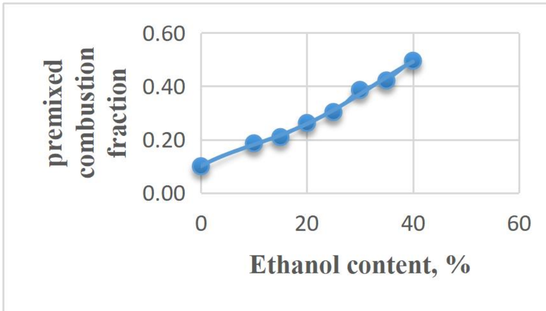


Fig 8. The variation of premixed combustion fraction with ethanol content

It can be seen that the premixed combustion ratio increases from 10% to over 50% when the ethanol content increases to 40%.

The variation of premixed combustion duration with ethanol content are shown in Fig. 9.

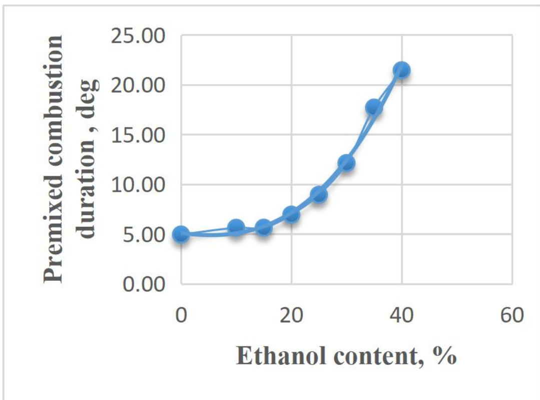


Fig 9. The variation of premixed combustion duration with ethanol content

It can be seen that the increase of ethanol content in the mixture and the consequent increase of reactive oxygen species increase 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 sources.

In particular, diesel-ethanol blends are injected, initially by the latent heat of vaporization and by the auto-ignition temperature of high certain number diesel, which is first auto-ignited, forming the ignition source.

Ethanol is also burned, and the combustion of ethanol is considered to be the same as that of ethanol in the spark-ignition engine. The ignition source formed by diesel fuel acts as an ignition spark and the ethanol-air mixture, which is injected, heat exchanged, evaporated, and mixed with air, is combusted as in the spark ignition engine. Especially, the large ignition delay duration will result in a larger amount of injected fuel and an increase in the heat exchange time, leading to a further increase in the proportion of combustible ethanol fuel. As the premixed combustion fraction increases, pressure and temperature rise rapidly, which leads to an increase in NO x emission, and this trend is consistent with the performance test results of the literature [29,30].

Diffusion combustion characteristics with ethanol content

Considering the change relation between the diffusion combustion duration and diffusion combustion exponent as Fig. 10 and 11.

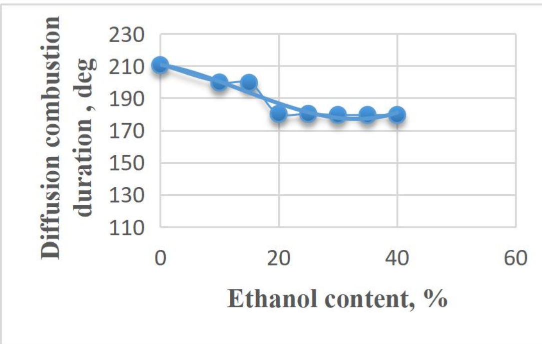


Fig 10 The variation of diffusion combustion duration with ethanol content

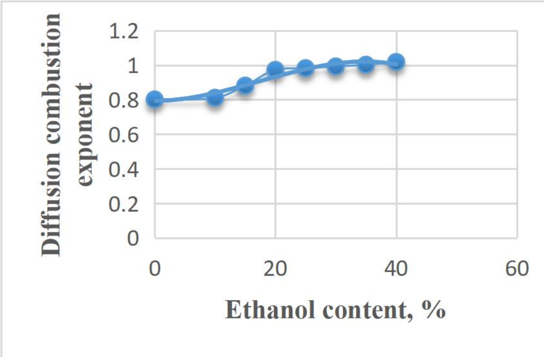


Fig 11 The variation of diffusion combustion exponent with ethanol content

As increasing the ethanol content, the diffusion combustion duration decreases and the diffusion combustion exponent increases. As the ethanol content increases, if premixed combustion fraction increases, then the pressure and temperature increases rapidly and so the combustion condition gets better and the combustion time decreases.

IV. CONCLSION

A simulation study of combustion characteristics in diesel engine fueled with diesel-ethanol blends was carried out in this paper. In diesel engine fueled with diesel-ethanol blends, ethanol increases oxygen content in the fuel mixture, thus affecting stoichiometric demand for air and contributing to changes in individual phases of the combustion process. Analysis of the results obtained in this research leads to the following conclusions:

(1)Using the dual combustion model of pre-combustion-diffusion combustion in the engine simulation tool GT-SUITE, the combustion characteristics and performance of the compression ignition engine using ethanol-diesel blend can be simulated the engine test approximately and can be used for analysis of the effect of ethanol content on combustion characteristics.

(2) Combustion indices change with increasing ethanol content, but change a little when the ethanol content reaches a certain value. In the simulation, the ignition delay duration does not change significantly

when the ethanol content is higher than 25%. Also, when the ethanol content is over 25%, the premixed combustion exponent and the diffusion combustion exponent change a little.

(3) According to increases ethanol content, the premixed combustion duration increases and the diffusion combustion duration decreases.

(4) It can be seen that the addition of over 30% ethanol content improves the combustion characteristics and the ethanol content should be chosen accordingly to achieve the desired performance indices.

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