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A numerical approach in the investigation of the effects of diethyl ether and ethanol mixtures on combustion characteristics and NO emissions in a DI diesel engine

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Abstract: In this study, the effects of adding ethanol and diethyl ether to diesel fuel on combustion characteristics and NO emissions were numerically investigated. Neat diesel fuel and by volume 90% diesel+10% ethanol blend, 80% diesel+20% ethanol blend, 80% diesel+10% ethanol+10% diethyl ether blend and 85% diesel+ 10% ethanol+5% diethyl ether mixtures were used as fuel. Analyzes were carried out using a single-cylinder, air-cooled, four stroke direct injection diesel engine at 2000 and 3000 rpm engine speed conditions. AVL FIRE software was used for numerical study. In-cylinder pressure, cumulative heat release rate, turbulent kinetic energy, NO emissions and velocity distributions in the combustion chamber were investigated for specified fuel. As a result, the in-cylinder pressure and heat release rate of ethanol and diethyl ether blended fuels were lower than diesel fuel at both speeds. It was observed that NO emissions decreased as the ethanol content in the fuel increased. At 3000 rpm, the highest TKE value was obtained in D90E10 mixed fuel, and the lowest value was found in D80E10DEE10 mixture fuel. Ethanol positively affected the turbulent kinetic energy at both engine speeds.

Keywords: Diesel fuel blends; ethanol; diethyl ether; NO emission.

1. Introduction

Today, with the increase in oil prices, the decrease in the demand for fossil fuels and the increase in air pollution, studies on the use of alternative fuels in diesel engines have increased as well as in gasoline engines [1,2]. When the studies in the literature are examined, there are many studies in which ethanol and diethyl ether are used as fuel at different rates.

Carvalho et al. investigated the effects of the use of diethyl ether (DEE) on engine performance and emissions in a diesel engine operating on biodiesel-ethanol mixtures. In this study, besides biodiesel and diesel fuel, B80E20 (80% biodiesel, 20% ethanol) and B76E19DEE5 (76% biodiesel, 19% ethanol and 5% diethyl ether) fuels were tested at different engine loads (2.7 kW, 5.4 kW, 8.1 kW). The highest fuel consumption and the highest NO emissions were observed when B80E20 fuel was used for all loads in the engine studied, while the best engine efficiency was determined with B76E19DEE5 fuel. This is due to higher oxygen content and lower cetane number. In all three load conditions, B100 fuel according to D100, CO and NOx increased while THC decreased. [3]. Wang et al. investigated the effects of using biodiesel-ethanol mixtures and diethyl ether (DEE) on engine performance and emissions in a diesel engine. In this study, besides diesel and biodiesel fuels, B95E5 (95% biodiesel, 5% ethanol), B90E10, B95DEE5 (95% biodiesel, 5% diethyl ether) and B90DEE10 fuels were used. Biodiesel fuel was produced from orange oil methyl ester. As a result, the thermal efficiency of the engine increased when the biodiesel-ethanol mixture was used compared to diesel fuel and biodiesel fuels. When using B90DEE10 fuel, CO emissions increased while HC, smoke and NO_x emissions decreased, better engine performance and lower emission values were obtained with this B90DEE10 fuel compared to other biodiesel mixture types [4]. Sivalakshmi and Balusamy investigated the combustion, performance and emission values of BD5 (5% diethyl ether and 95% neem oil biodiesel), BD10 and BD15 fuels using 5%, 10% and 15% diethyl ether as an additive in a diesel engine. When the BD5 mixture was compared with other fuels,

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the highest in-cylinder pressure value and heat release rate was reached in the engine than the neat biodiesel fuel. Also, BD5 fuel compared to other neat biodiesel fuel, it reduced CO emissions at full load, while soot emissions decreased under all load conditions. NO and HC emissions increased at almost all loads. Higher thermal efficiency was obtained in BD5 fuel [5]. Ibrahim used diesel fuel, a mixture of cottonseed biodiesel and diethyl ether (D100, D70B30, D70B25DEE5 and D70B20DEE10) in a diesel engine operating at 1500 rpm. In the engine in which D70B30 mixture was used, it was observed that the specific fuel consumption increased while the thermal efficiency decreased compared to diesel fuel. When the results were examined, there was an increase in the specific fuel consumption and a decrease in the thermal efficiency with the increase in the DEE ratio in the mixture compared to diesel fuel, 8.1% and 6.8%, respectively. DEE did not affect the stable operation of the engine in any negative way. However, diesel fuel compared to other fuels had lower heat release rate and longer burning duration at higher engine loads [6]. Pugazhvadivu and Rajagopan compared their effects on emissions by adding 10%, 15% and 20% diethyl ether to B25, B50, B75, B100 and diesel fuel. Addition of diethyl ether to these mixtures resulted in significant reductions in NOx emissions at low and medium loads. However, NO_x emission was higher compared to diesel, and it lower compared to biodiesel blend at high loads [7]. Verma et al. looked at the effect of ethanol (E)-methanol (M)-diesel-microalgae biodiesel (S) blend on engine performance and emissions. In this study, seven different fuels were tested, namely S0E0M0, S20E0M0, S40E0M0, S20E20M0, S40E20M0, S20E0M20 and S40E0M20. The engine was operated under different load conditions (25, 50, 75 and 100%). As a result, it had been observed that mixtures containing ethanol increased the torque and decrease the exhaust gas temperature. On the other hand, the methanol helped to increase the cylinder pressure. In addition, the ignition delay duration and burning duration were increased with the methanol [8].

Venu and Madhavan investigated the effects on the engine by adding diethyl ether to ethanol-jatropha plant biodiesel-diesel (EBD) and methanol-biodiesel-diesel (MBD) fuels. EBD (20% ethanol, 40% biodiesel and 40% diesel) and MBD (20% methanol, 40% biodiesel and 40% diesel) mixtures were added with 5% and 10% diethyl ether. These fuels were named as EBD-5DEE, EBD-10DEE, MBD-5DEE and MBD-10DEE. As a result, adding DEE to EBD increased the in-cylinder pressure value while reducing the burning duration and NO_x emissions. On the other hand, with the addition of DEE to MBD, the brake specific fuel consumption, cylinder pressure and heat release rate were decreased and in the exhaust emissions, soot, CO and CO_2 emissions increased. In general, blends of EBD-5DEE and MBD-5DEE gave better results than blends of EBD-DEE10 and MBD-DEE10. The lowest NO_x emission and the bsfc were reached for MBD-5DEE fuel [9]. Sugash et al. studied the effects of a blend of cottonseed biodiesel with diethyl ether on performance and emissions. In the study, diesel fuel, B20 (20% biodiesel, 75% diesel fuel, 5% diethyl ether), B40 (40% biodiesel, 55% diesel fuel, 5% diethyl ether) and B60 (60% biodiesel, 35% diesel fuel, 5%) diethyl ether) fuels were tested at different engine loads (25%, 50%, 75%, 100%). When the results were examined, it was determined that the thermal efficiency of B20 fuel was close to the diesel fuel. In the results obtained with B60 fuel, the lowest fuel consumption and NO emission values were obtained, and the maximum CO and UHC emissions was reached. Addition of diethyl ether the specific fuel consumption of fuel was reduced [10].

Nishanth et al. researched the effects of Jatropha Curcas biodiesel-diethyl ether mixture on a variable compression ratio diesel engine. Diesel, B1 (20% biodiesel, 75% diesel fuel, 5% diethyl ether), B2 (25% biodiesel, 70% diesel fuel, 5% diethyl ether) and B3 (30% biodiesel, 65% diesel fuel, 5% diethyl) ether) fuels were used. In the study, it was emphasized that B2 fuel increased thermal efficiency and mechanical efficiency, 6.32% and 3.15%, respectively. Also, HC and NO_x emissions reduced with this fuel type [11]. Ayhan and Tunca used as fuel diesel fuel and mixtures containing 3%, 5% and 7% DEE in a single-cylinder direct injection diesel engine. A decrease in torque and effective power was observed as the DEE ratio increased at full load and 1000, 1300, 1600 and 2000 rpm speed conditions. The specific fuel consumption and effective efficiency were improved in the engine. Moreover, it was determined that there were significant reductions in NO_x and soot emissions [12]. Loganathan et al. used hydrogen-enriched cashew nut shell biodiesel by adding diethyl ether in a single-cylinder diesel engine. As a result, when hydrogen was added to the mixture, CO and HC emissions decreased, and it was determined that they were further reduced by adding diethyl ether to these mixtures, 43% and 50% decrease respectively. Also, in-cylinder pressure, NO_x emission and heat release rate were increased with the addition of DEE [13]. The high cetane number, oxygen content, low calorific value, the high latent heat of vaporization parameters of DEE affect NO_x formation. High cetane number reduces ignition delay and also reduces NOx emissions by shortening combustion duration [14]. The high burning velocity of DEE makes an extra contribution to the shortening of the burning time. Improves engine performance and exhaust emissions thanks to shorter combustion durations [15]. The addition of ethanol and DEE to diesel fuel decreased the in-cylinder chamber temperature with both increased heat of vaporization and low flame temperature. As a result, the combustion temperature has decreased and therefore NO_x emissions have decreased [16]. Banapurmath et al. [17] tested different volume ratios of ethanol and DEE in a single-cylinder direct injection diesel engine. Effect of ethanol and diethyl ether-diesel blends on diesel engine performance, combustion and emissions was investigated using four different blends of ethanol (E0 -neat diesel, E5, E10, E15 and E20)and diethyl ether (DEE0 - neat diesel, DEE5,DEE10, DEE15 and DEE20). As a result, as the ratio of ethanol and DEE increased, NO_x decreased at low loads, while HC increased because of high fuel consumption and high latent heat of vaporization. On the other hand, CO emissions decreased at high loads, and brake thermal efficiency increased.

Usta et al. tested an ethanol-diesel mixture containing 15% ethanol and two different biodiesel blended fuels in a four-cylinder turbo diesel engine with a pre-combustion chamber. While ethanol in the mixture decreased CO emissions, it increased NO_x emissions. While ethanol caused a decrease in engine power, the addition of biodiesel did not affect power compared to diesel fuel [18].

Today, many software are used that allow flow, heat and emission modeling in internal combustion engines [19-23]. These software provide significant advantages in terms of both time and cost reduction. Xu et al. experimentally and numerically investigated the effects of diesel fuel and acetone-butanol-ethanol (ABE) mixtures on combustion and soot formation in a diesel engine. The numerical study was carried out with CFD KIVA-3V software, combined with the CANTERA code. In this study, diesel fuel was used as the reference fuel. In addition, the ABE were mixed with a magnetic stirrer in the ratio of 3:6:1 (acetone: butanol: ethanol) among themselves by volume. The ABE mixtures were tested after blending with diesel fuel at 20% and 50% by volume. As a result, in-cylinder pressure, heat release rate, combustion efficiency and soot emissions decreased with the increase of the ABE ratio in the mixture. Moreover, the combustion temperature distribution was more uniform, and the flame lift-off length increased [24].

When the literature is examined, it is seen that studies with ethanol diesel mixture are frequently encountered. However, numerical modeling of diethyl ether additives with ethanol-diesel mixtures and comprehensive combustion analysis studies are very limited. The aim of the study was numerically researched the effects of the addition of ethanol and diethyl ether to diesel fuel on the combustion characteristics, velocity distribution in the chamber and NO_x emissions. Depending on the change of fuel properties, the effect of different combustion fractions formed in the engine on NO emissions had been extensively studied in a DI diesel engine.

2. Numerical Study

In this study, 100% diesel fuel (D100) and by volume 90% diesel+10% ethanol blend (D90E10), 80% diesel+20% ethanol blend (D80E20), 80% diesel+10% ethanol+10% diethyl ether blend (D80E10DEE10) and 85% diesel+ 10% ethanol+5% diethyl ether mixture (D85E10DEE5) was used as fuel. These five different mixed fuels were tested at 2000 and 3000 rpm speed conditions. The physical and chemical properties of five different fuel types were given in Table 1.

Table 1. Properties of diesel fuel, ethanol and diethyl ether fuels [25,26]

Properties	Diesel fuel	Ethanol	Diethyl Ether
Chemical formula	C_xH_y	C_2H_6O	$C_4H_{10}O$
Oxygen content (% mass)	-	34,7	21
Density (g/cm³)	~0,83	0,789	0,713
Viscosity (mm²/s)	2,6-4,1	1,19	0,23
Boiling point (°C)	180-360	78,4	34,6
Sulfur content (ppm)	~250	—	—
Ignition temperature (°C)	315	235	160
Lower calorific value (kJ/kg)	42500	26800	33900
Latent heat of evaporation (kJ/kg)	250	825	356
Cetan number	40-55	5-8	~128
Theoric air/fuel ratio	14,6	9	11,1
Molecular weight	190-220	46,04	74,12
Carbon content (% mass)	87	52,2	64,9
Hydrogen content	13	13	13,5

ANTOR 3 LD 510 direct injection diesel engine was used within the scope of numerical study. The technical data of the engine were given in Table 2. The modeling was done by defining the initial boundaries in the ESE DIESEL section of the AVL FIRE software. Five different **mixed** fuels were defined from the library of the AVL FIRE software. In the modeling, WAVE model was used as spray model, k-zeta-f model was used as turbulence model and ECFM-3Z model was used as combustion model. The k-zeta -f turbulence model has been widely adopted for computational networks and flow conditions at any dimensionless distance close to the wall. The ECFM-3Z model can also be used for the combustion model with a combination of injection and EGR. ECFM-3Z is widely used in combustion analysis of direct injection engines [27-29]. Wave breakup model was used in the software. This model performs spray modeling depending on the physical and chemical properties of the used fuel. This model is used in injection simulation for diesel fuel and similar fuel types [30].

The intake temperature and intake pressure values at the initial conditions were defined as 293 K and 1 bar, respectively. The initial density of the ambient gas was calculated according to the ideal gas law. The velocities of the walls in all directions were defined as 0 m/s for the boundary conditions. In addition, the wall temperatures were taken as 450 K. In the boundary conditions, the type of fuel was defined as the moving boundary condition by keeping the cylinder walls, cylinder head values, piston and all other variables constant. Thus, only the effect of fuel on combustion and NO emissions was examined.

Zeldovich mechanism was used for NO in the analysis. The thermal NO reaction mechanism was first described by Zeldovich as a two-stage reaction mechanism. Then, this mechanism was improved by adding the effects of OH radicals on thermal NO. The mechanism can be expressed as follows [31].

$$O + N_2 \leftrightarrow N + NO, k_{f1} = 1.8x10^8 e^{-38370/T}$$
$$k_{b1} = 3.8x10^7 e^{-425/T} \quad (1)$$
$$N + O_2 \leftrightarrow O + NO, k_{f2} = 1.8x10^4 Te^{-4680/T}$$
$$k_{b2} = 3.81x10^3 Te^{-20820/T} \quad (2)$$
$$N + OH \leftrightarrow H + NO, k_{f3} = 7.1x10^7 e^{-450/T}$$
$$k_{b3} = 1.7x10^8 e^{-24560/T} \quad (3)$$

Here k_{f1}, k_{f2} and k_{f3} are the forward reaction constants and k_{b1}, k_{b2} and k_{b3} are the backward reaction constants. The eq. (1) represents the thermal NO formation rate due to the high activation energy, which indicates the high temperature dependence of NO formation.

The wall interaction model known as "Walljet10" was used for all test fuels in the modelling. In all numerical studies, the spray angle was determined as 126°. In Figure 1, the combustion chamber geometry mesh structure of the test engine was given. Analyzes were performed on approximately 100000 cells.

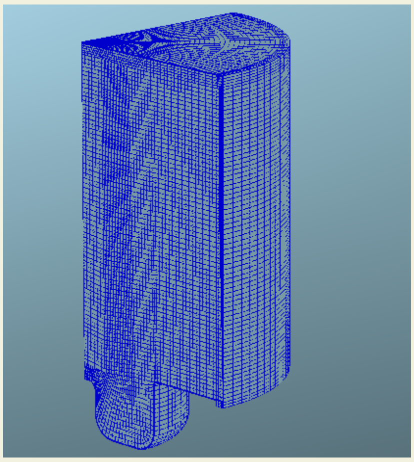

Figure 1. Mesh structure of chamber geometry

3. Results and Discussion

The effects of different fuel mixtures on combustion and emissions were investigated in the AVL-FIRE program at 2000 and 3000 rpm speed conditions. In the study, D100, D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels were selected from the AVL library and defined to the software in different mixing ratios. The variation of in-cylinder pressures depending on the crank angle of the engine analyzed using different fuel mixtures was given in Figure 2. It was observed that the maximum in-cylinder pressures for all test fuels were obtained after the 725° CA. There are many parameters that affect the in-cylinder pressure distribution. Some of these can be listed as fuel density, cetane number, evaporation ability, ignition temperature and calorific value. When the in-cylinder pressure distributions were examined, it was seen that the maximum pressures were lower in all blended fuels compared to D100 fuel at both engine speeds. As seen in Table 1, this was because ethanol and diethyl ether have lower heating values than diesel fuel. For the 2000 rpm operating conditions of the engine, the maximum in-cylinder pressure values for D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels compared to the D100 fuel decreased by 0.5%, 1.3%, 3.5% and 2%, respectively. For the engine's 3000 rpm operating conditions, the decreased in maximum pressures of D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels compared to Diesel fuel were 0.5%, 1%, 3% and 1.7%, respectively.

Table 2. Technical properties of the test engine

Engine Name	Antor 3 LD 510
Engine Type	Four stroke, air-cooled, single-cylinder and direct injection diesel engine
Piston displacement	510 cm^3
Stroke x Bore	90 x 85 (mm x mm)
Compression ratio	17.5:1
Power	6.6@3000 (kW)
Torque	32.8@2000 (Nm)
Injection angle	126°
Injector hole number	4

Ethanol has a lower cetane number than diesel fuel. Especially low cetane number causes diesel knock in engines as well as worsening combustion. Due to this feature of ethanol, there are many studies where it is mixed with diesel fuel at low rates such as 10%, 20% and 30% [23,32]. Therefore, the selected ethanol-blended fuels did not cause significant pressure fluctuations in the diesel engine. Also, pressure distributions obtained using D100 fuel and ethanol blended fuels gave close results.

When the cumulative heat release curves are examined in Figure 3, it is seen that they are parallel with the in-cylinder pressure distributions. It could be said that the high calorific value of D100 fuel compared to other mixtures is effective especially in the maximum pressure and heat releases in the cylinder. It was observed that the heat release rates decreased with the increase in the ethanol ratio in the mixture. It could be said that this situation causes a decrease in in-cylinder temperatures.

As is known, diethyl ether has a low calorific value compared to D100 fuel, and diethyl ether has a high calorific value compared to ethanol fuel. At the same time, the low cetane number of ethanol fuel causes some disadvantages to the engine compared to D100 fuel [25]. In order to eliminate these problems, it was aimed to increase the cetane number of the mixture by adding 5% and 10% diethyl ether fuel to the ethanol fuel. According to the results of the analysis made in this direction, it was seen that the mixtures of D85E10DEE5 and D80E10DEE10 reduced the maximum heat release rates compared to D80E20, D90E10 and D100 fuels. It could be said that the cetane number affects this situation. The increase in the cetane number in the mixture increased the ignition ability of the fuel, shortened the ignition delay times and thus increased the maximum pressure and heat releases in the cylinder. On the other hand, the high heat of vaporization of DEE fuel could be demonstrated.

The turbulent kinetic energy/crank angle change of the engine using different fuels is seen at 2000 and 3000 rpm engine speeds (Figure 4). For all fuels, it was seen that the TKE value increases around TDC and decreases with the expansion stroke. When the results were examined, the highest TKE value was obtained for both engine speeds in the D80E20 mixed operation, and the lowest in the D80E10DEE10 mixture operation. When looking at D100 and other blended fuels in general, results were close to each other. The turbulent kinetic energy changes showed a 0.6% decrease in D90E10 fuel, a 2.7% decrease in D80E10DEE10 fuel, a 1% decrease in D85E10DEE5 fuel, and an increase of 0.05% in D80E20 fuel compared to the D100 fuel. Figure 4 shows the variation of turbulent kinetic energy (TKE) according to the crank angle obtained from different fuels at 3000 rpm operating conditions. Increasing engine speed and air movement caused the TKE value of the mixture to increase. In particular, the fuel injected into the cylinder with the movement of the piston towards the TDC caused the TKE value to increase in all test fuels. Decreased pressure and flow mobility with the development of combustion event and the movement of the piston towards the BDC caused the TKE value to decrease in all test fuels. At 3000 rpm engine speed, the highest TKE value was obtained in D90E10 mixed operation, and the lowest TKE value was obtained in D80E10DEE10 mixed operation.

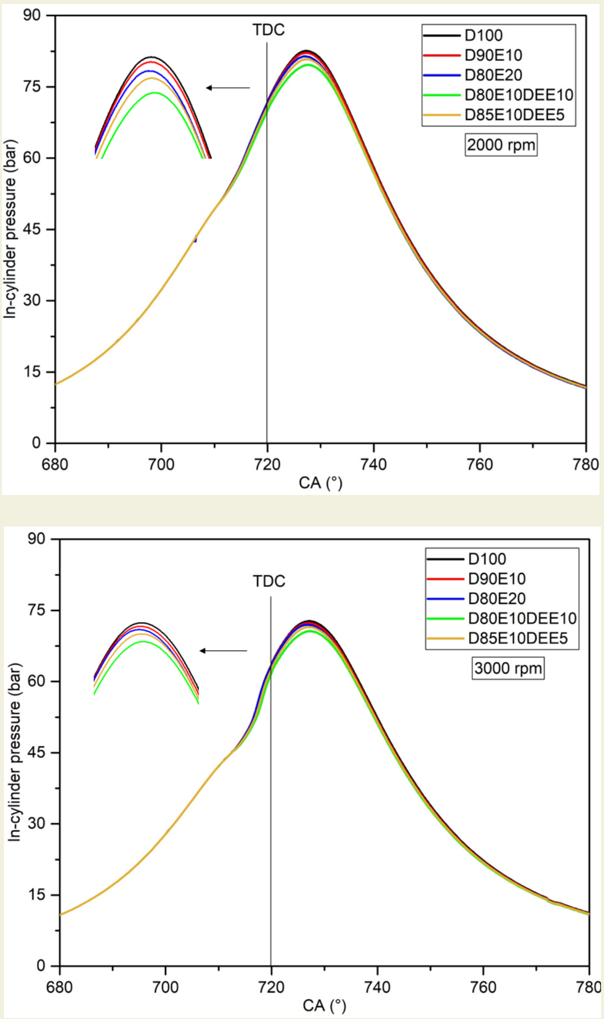

Figure 2. Variation of in-cylinder pressure according to crankshaft angle in different mixtures ratios at 2000 and 3000 rpm.

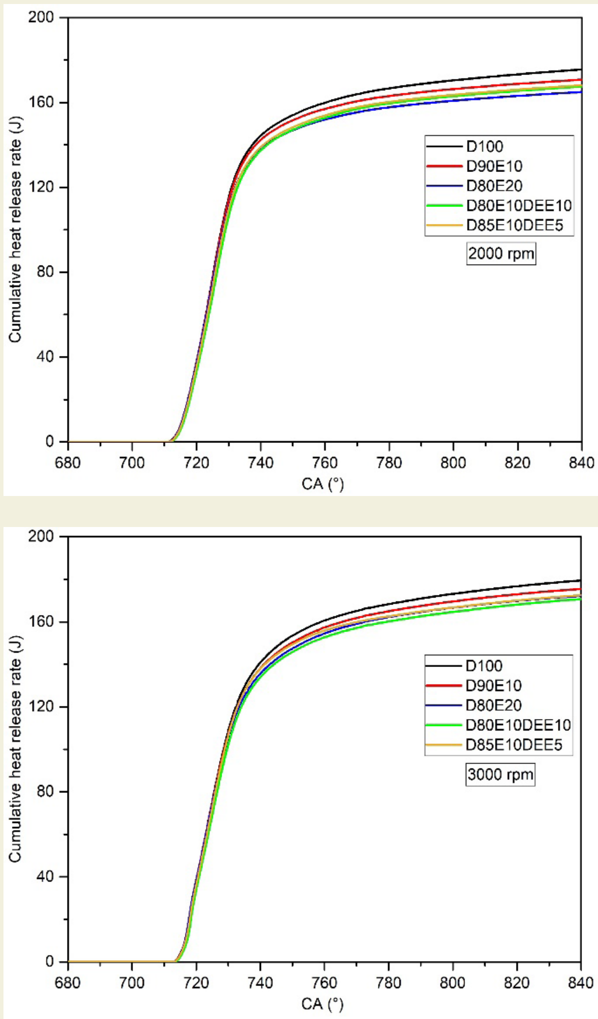

Figure 3. Variation of cumulative heat release rates according to crankshaft angle in different mixtures ratios at 2000 and 3000 rpm.

The variation of NO emissions according to the crank angle for different fuels is given in Figure 5. In parallel with the increase in in-cylinder temperature, NO emissions increased. There are many parameters that affect NO emissions in engines. Some of these are combustion temperature, excess air coefficient and burning duration. Especially the use of different fuels in engines causes these parameters to change and NO emissions to differ. All ethanol and diethyl ether additive test fuels reduced NO emissions compared to diesel fuel. This was thought to be due to the calorific value of the fuel. When the in-cylinder pressure and heat release rates were examined, the lowest value was obtained in the D80E10DEE10 fueled operation. It had been observed that NO emission was in parallel with the emission values. Although the DEE additive had a higher heating value compared to the ethanol additive, the maximum heat release rate and NO values were lower [14, 16]. This had been accepted as an indication of the complex process of combustion in the engine. It could be said that especially the high cetane number of DEE additive shortens the ignition delay time and caused a decrease in maximum pressure and temperatures. Therefore, NO emission decreased depending on the temperature. It would be expected that NO emissions would increase due to the oxygen they contain in the structure of diethyl ether and ethanol. However, the combustion end temperatures of ethanol and diethyl ether fuels decreased due to their low calorific value and high latent heat of evaporation compared to diesel fuel. NO emissions decreased D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuel compared to D100 fuel by 2%, 10%, 15% and 9% respectively. The variation of NO emission of test fuels at 3000 rpm speed condition is seen (Figure 5). Increased engine speed, causes an increase in the amount of injected fuel into the combustion chamber per unit time and an increase in in-cylinder temperatures. NO emission increased with increasing speed. In both speed conditions, lower NO emission was obtained in the blended fuels. When Figure 5 is examined, the lowest NO value was obtained for 3000 rpm in the D80E20 blended fuel study. In this case, the combustion end temperatures decreased due to the low calorific value and high latent heat of evaporation of the blended fuels compared to the D100 fuel. Therefore, NO emission decreased. For 3000 rpm engine speed, the reductions in NO emissions of D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels com-

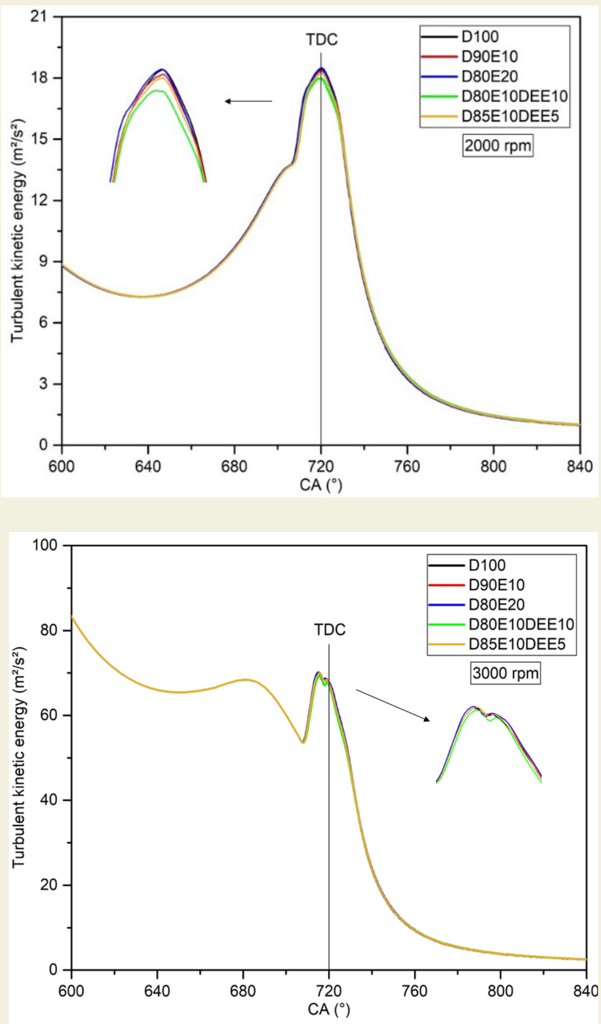

Figure 4. Variation of turbulent kinetic energy according to crankshaft angle in different mixtures ratios at 2000 and 3000 rpm.

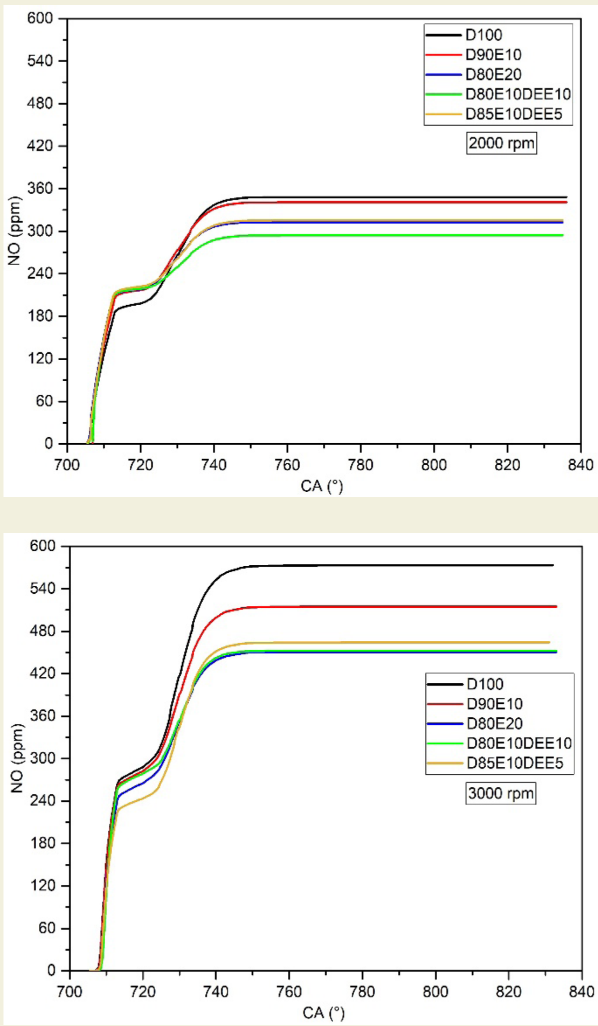

Figure 5. Variation of NO emissions according to crankshaft angle in different mixtures ratios at 2000 and 3000 rpm.

pared to diesel fuel were 10%, 21%, 20% and 16%, respectively.

Figure 6 shows the velocity distributions for five different fuels at different crank angles. In the simulations, the injection process for all test fuels took place in the crank angle range of 705-729°. Especially at the crank angle of 730°, it was seen that the spraying is completely stopped and the radial flow velocity is reduced for all test results. It was showed that the flow rate was maximum compared to other angles at the 720° crank angle. The increase in speed in the engine causes squish movements and swirl

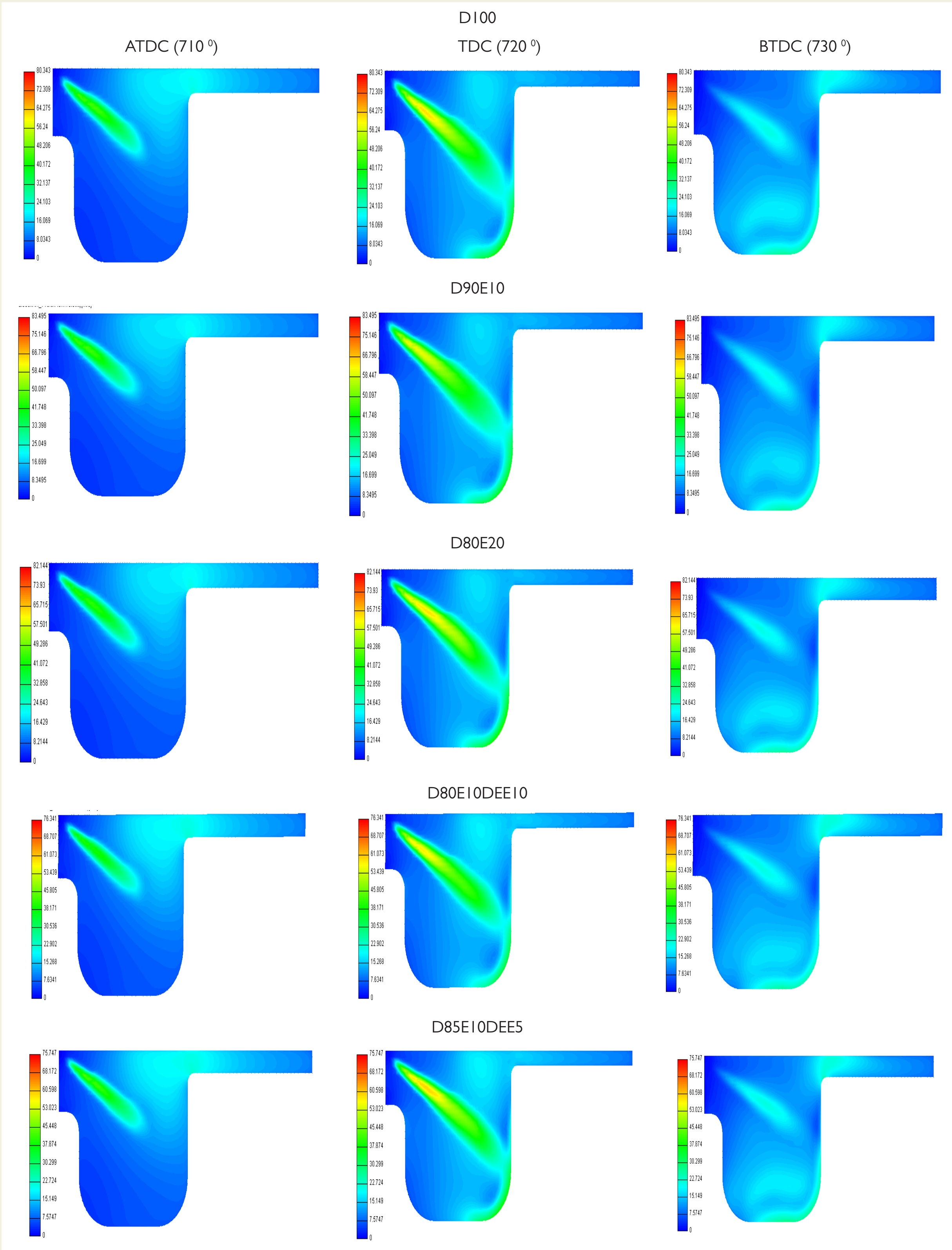

Figure 6. Speed distributions for five different fuels at 710°, 720° and 730° crank angles

rates to increase. Depending on both engine speed and piston movement, the velocity distribution of the fresh charge increased.

The velocity gradient of the fuel droplets sprayed into the combustion chamber varies depending on fuel viscosity, surface tension, droplet size and number, degree of evaporation and other factors (temperature, pressure and air flow mobility, etc.) in the combustion chamber.

At 3000 rpm speed condition, a slight increase in flow rate was obtained depending on the ethanol ratio added to the D100 fuel. On the other hand, the flow rate decreased with the addition of DEE to the blended fuels. This decrease is more clearly seen especially in the D80E10DEE10 mixture containing 10% DEE (Figure 6). As it is known, the density, surface tension and viscosity of fuel have important effects on the spray dynamics. Here, it was considered that the liquid + vapor interaction affects the fuel velocity distributions for all test fuels. Although ethanol has lower density and viscosity compared to diesel fuel, it can be said that diesel fuel, which shows early evaporation ability, breaks down more easily. This was thought to reduce the flow rate and inertia of diesel fuel. This approach supports velocity distribution graphs created by adding DEE to the mixture. In addition, when the flow velocity distributions of DEE added fuels were examined, a decrease was observed in the flow rate of DEE added fuels compared to diesel and diesel/ethanol mixtures.

4. Conclusions

In this study, in-cylinder pressure and heat release rate, TKE and NO analysis of four different mixed fuels (D90E10, D80E20, D80E10DEE10 and D85E10DEE5) and D100 fuel were performed. Depending on the change of fuel properties, the effects of different combustion fractions formed in the engine on NO emissions, which were extremely important for diesel engines, had been comprehensively discussed. In addition, velocity distributions of five different fuels in the combustion chamber were evaluated. Analyzes were made in the ESE DIESEL part of the AVL FIRE software. Engine speeds of 2000 and 3000 rpm were chosen as the operating condition. The obtained findings as a result of the study can be listed as follows.

- The maximum in-cylinder pressure of D100 fuel was higher than other blended fuels. The maximum in-cylinder pressure values for D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels compared to the D100 fuel decreased by 0.5%, 1.3%, 3.5% and 2%, respectively, and for 3000 rpm were obtained 0.5%, 1%, 3% and 1.7%, respectively. This is because ethanol and diethyl ether have lower heating values than diesel fuel.
- The cumulative heat release rates obtained at 2000 rpm for D100, D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuels were 176.63 J, 171.65 J, 165.76 J, 168.46 J and 169.02 J, respectively. At 3000 rpm, these values were 180.55 J, 176.4 J, 173.05 J, 171.75 J and 173.51 J respectively. It was concluded that this situation was caused by the calorific value of diesel, diethyl ether and ethanol.
- NO emissions decreased D90E10, D80E20, D80E10DEE10 and D85E10DEE5 fuel compared to D100 fuel by 2%, 10%, 15% and 9%, respectively. At 3000 rpm, these values were 10%, 21%, 20% and 16%, respectively. The reason for this decrease was the lower calorific value and the high latent heat of vaporization of the mixtures compared to diesel fuel.
- The highest TKE value was obtained for 2000 rpm in D80E20 (0.05% increased) mixed fuel, and the lowest TKE value was obtained in D80E10DEE10 (2.7% decreased) mixed operation. At 3000 rpm, the values were close to each other, the highest TKE value was obtained in D90E10, and the lowest was obtained in D80E10DEE10. Ethanol had a positive effect on turbulent kinetic energy.
- The flow rate of ethanol was higher than diesel fuel. Diethyl ether was found to decrease the flow rate. This situation was due to the early evaporation ability of the fuel.

As a future study proposal on this subject, combustion chamber temperature, mass fraction burned, equivalence ratios and temperature distribution of the fuel injected into the combustion chamber at different crankshaft angles, engine performance curve and other exhaust emissions such as soot, CO_2, CO, HC can be analyzed.

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