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Modification of piston bowl geometry and injection strategy, and investigation of EGR composition for a DME-burning direct injection engine

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ABSTRACT

The amount of pollutant gases in the atmosphere has reached a critical state due to an increase in industrial development and the rapid growth of automobile industries that use fossil fuels. The combustion of fossil fuels produces harmful gases such as carbon dioxide, nitrogen monoxide (NO), soot, particulate matter (PM), etc. The use of Dimethyl Ether (DME) biofuel in diesel engines or other combustion processes have been highly regarded by researchers. Studies show that the use of pure DME in automotive engines will be possible in the near future. The present work evaluated the environmental and performance effects of changing the injection strategy (time and temperature), piston bowl geometry, and exhaust gas recirculation (EGR) composition for a DME-burning engine. The modification of piston bowl parameters and engine simulation were numerically performed by using AVL fire CFD code. For model validation, the calculated mean pressure and rate of heat released (RHR) were compared to the experimental data and the results showed a good agreement (under a 70% load and 1200-rpm engine speed). It was found that retarding injection timing (reduction in in-cylinder temperature, consequently) caused a reduction in NO emissions and increased soot formation, reciprocally; this occurred because of a reduction in temperature and a lower soot oxidation in the combustion chamber. It became clear that 3 deg before top dead center (BTDC) was the appropriate injection timing for the DME-burning heavy duty diesel engine running under 1200 rpm. Also, the parametrical modification of the piston bowl geometry and the simultaneous decrease of Tm (piston bowl depth) and R3 (bowl inner radius) lengths were associated with lower exhaust NO emissions. For the perfect utilization of DME fuel in an HD diesel engine, the suggested proper lengths of Tm and R3 were 0.008 and 0.0079 m, respectively. Furthermore, various EGR compositions for the reduction of exhaust NO were investigated. The simulation results showed that a 0.2 EGR composition led to a reduction in the exhaust NO by 37%.

1. Introduction

The rising concentration of pollutants in the environment have adverse effects on human health. The increase in greenhouse gases poses a new crisis for mankind, which is known as rising global temperature. Researchers have proposed biological fuels as clean and environmentally friendly fuels. Therefore, the use of biofuels in diesel engines in pure or blended form have been widely investigated. The use of clean fuels (and changing combustion chamber parameters) to reduce greenhouse gases in automobiles, rockets and power generation have

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been accompanied with positive effects [1-6]. There are several ways to enhance biofuels production [7-11]. Biofuels are very diverse. Investigations [1] show that taking advantage of bio-butanol fuel blended with conventional diesel leads to lower NO and soot emissions. In a dual fuel engine, speed has significant effects on emissions and an optimal injection timing for this mixed fuel has been presented [1,2,4]. The increasing use of hydrogen as a supplement gaseous fuel for diesel through the intake valve affects exhaust emissions [3,12,13,14]. Pollutant gases from diesel engines have a relationship with the piston bowl geometry; the improvement of the bowl size has a beneficial impact in terms of exhaust emissions and engine performance. Therefore, the optimization and modification of the piston bowl geometry, especially piston bowl depth, is effective for the reduction of exhaust emissions [5]. A change in the injection timing or pattern of fuel in the combustion chamber may reduce exhaust emissions [15-18]. *Hydrogen*-diesel (*blended* fuel) has been experimentally examined in order to determine the PM emission of a diesel engine under different loads and speeds; the results showed that load and speed are the most effective parameters (in terms of environmental effects) [19]. In order to simulate the diesel engine and construct models, various CFD software can be utilized, e.g., AVL FIRE and KIVA. For validation of AVL Fire software, a single-cylinder engine has been simulated and compared to the experimental data; the results gave an acceptable agreement [20]. EGR is a widely adopted technique that decreases exhaust NO emissions. This includes recirculating a controllable ratio of the engine's tailpipe back to the in-cylinder through the intake valve. A valve is utilized to control the flow rate, and the EGR pipe can be closed. This technique has remarkable effects on NO emissions. For this reason, studies on various engines examined the influence of using EGR rates and the appropriate EGR rates for diesel engines have been proposed [21-25]. In addition, the increase in induction of swirl in various small piston bowls led to a reduction in some tailpipe emissions with increasing air-fuel mixing [26]. Investigations show that using a biodiesel fuel or DME in compression ignition engines can increase the exergy efficiency [27-28]. The present work studied the environmental effects of using DME biofuel in the combustion chamber of a compression ignition engine. Initially, for pure diesel fuel, validation of numerical simulation was carried out under a 70% load and a 1200-rpm engine speed. Subsequently, the obtained results of this simulation were developed for DME biofuel. The present research investigated injection timing, injection temperature, piston bowl depth, and EGR composition for a DME-burning engine. Firstly, four injection timings (for pure DME) before top dead center (BTDC) were examined to choose the best injection timing at a constant speed of 1200 rpm. It is acknowledged that dimension modification of the piston bowl may facilitate reaching the optimal environmental condition. For this reason, piston bowl depth and R3 length have been further investigated. Tm is the piston bowl depth and its length is very effective. R3 is the inner radius parameter of the piston bowl. Increasing the length of this parameter causes enhancement of the piston bowl volume. In studies on a ISM 370 diesel engine, the EGR System has not been examined [3,12]. The influence of EGR system on NO emission were investigated with different compositions (0, 0.11, 0.15, 0.2, and 0.8). It was predicted that the EGR system would reduce exhaust NO. However, exhaust gases such as soot, CO_2, etc. were also affected by exhaust gas recirculation. The computational domain was commenced at IVC (570 deg crank angle(CA)).

2. Simulation

An ISM 370 diesel engine was selected to study the effects of DME biofuel as an alternative to diesel fuel. The utilized engine characteristics are listed in Table 1. Various parameters were selected for conducting the simulation including turbulence model and mechanism of pollutant gases formation [29-30]. K-zeta-f as the preferable model for turbulence and turbulent wall heat transfer modeling was employed [31]. This model can be utilized for including grids with unstable boundaries and compressed flows. The coherent flame (ECFM-3Z) model distinguishes between three major regimes relevant in the diesel combustion process: auto ignition, premixed flare, and non-premixed [32]. Furthermore, the extended Zeldovich (NO) and Lund flamelet models (soot) for emissions have been considered. The Wave Breakup (which is utilized for atomization modeling of producing droplets) [33] and Dukowicz evaporation (for purification of the droplets) [34] submodels were employed for modeling spray process.

Table 1. Specifications of the direct injection diesel engine

Engine model	Cummins ISM 370
Displacement	10.8 L
Bore	0.125 m
Stroke	0.147m
Engine layout	Inline
Number of cylinders	6
Number of injection holes	4

To ensure a lack of relationship between the simulation results and the mesh number, a sensitivity analysis was performed (below 1 percent error). The results showed that a change in the number of meshes above 70000 did not impact the results and the curves were similar, meticulously. The results obtained from mesh analysis can be observed in Figure 2.

3. Results and discussion

3.1. Validation

Validation of the modeled engine was performed under a 70% load and 1200-rpm engine speed. The initial scheme was designed as in Figure 1. For creating and stabilizing this condition (70% load and 1200 rpm), a computational domain rate was calculated at about 35mg [12]. The HRR and mean pressure graphs obtained by numerical simulation was compared to the experimental data [12]. Figure 2 shows that the predicted pressure traces, as a function of crank angle, had a good agreement with the experimental results and the total error (with taking 21 data from 680 to 780 crank angle) was calculated to be less than 3%. 5 an error below 5% for the pressure curve is acceptable [5]. As can be seen in Figure 2, computational errors from 730 to 740 for the crank angles are larger than other angles. The errors were created because of some indeterminate input parameters. The estimation of input parameters such as inlet pressure, injection timing, etc. led to higher computational errors.

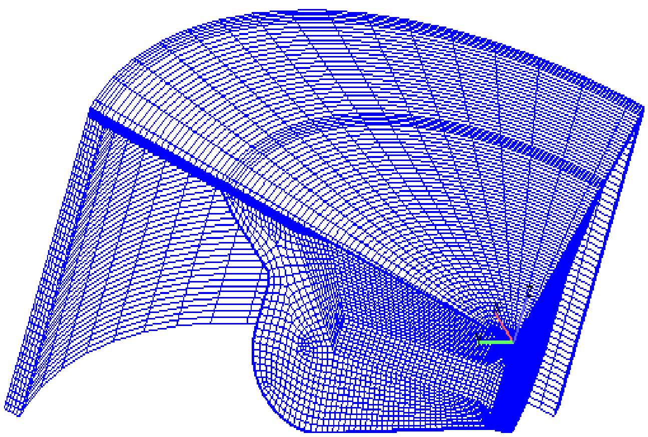

Fig. 1. Baseline Piston bowl geometry at TDC

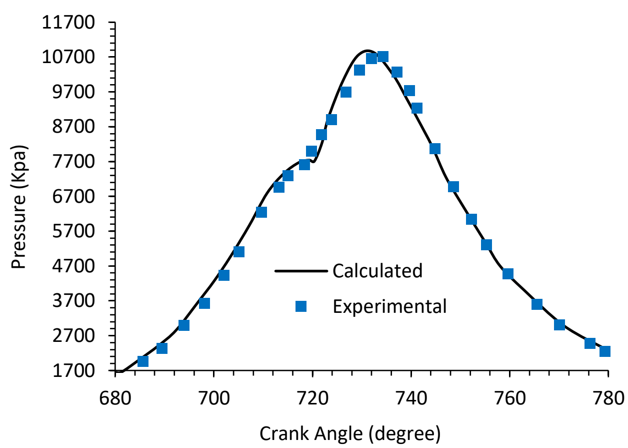

Fig. 2. Comparison of the in-cylinder pressure between experimental and simulation results (start of injection (SOI)= 1.5 deg BTDC)

Figure 3 shows the heat release rate distribution. In this case, simulation errors were higher than experimental results. Without considering heat transfer, radiation energy, latent heat of vaporization, etc., computational errors were inevitable. In similar works, this subject has been confirmed [2,14]. The air motion in the combustion chamber before the injection of the fuel was very significant to meet an appropriate air-fuel mixture. This case justifies the fact that combustion chamber air motion has a fundamental role on the complete combustion in the combustion chamber. The velocity and TKE distribution were calculated and are presented in Figure 4 (at 650 °CA and 718 CA). As illustrated, there are strong airflows in the in-cylinder near the underside of the bowl and cylinder walls. These jet flows (which have clockwise and counter clockwise direction) led to more air-fuel mixing after fuel injection and flow turbulences increased in the compression cycle.

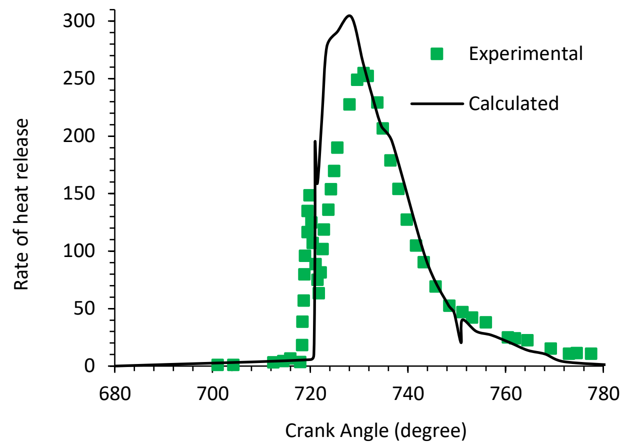

Fig. 3. Comparison of the in-cylinder HRR between experimental and simulation results (SOI=1.5 deg BTDC)

3.2. Environmental effects of changing fuel injection timing for DME biofuel

A detailed analysis of injection timing and intermittent fueling of the engine was a significant step toward decreasing the toxic fumes and greenhouse gases. To investigate the performance enhancement of HD diesel engine fueled with pure DME, the effects of changing fuel injection timing were numerically analyzed. As can be seen in Figure. 5, maximum peak pressure occurred while injection timing was adjusted to 5 deg BTDC. It should be noted that rising peak pressure in the combustion chamber may lead to side effects on engine construction. Furthermore, pressure drop was not appropriate for the combustion process [5]. The accumulated heat release as a function of crank angles (from 680 to 780 CAs) were calculated and are presented in Figure. 5. As indicated, with a delay in fuel injection timing into the combustion chamber, the amount of evaporated fuel did not change much and peak pressure drop did not prevent combustion for the considered injection timings.

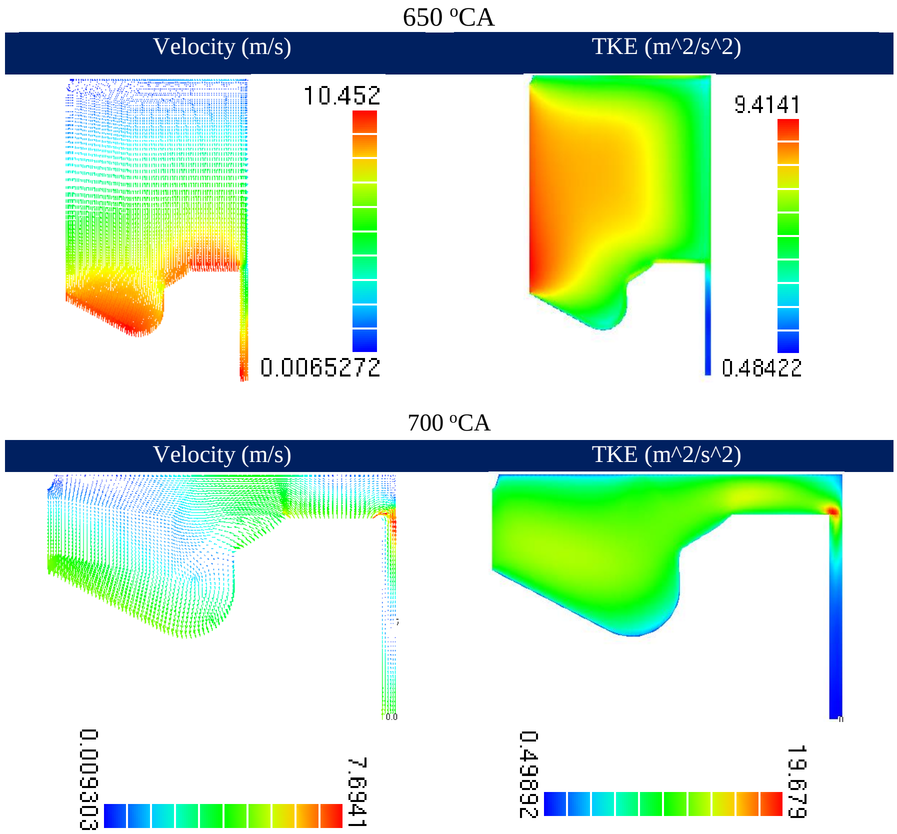

Fig. 4. Velocity and turbulence kinetic energy (TKE) distribution before fuel injection

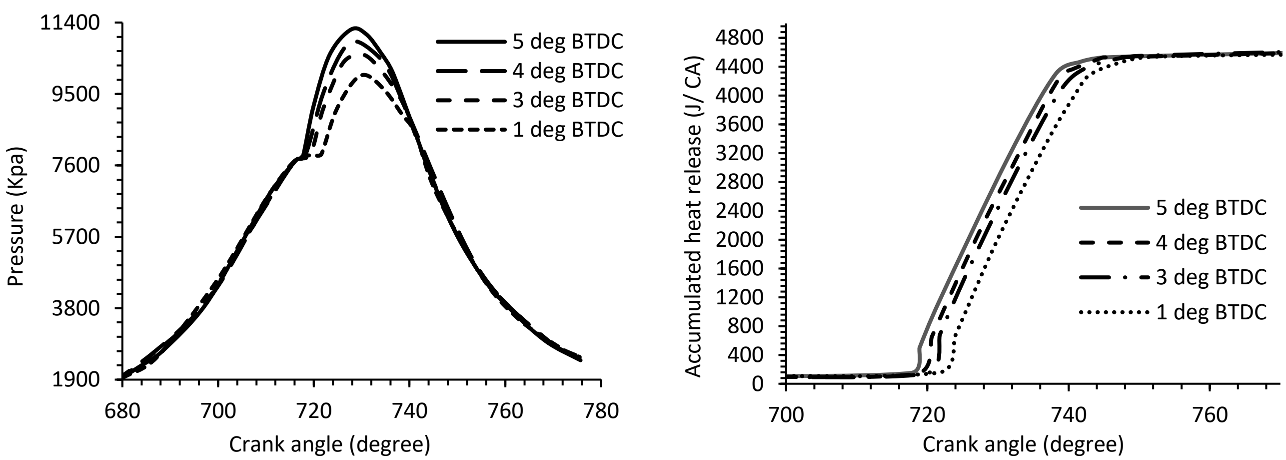

Fig. 5. Mean pressure and accumulated heat realease in various injection timings

As can be seen in Figures 6 and 7, four injection timings were considered (715, 716, 717 and 719 ºCAs) and for these injection timings, the mean mass fraction of NO and soot were subsequently calculated. According to Figure 6, changing injection timing caused a change in the NO emissions. Figure 7 displays an engine equipped with a DME fueling system; a delay in injection timing before TDC (TDC is 720 crank angle) led to an increase in soot formation. The maximum rate of soot formation occurred in 1 deg BTDC (this emission on the contrary produces NO). Thus, the balance between soot and NO productions should be taken into consideration when selecting the injection timing. Nevertheless, 3 deg BTDC was suggested for a DME-burning HD diesel engine (with consideration of a balance between produced NO and soot).

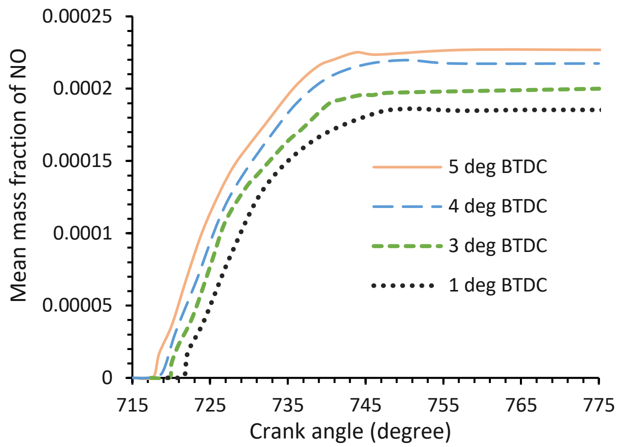

Fig. 6. Mean mass fraction of NO as a function of crank angle in various injection timing

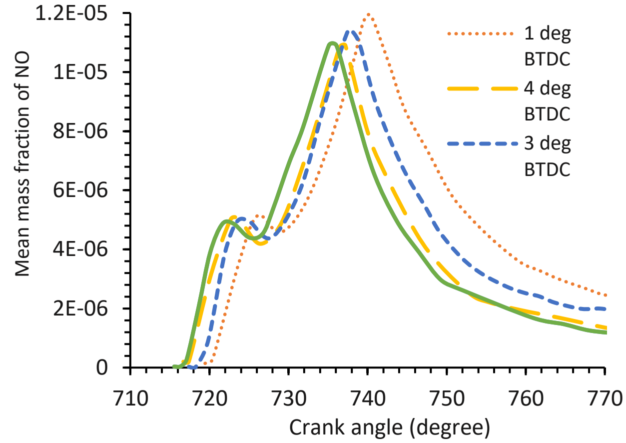

Fig. 7. Mean mass fraction of soot as a function of crank angle in various injection timing

Figure 8 indicates the amount of NO emission and the temperature profile in the combustion chamber for 740 and 724 crank angles. At 724 ^{o}CA, the temperature profile in the piston bowl wall (the red color was the highest temperature at 2400 K and the blue color was the minimum in-cylinder temperature at about 1064 K) was more than elsewhere in the combustion chamber. In the 724 crank angle, only a small amount of NO (about 0%) was produced (the highest mass fraction of NO is indicated in Figure 6 and the minimum mass fraction of NO caculated to around 2.9e-30).

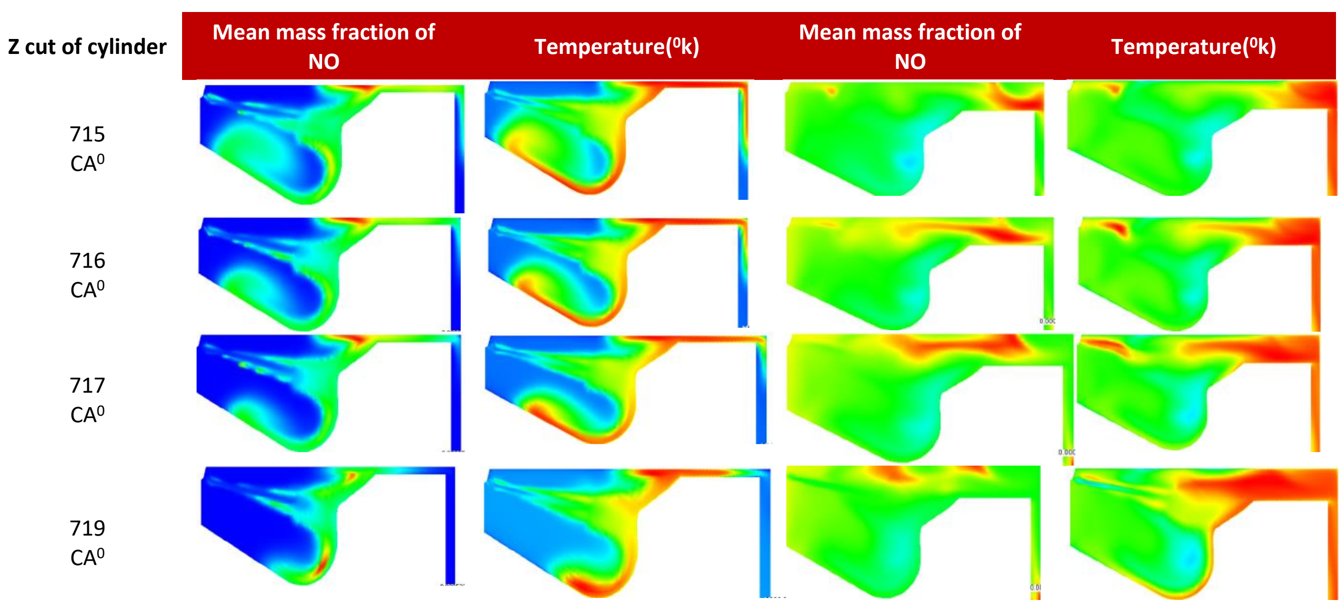

Fig. 8. Exhaust NO (for DME fuel) and in-cylinder temperature distribution at two crank angles (a = 724, b = 740) and under four injection timings (715, 716, 717 and 719 CA)

3.3. Efficacy of fuel injection temperature

The proper injection pressure and injection temperature can result in positive environmental effects [35]. The results of simulation in the case of rising injection temperatures have been presented in Figure 9 and 10. It was determined that the exhaust NO was reduced about 30% by increasing the fuel injection temperature from 320 to 360 K, whereas taking higher temperature caused soot formation to rise by 33%. Thus, increasing injection temperature created a competition between NO and soot formation.

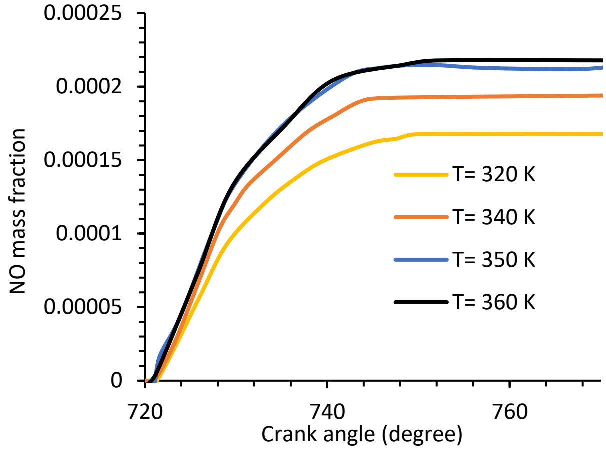

Fig. 9. Productions of exhaust NO caused by different injection temperature

3.4. Improvement of baseline piston bowl

NO pollution is highly toxic and a known carcinogen. Investigations show that parametric modification of piston bowl design affects NO emissions. Figure 11 illustrates that piston bowl geometry is classified into two parameters including R3 and Tm (bowl depth). The modifications of the R3 length are listed in Figures 12 and 13. It showed that exhaust NO can be reduced by rectification of the piston bowl dimensions (R3 and Tm length). It was found that the amount of exhaust NO was reduced (5.88 percentage) by the declining the size of R3 to 0.0079 m. Decreasing R3 length under 0.0079 is not recommended since the piston bowl geometry loses its balance.

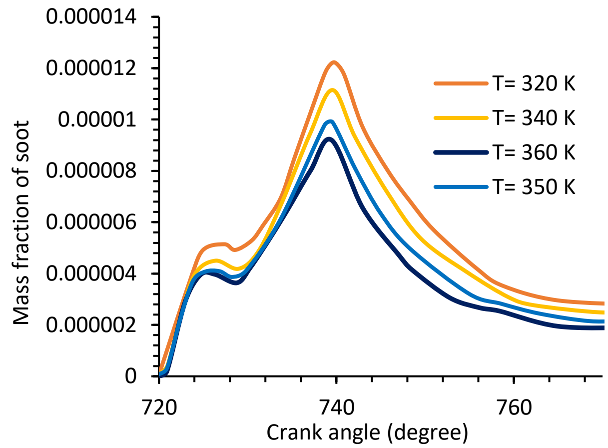

Fig. 10. Amount of produced soot caused by different injection temperature

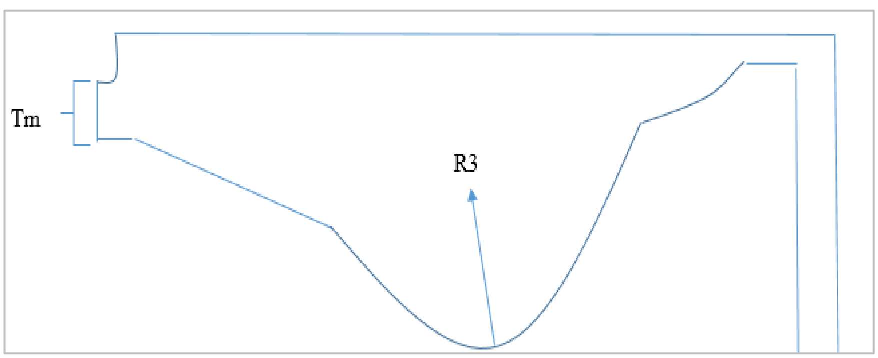

Fig. 11. Characteristics of baseline piston bowl

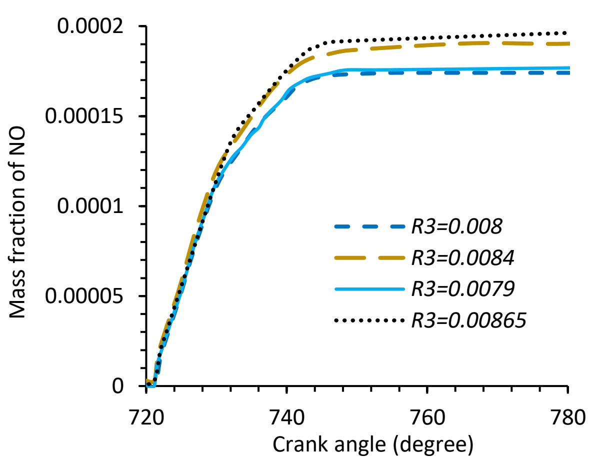

Fig. 12. Decreasing size of R3 and environmental effects (SOI= 1.5 deg BTDC)

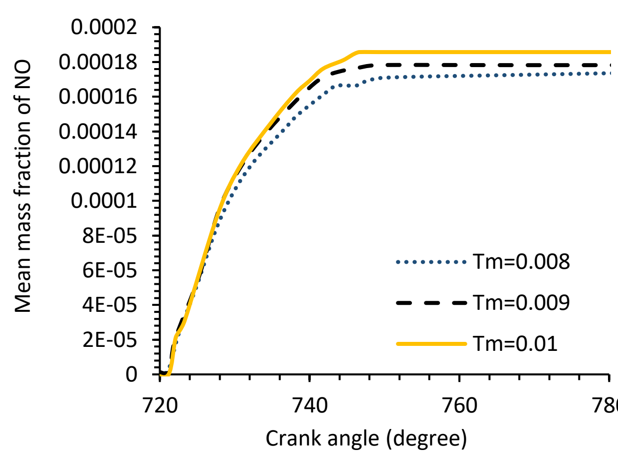

Fig. 13. Decreasing size of Tm and environmental effects (SOI= 1.5 deg BTDC)

As can be seen in Figure 13, exhaust NO was reduced (around 3.95%) by declining the Tm length to 0.008. For the baseline piston bowl, the reduction of the Tm length under 0.008 is not recommended.

3.5. Engine performance under various injection timings

The mean effective pressure (IMEP) is known as a variable relating to produce pressure. This is a measure of engine displacement to perform the process that is independent of any diesel engine capacity. As seen in Figure 14, indicated power, efficiency, and IMEP did not changed very much with a change in injection timing. This case confirmed that a 3 deg BTDC injection timing for a DME-burning HD engine in terms of environmental and engine performance was preferable at a constant injected mass and with the same conditions applied.

3.6. *Performance* of DME-burning engine vs. baseline engine:

As seen in Figure 15, thermal efficiency with changing fuel improved by around 21%. As indicated, IMEP and indicated power were higher than DME when taking pure diesel into account. It was predicted that by using different piston bowl geometry or advancing injection timing, this reduction in IMEP and indicated power can be compensated.

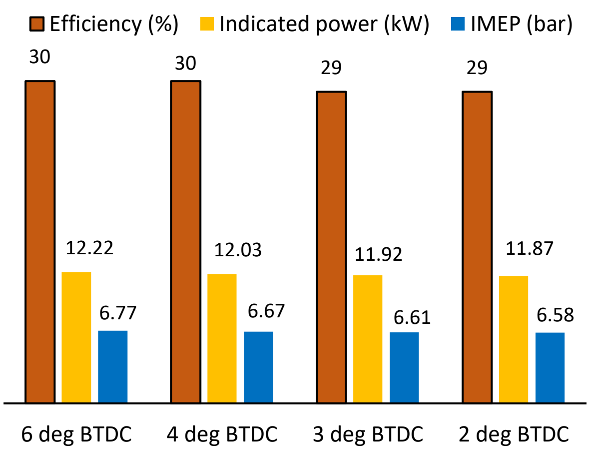

Fig. 14. Engine performance under various injection timings

3.7. Using EGR system for the reduction of NO emission:

The EGR system is one common way that can be used to reduce NO emissions. In studies on an ISM 370 engine, an EGR system has not been evaluated [10,3]. The effects of increasing the EGR composition on the exhaust NO were researched and are listed in Table 2. According to Figure 16, applying more EGR composition caused a reduction in NO emissions. The exhaust NO was reduced (37%) by increasing the EGR composition from 0 to 0.2. The reduction of exhaust NO by implementation of different EGR composition is determined in Table 2.

As displayed, the NO produced at 780 °CA was reduced by increasing the EGR composition and NO was reduced by about 19% by applying 0.11 of EGR composition. In addition, NO emission was dramatically reduced (95%) by considering a 0.2 EGR composition (in comparison to 0.11 EGR composition). It was found that the appropriate EGR composition for a DME-burning diesel engine was 0.2. It was clear that using the EGR technique affected in-cylinder mean pressure. The results of this study showed that this efficacy (changing in-cylinder pressure) for a DME-burning diesel engine was negligible (with lesser than 1 percent error).

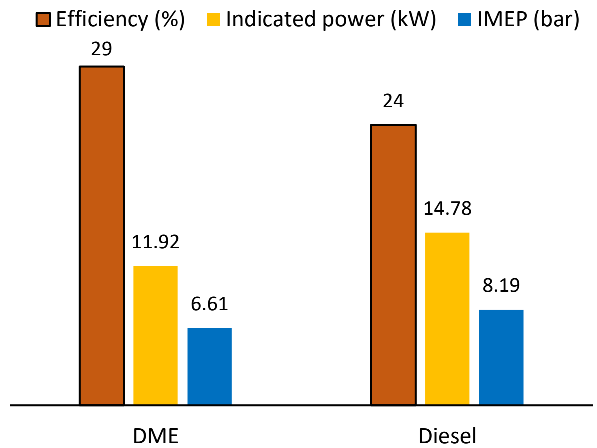

Fig. 15. DME-burning engine performance vs. baseline diesel engine

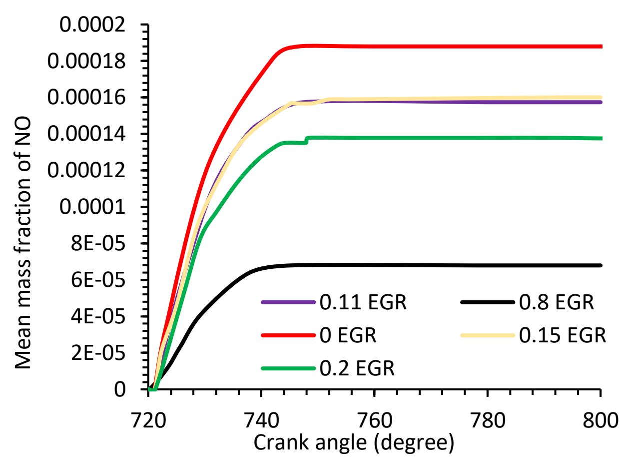

Fig. 16. Effects of increasing EGR composition on NO emissions after fuel injection

Table 2. Reduction of exhaust NO emissions with rising EGR composition (at EVO crank angle)

EGR Composition	%NO
0.11	19
0.15	17
0.2	37
0.8	72

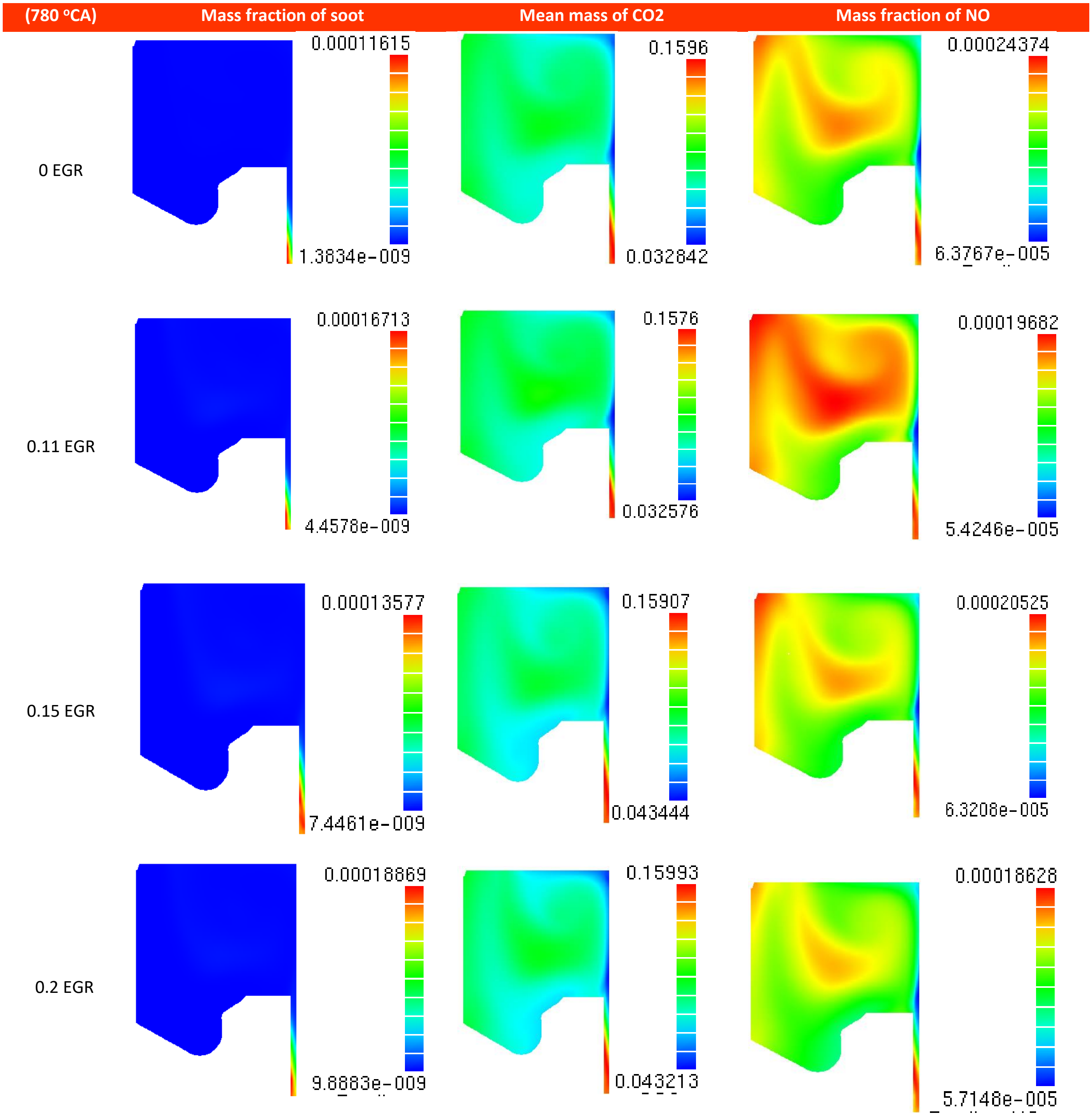

Fig. 17. Exhaust emissions distribution at 780 °CA

Exhaust gas recirculation (EGR) into the combustion chamber affected NO and other emissions (carbon dioxide and soot). As can be seen in Figure 17, soot emission increased by considering an EGR composition (declining the oxygen content and temperature with a rising EGR composition caused a reduction in the oxidation of the soot). It should be noted that the amount of soot formation was extremely lower than other pollutant gasses (carbon dioxide emission was over more than 1000 times of soot emission). It was determined that the production of NO and CO_2 was reduced with increasing EGR.

4. Conclusions

In the present study, the effects of injecting DME biofuel into an ISM 370 HD diesel engine were investigated. The results of the conducted studies showed that with a delay (changing the injection timing from 5 deg BTDC to 1 deg BTDC) in injection timing, NO reduction occurred by around 23% while the soot emission increased (about 52%). It was found that the preferable fuel injection timing to meet the lowest emissions (with considering a competition between soot and NO production) in a DME-burning HD diesel engine was 3 deg BTDC. Furthermore, changing the accumulated heat release at this injection timing in comparison to the baseline (2 deg BTDC) was inconsiderable (error is under 6%). Specifically, the delay in injection timing caused a reduction in peak pressure (from any injection timing to 740 crank angle). The results showed that a very small modification (1 mm) in the piston bowl geometry had a significant impact on emissions. It became clear that NO emissions were reduced by decreasing R3 and Tm lengths (piston bowl geometry). Subsequently, with the modification of the R3 length from 8.4 mm to 7.9 mm, the amount of exhaust NO was reduced by around 10.25 %; with a decrease in Tm from 10 mm to 8 mm, NO production was reduced by around 5.9 %. In the present work, the appropriate length for R3 (7.9 mm) and Tm (8 mm) were specified. The EGR system reduced tailpipe NO emission. By considering DME fuel and by applying 0.2 EGR composition, the exhaust NO was reduced by around 37%. The appropriate (from the environmental standpoint) temperature for DME fuel injection into a combustion chamber was determined to be 340 K. As in other studies, the dimethyl ether fuel was considered usable for an ISM 370 HD diesel engine.

Nomenclature

DI	direct injection
HD	heavy duty
SOI	start of injection
EGR	exhaust gas recirculation
CA	crank angle
BTDC	before top dead center
IMEP	mean effective pressure

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