Designing different piston bowls for reduction of emissions and improvement of power parameters in a dimethyl ether-burning diesel engine

Kianoosh Shojae[1] **and Majid Mahdavian**[2]

[1] *Faculty of Chemical Petroleum and Gas Engineering, Semnan University, Semnan, Iran*
[2] *Chemical engineering, Quchan University of Technology, Quchan, Iran*
[*] *Corresponding author: m.mahdavian@qiet.ac.ir*

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By development of industries, excessive consumption of the fossil fuels is becoming the worldwide crisis in the years ahead. Besides the proper thermal and power performance, the compression ignition engines make lower greenhouse-gas emissions compared to the spark-ignition engines. Using dimethyl-ether (DME) as an environmentally friendly fuel in a diesel engine can improve its advantages. In the present paper, the effects of the piston bowl geometry in a DME-burning direct injection (DI) engine have been numerically evaluated. In order to enhance combustion and emission characteristics, various piston bowls with different bowl volumes are designed and examined. Furthermore, the influences of engine speed and compression ratio are further investigated. The results showed that lower bowl volume led to more air-fuel mixing. Reduction of the bowl volume size from 6.446e-005 m3 to 1.5282e-005 m3 caused reduction in emissions of the soot, NO and CO by 91%, 9.3%, and 99%, respectively (the exhaust CO2 concentration was almost identical for each piston bowls). in addition, lower compression ratio caused reduction in temperature and NO emissions. It is determined that amount of exhaust emissions is affected by increasing the engine speed, and mean pressure of the engine cylinder reduced, dramatically. It is found that taking advantage of DME fuel in the ISM 370 diesel engine caused reduction in the NO, soot, CO and CO2 by 75%, 20%, 8%, and 44.43%, respectively, under 1200 rpm engine speed.

keywords: *DME fuel, piston bowl design, exhaust emissions, engine performance, air-fuel mixing.*

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1. INTRODUCTION

Greenhouse gases are the main reason of the global warming. It is expected that the use of fossil fuels will reduce in the next few years regarding the Environmental Protection Agency (EPA) requirements, harmful impacts, and capacity limitations. Investigations on dimethyl ether (DME) fuel have been started since 1995. DME was introduced as a clean and environmentally friendly fuel at around 1999, which was known as a future fuel with great potential. DME has been currently produced in many countries such as Japan, China, Sweden, etc. The major benefit of DME fuel is that it can be produced through different processes from various feed-stocks. Furthermore, DME fuel has a high cetane number, which makes it a proper fuel for compression ignition engines. Several investigations have determined the following properties of DME compered to diesel; 1 DME can produce ultra-low emissions, 2 DME makes better thermal efficiency, 3 noise of a DME-burning engine is similar to that spark engines, and has higher vapor pressure than diesel, which leads to faster evaporation [1]. The compression ignition engines such as ISM 370 engine are widely used in heavy-duty (HD) vehicles, buses, etc. due to its many benefits including lower costs, higher performance, and longer expected life compared to the spark engines [2]. Therefore, topic of using the clean fuels like DME as an alternative to diesel fuel in a HD diesel engine is very noteworthy (in addition to DME, biodiesels and hydrogen fuels are also promising alternatives instead of pure diesel, which have superior environmental and combustion advantages in comparison to fossil fuels [3–5]). Combustion performance and the emission characteristics of a turbocharged diesel engine fueled with DME and biodiesel blends have been experimentally investigated and lower the NO emissions were resulted by choosing a proper nozzle size [6]. Fuel with a high cetane number enhances the performance of a compression ignition engine by improving the combustion of the charge inside the engine. Furthermore, the quality of a fuel can be improved by modifying its structure or adding a fuel with a high cetane number [7, 8]. To reduce the exhaust emissions, various methods can be examined like using the Exhaust Gas Recirculation (EGR) system. The EGR system dilutes the oxygen content in intake air and causes the recirculated gases to work as heat absorbents to decrease in-cylinder temperature because the NOx emissions are produced in high-temperature conditions. Investigations show that the EGR system or Atkinson cycle combined with EGR can be greatly useful [9, 10]. An experimental investigation has been conducted on a turbocharged diesel engine managing high/low-pressure EGR systems, and reduction in exhaust NO was achieved around 50% [11]. Thermal efficiency and exhaust emissions from each engine are very effective factors in engine design, and the brake specific fuel consumption (BSFC) are not fully known for some hybrid engines, and a compression ignition engine should be designed considering these parameters [12]. The impacts of cylinder design on engine performance and tailpipe emissions have been evaluated by many scholars in the past, since the suitable design has significant advantages, which can be obtained regarding the lower exhaust emissions, without affecting the engine power. The effect of piston bowl geometry on the engine efficiency and power is complex because of its impacts on the in-cylinder flow and the air spray interaction [13]. Hydrogen as a carbon-free fuel or low-carbon fuel like DME has excellent the combustion and emission characteristics [14, 15]. Lalv et al [16] investigated the emission as well as combustion characteristics of a diesel engine in comparison to its modified piston. They have used conventional diesel fuel with 20% blend of adelfa biodiesel and determined that mixed fuel reduced brake thermal efficiency. However, combustion characteristics, emissions (like carbon mono oxide and hydrocarbon), and thermal brake efficiency were improved by modifying piston shape. Kim and park [17] conducted an optimization investigation on emissions along with fuel consumption in a DME-burning diesel engine. In this regard, they have used the genetic algorithm (GA) method to indicate the optimal operating conditions in the case of decreasing exhaust emissions and fuel consumption at different operating conditions by taking into account of the KIVA-3V code. They indicated that the equivalence ratio (ER) of the optimized case by utilizing DME fuel might be decreased in comparison to diesel fuel because of the soot-free produced during the combustion of DME. Nazemian et al [18] investigated the piston shape and injection pattern of RCCI engines for improving waste heat recovery by using DOE method and showed that utilizing improved injection strategy, bowl diameter and depth have the most effective influences on the waste heat recovery capacity from exhaust emissions. Benajes et al [19] studied the effect of piston bowl shape on the RCCI diesel engine, experimentally, and showed that piston shape has a high influence on combustion progression. They showed that using double injection strategies can reduce the NOx and soot emissions. Channappagoudra et al [20] have conducted a comparative study in the case of a diesel engine and enhanced engine with changing different sections of piston bowl. Their experimental studies showed that the modified piston (Re-entrant Toroidal Piston Bowl Geometry) provided a proper performance and combustion as well as emission properties in comparison to standard engine and modified engine (ME) by considering various piston bowl shapes that called Hemispherical Piston Bowl Geometry, Toroidal Piston Bowl Geometry, and Straight Sided Piston Bowl Geometry. This enhancement might be ascribed to enhanced fuel atomization, smaller size droplets, swirl, and squish, etc. Khan et al [21] have investigated the combined influences of spray angle as well as the piston bowl geometry in terms of on mixing, performance and emission properties of a diesel engine. For simulating the combustion chamber, AVL FIRE CFD code has been used, and experimental outcomes for the baseline hemispherical bowl are utilized for validating the numerical model. They have shown that spray angle can dramatically affect the mixing and combustion progress for all considered piston bowl. They stated that the engine having TRCC type of combustion chamber can provide better combustion and emission performance.

Table 1. Models applied simulation stages

Models	
Turbulence	k-zeta-f
Evaporation (spray)	Dukowicz
Breakup (spray)	Wave
NO emission	Extended Zeldovich
Soot emission	Kinetic model
Combustion model	Coherent flame model
Ignition model	ECFM-3Z

The ISM37 heavy-duty diesel engine still needs more investigations in the case of using different fuels like DME under different conditions. In addition, matching these fuels with the engine by modifying various parameters like the piston bowl geometry also requires more examinations. In our previous work [22], we investigated an EGR system and some piston bowl parameters only for 1200 rpm engine speed (fueled by DME fuel). However, in this paper, we have considered different piston bowl geometries under different engine speeds and also compared the results with the same considered engine in identical conditions. The objective of this study is threefold: first, to investigate in detail the effects of the piston bowl (PB) geometry on combustion and emission characteristics in a HD diesel engine fueled by the DME in the range of engine speeds from 1200 to 2000 rpm; then, to investigate the effects of compression ratio (CR) value on combustion and emission performance; finally, to study on the influences of the DME as an alternative fuel instead of pure diesel. 1D, 2D, and 3D analyses have been considered in the paper and to construct a numerical model of the engine, AVL FIRE software has been employed.

2. SIMULATION

The ISM 370 Cummins HD diesel engine has been chosen to investigate the effects of the DME on combustion and emission characteristics. The piston bowl geometries were designed in AVL ESE, and other analyses (1D, 2D, and 3D) were done in AVL WM. For modeling the spray and combustion process, listed models in Table 1 have been employed.

For determination of turbulences, the K-Zeta-F model was employed. This model has a good constancy for the unfixed meshes and simulation of the ISM 370 engine with it has been successfully performed [14]. This turbulence model is derived from Reynolds stress transport model. It also considered as simplified second moment closures. In this model, the wall damping

Table 2. Engine specifications

ISM 370 diesel engine	
Engine model	ISM 370
Engine layout	inline
Number of cylinders	6
Injection quantity fuel for each turn	35.634 mg
Displacement	10.8 L
Bore	0.125 m
Stroke	0.147m

influences is combined using an auxiliary equation that used as a measure in the case of the normal to the wall oscillations. The engine characteristics are listed in Table 2. Three main sources of NOx emissions exist in combustion process including fuel and thermal as well as prompt. For modeling NO emission, the Extended Zeldovich model was utilized because the diesel engine works in high temperature and the fuel and prompt NO is inconsiderable. In this paper, the ignition model of three-zone extended coherent flamelet (ECFM-3Z) is employed for modeling ignition. The model of ECFM-3Z is a sort of ignition model that is relevant to laminar flamelet motion. Based on the ECFM-3Z model the DME reaction mechanism is introduced as follows [23]:

$$C_2H_6O + 3O_2 \rightarrow 2CO_2 + 3H_2O \tag{1}$$

$$C_2H_6O + \frac{1}{2}O_2 \rightarrow 2CO_2 + 3H_2 \tag{2}$$

The Kinetic model for modeling soot emissions was applied. Kennedy [24] has investigated models of soot formation and oxidation. For the prediction of the collision of the fuel jet with cylinder wall and heat transfer the standard wall jet model was used, and uniform distribution of temperature was predicted by the Dukowicz evaporation model [25]. Boundary conditions and the fuel properties considered for all designed piston bowls are shown in Table 3.

The simulations were started from 570 CA (when the intake valve closed) and were ended 810 CA (when the exhaust valve opened). Initial pressure and temperature are 0.19 Mpa and 360 K, respectively. Intake air is ordinary and naturally aspired. The swirl number [26] can characterize the degree of swirl for flow and has been introduced as follow:

$$S_{CB} = \frac{G_{tg}}{RGax} = \frac{\int_0^R wur^2 dr}{R\int_0^R u^2 r dr} \tag{3}$$

G_{tg} is the axial flux of the tangential momentum, Gax is the axial momentum and where R stands for the outer radius of the annulus.

A. Mesh sensitivity analysis

In the paper, the mesh sensitivity analysis and the validation of simulation have been performed under a constant engine speed (1200 rpm). Due to the complex grid structure of the piston bowl geometry, a wide range of mesh numbers have been examined (80000 to 200000). Fig 1 shows the mean pressure curve calculated by considering the different number of meshes

Table 3. The boundary conditions considered piston bowls and fuel properties [23]

Boundary conditions	
Piston (Wall-Temperature)	540K
Linear (Wall-Velocity-Temperature -Mesh movement)	400K
Axis (Symmetry)	-
Com. Volume (Wall-Heat flux-Mesh movement)	-
Head (Wall-Velocity-Temperature)	515K
Fuel properties	
Molecular Formula	CH_3OCH_3
Ignition point	$235°C$
Molecular Weight	$46.069\, g/mol$
Cetane number	60
Viscosity	$3\, kg/ms\, at\, 25°C$

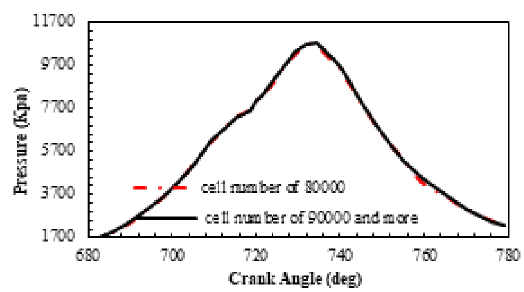


Fig. 1. Comparison of the cylinder pressure curves after cell number of 80000.

from 80000 to 200000. It is observed that in the considered range of meshes, pressure distribution curves are perfectly matched for 90000 and more cells and thus the simulations have been performed with 90000 meshes, approximately. All the analyses in the paper were conducted at a constant injection timing (1.5 deg BTDC).

3. RESULT AND DISCUSSION

A. Validation

In the present paper, AVL fire CFD code was employed and validation of this software was carried out on the ISM 370 engine (fueled by diesel) and a Common-Rail (CR) engine fueled by neat DME. To confirm the reliability of the simulation results, the model should be guaranteed for some verifiable simulation data. For this reason, in-cylinder mean pressure, peak pressure, rate of heat release (RHR), mean maximum bulk mixture tem-

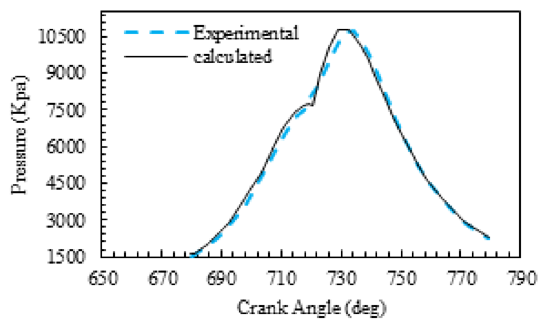


Fig. 2. Comparison of mean pressure between experimental [14] and predicted data for the ISM 370 engine (maximum error by 5%)

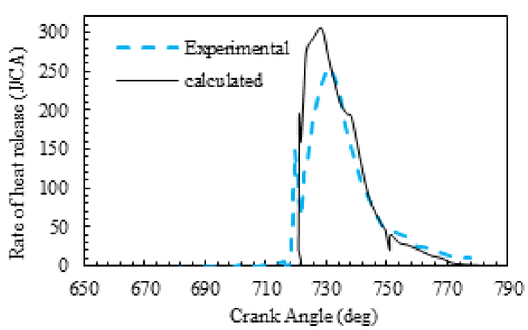


Fig. 3. Comparison of the experimental [14] and calculated RHR for the ISM 370 engine (maximum error by 10%)

perature, the HC and NO have been considered to simulation assays. As seen in Figures 2 and 3, for the ISM 370 diesel engine the calculated mean pressure has a good agreement with the experimental data [14] with mean errors under 3% and the calculated, and experimental RHR curves are in general accord with each other. However, the predicted RHR curve is larger than experimental data from 720 to 734 CAs, which can be justified by the following reasons; (a) errors made by the simulation, calculations did not consider the heat transfer, thermal radiation, latent heat of vaporization, etc., (b) potential errors during the experiment [14].

The predicted NO emission and maximum bulk mixture temperature values compared to the experimental data are shown in Table 4. It is observed that there are good agreements between measured and calculated values.

Table 5 illustrates the specifications of a CR engine [27] fueled by DME to examine the software when DME fuel is used. Table 6 shows that experimental [27] and calculated the HC, CO, mean pressure, and peak pressure values are in a good agreement with each other.

It is determined from the above results that the predicted and experimental data had values close to each other for both ISM 370 engine (fueled by diesel) and the CR engine (fueled by DME). Therefore, the constructed numerical model is reasonable and the ISM engine simulated by the AVL software can be launched by DME fuel if all considered parameters during simulation of the engine fueled by the diesel such as injected mass, injection timing, initial pressure, swirl, etc. remain constant.

Table 4. Comparison of NO emissions and maximum bulk mixture temperature between calculated and experimental data for the ISM engine fueled by neat diesel

	Calculated value	Experimental value [15]
NO (g/kW-hr)	3.97	4.38
Max. Bulk Mix. Temp. (K)	1910	1630

Table 5. Specifications of the CR engine

The CR engine	
Engine layout	inline
Valve system	4 valves
Injection quantity fuel for each turn	8 mg
Displacement	373.3 cm3
Bore & Stroke	75 & 84.5 mm

Table 6. Comparison of the calculated and experimental values for the CR engine fueled by pure DME

	Calculated value	Experimental value [27]
HC (ppm)	0.024	0.03
CO (%)	0.0161	0.021
Peak cylinder pressure (Mpa)	5.6	5.6
	Mean errors from 570 to 810 CAs	
Mean pressure	Under 5%	
RHR	About 20%	

B. Environmental effects of the engine speed and piston bowl geometry

In an engine fueled by DME peak pressure reduces compared to the engine fueled by diesel in same conditions such as identifying the injected fuel mass, initial pressure, etc. due to the lower heating value of DME fuel compared to diesel fuel [1]. For this reason, various piston bowls in a constant CR value were preliminarily tested (by changing piston bowl parameters) in case of making acceptable in-cylinder mean pressure compared to the diesel-burning engine and among of them only geometries listed in Fig. 4 did not prevent the combustion and peak

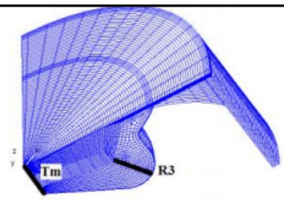
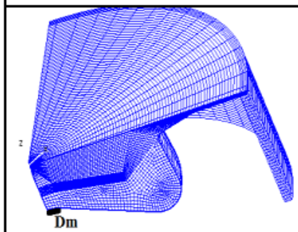
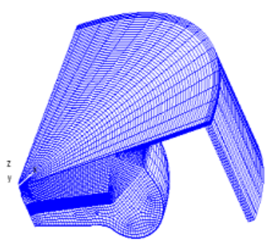
Piston bowl	Some of geometrical variables
 Piston bowl 1- Single curved (PB1)	R3: 0.00865 m Tm: 0.01 m Dm: 0.001 m Bowl volume:7.446e-005 m^3
 Piston bowl 2- Omega shape (PB2)	R3: 0.00208 m Tm: 0.02235 m Dm: 0.01 m Bowl volume:8.29679e-005 m^3
 Piston bowl 3 (PB3)	R3: 0.0051 m Tm: 0.00496 m Dm: 0 m Bowl volume:1.5282e-005 m^3

Fig. 4. The considered piston bowls designed by changing geometrical parameters of baseline piston bowl

pressure generated by these piston bowls (when engine is fueled by DME) had acceptable values compared to the diesel-burning engine. Therefore, the piston bowl geometries introduced in Fig. 4 have been chosen and examined to find the best piston bowl among them. In the following the effects CR value and using DME fuel for the baseline engine will be investigated.

It is known that better air-fuel mixing leads to improving the combustion and the piston bowl geometry has direct effects on it. Fig.5 illustrates the relationship between engine speed and NO emissions. It is determined from comparing the upper and lower speeds that NO emissions are reduced by increasing the engine speed and 2000 rpm engine speed makes lower NO emissions compared to other speeds because in the higher engine speeds, mean temperature is reduced (as seen in Fig. 9).

Fig. 6 shows soot mass fraction versus crank angle in vari-

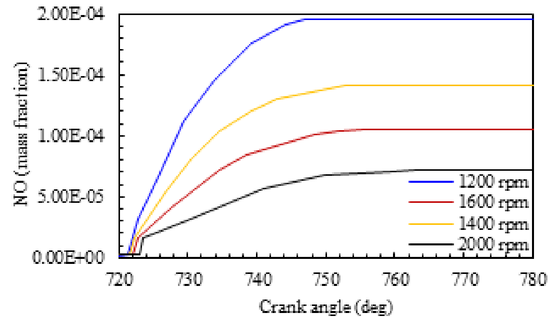


Fig. 5. The engine speed effects on NO emissions (PB1)

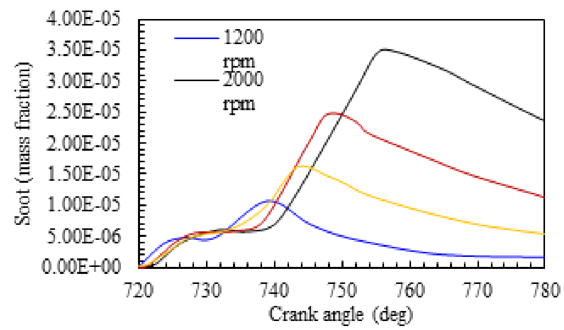


Fig. 6. The engine speed effects on soot emissions (PB1)

ous engine speeds. As illustrated, in engine speed of 2000 rpm amount of soot emission is in the highest concentration in comparison with other engine speeds for PB1. As determined in the previous section (study on NO emissions), amount of exhaust NO is increased in higher engine speed, mutually, soot formation is reduced by increasing the engine speed because lower speed makes higher temperature and causes soot oxidation to occur, dramatically. It should be noted that the DME is an oxygenated fuel and does not have C-C bonds in its chemical structure so soot emissions from a DME burning engine are inconsiderable [27].

Amount of NO and soot mass fraction of the engine equipped to the PB2 is calculated and presented in Fig. 7. As denoted, by increasing the engine speed amount of NO emissions are reduced and the same results as the PB1 are obtained for the PB2.

Fig. 8 shows NO and soot emissions from the engine equipped by the PB3. The PB3 has the lowest bowl volume among of designed the piston bowls (PBs), which caused negative effects on air-fuel mixing but producing lower NO emissions is one of its advantages (as seen in Fig. 10).

Fig. 9 (a) and (b) denote that by increasing engine speed from 1400 to 2000 rpm, the in-cylinder mean pressure and temperature are decreased and combustion is delayed. Furthermore, increasing the engine speed reduces the time required for chemical reactions, which causes more combustion that is incomplete. In

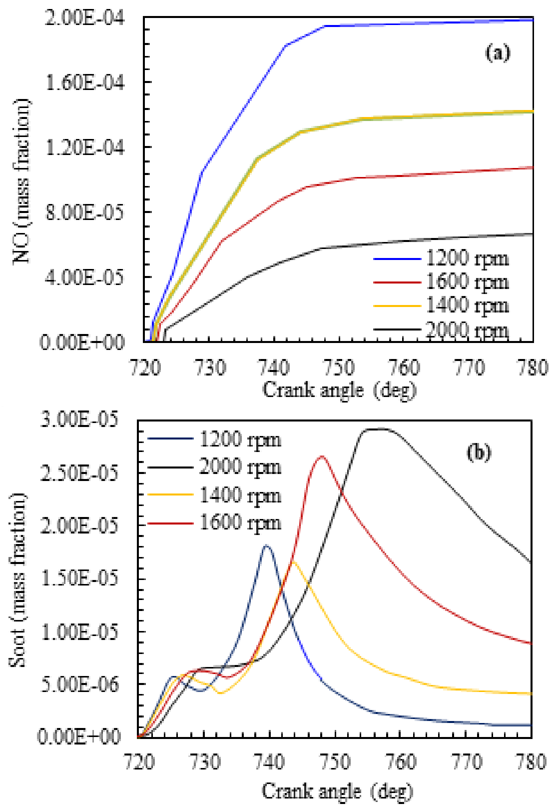


Fig. 7. The engine speed effects on soot and NO emissions (PB2) (a): mass fraction of NO (b): mass fraction of soot

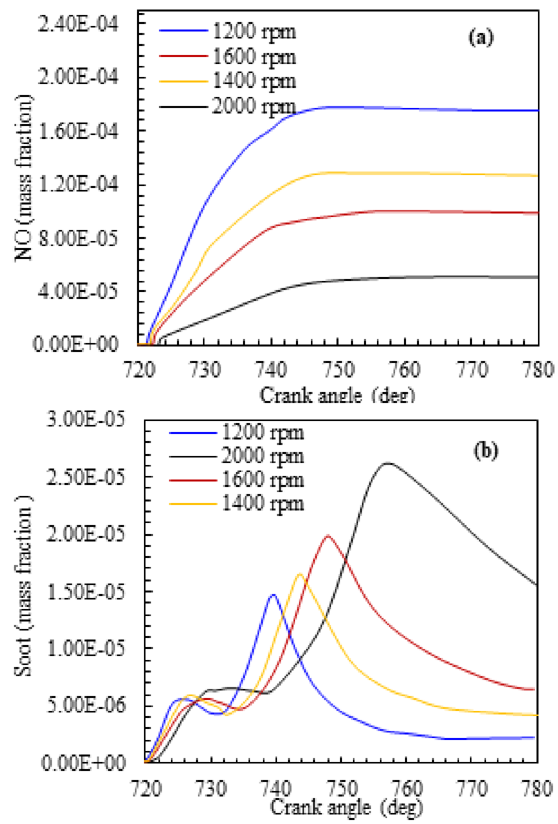


Fig. 8. The engine speed effects on soot and NO emissions (PB3) (a): mass fraction of NO (b): mass fraction of soot

addition, higher engine speed reduces the time for heat transfer, which may cause the higher average temperature in the engine cylinder [28].

In this section, results were presented only for a constant speed (1400 rpm) due to the similarity of the obtained results. Fig.10 shows that the PB2 produced almost identical NO in comparison to the PB1 and when the engine cylinder is equipped with the PB3, NO mass fraction is reduced nearly 9.3% and soot formation is decreased by around 91%, approximately. The PB3 has lower bowl volume so air-fuel mixing performed properly. Lower air-fuel mixing can cause increasing incomplete combustion. However, as can be seen in Fig. 10, using the PB3 led to reducing NO emissions. This reduction in NO emissions is due to the fact that the PB3 make a wide high-temperature zone in the range of 750 to 780 deg crank angles. It is obvious that increasing temperature leads to more NO production and the lower high-temperature zone is the main reason for lower NO emission of the PB3.

As seen in Fig.11, a lower equivalence ratio is observed for the PB3 structure, which results in improved air-fuel mixing compared to other piston bowls. This is a reason for the lower soot emissions from the engine equipped with the PB3.

Fig. 12 indicates the relationship between the piston bowl geometry and emissions of the CO and CO2. As seen, CO concentration is mainly affected by changing the piston bowl geometry and replacing the PB2 or PB3 instead of the PB1 caused a reduction in its value. Therefore, proper piston bowl geometries (in term of CO emissions) are found the PB2 and PB3. High temperature and oxygen content values in cylinder lead to the process of the CO converts to the CO2 and this is the reason of lower CO formation of the PB3 or PB2. Fig. 12 (b) determines that the PB1 structure caused lower CO2 emissions compared to other the PBs.

The most important factors affecting the formation of NO emissions are oxygen concentration, gas temperature, and high-temperature duration. It is determined that the PB3 causes more NO emissions compared to other PBs and makes an increase in soot emissions. Figs. 13 and 14 show that mean average temperature in cylinder and the mass fraction of oxygen, respectively, which are affected by changing the piston bowl geometry. As shown in Fig. 13, the PB3 makes the lower temperature in the cylinder between 750 to 780 CAs and this wide low-temperature region is the main reason of lower formation of NO emissions and more oxygen consumption by the fuel, soot, etc. It is found that the PB1 compared to other the PBs increases the in-cylinder temperature after 750 CA and causes reduction in oxygen content.

As seen in Fig. 14, the mass fraction of unburned DME from the engine is very low for the designed piston bowls. As previously mentioned, various piston bowls were preliminary designed and examined in the case of making the proper mean pressure compared to the engine fueled by diesel and only de-

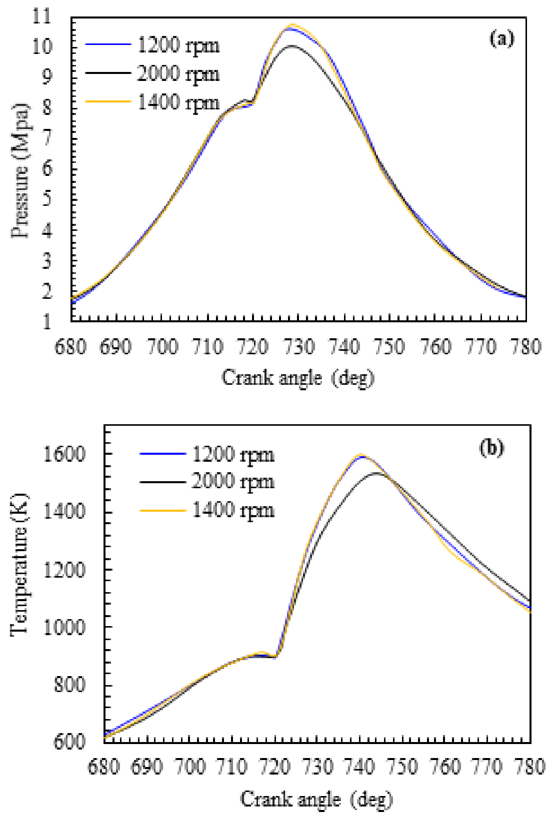


Fig. 9. The engine speed effects on in-cylinder mean pressure and temperature (a): mean pressure (b): mean temperature

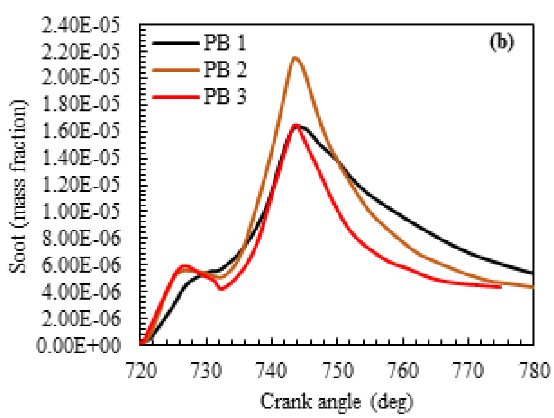


Fig. 10. The piston bowl geometry effects on NO and soot mass fraction (a): NO mass fraction (b): soot mass fraction

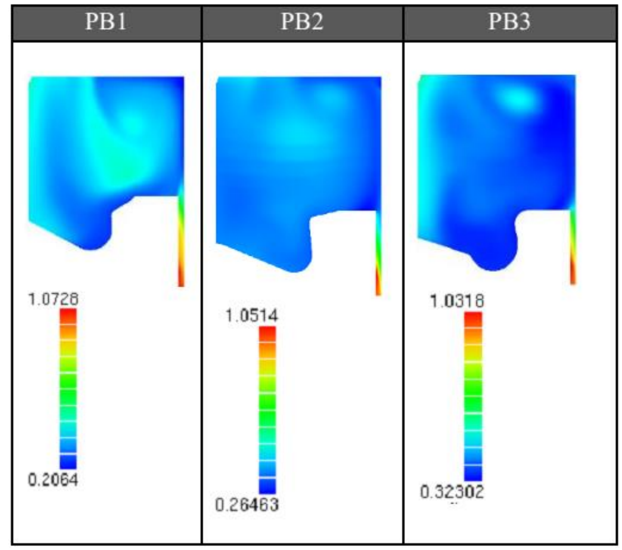


Fig. 11. Equivalence ratio distribution almost when the exhaust valve is opening for different piston bowls

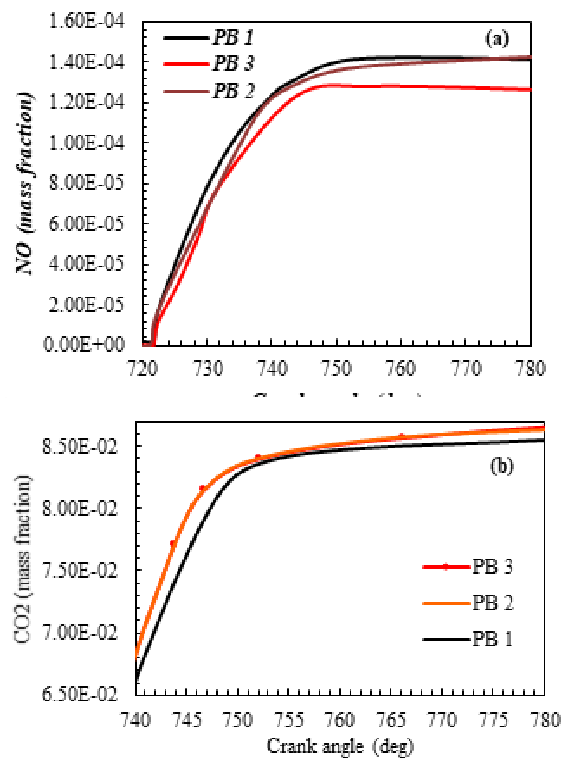


Fig. 12. Effect of changing the piston bowl geometry on the (a): CO and (b): CO2

signed piston bowls introduced in Fig. 4 had acceptable the in-cylinder pressure (or temperature); and one of the reasons for low unburned DME emissions for three designed piston bowls is proper the in-cylinder temperature. However, among three piston bowls designed by AVL ESE the PB2 and PB3 make lower unburned fuel compared to the PB1. As seen in Fig. 14, the PB2 and PB3 consumed more oxygen amount due to making a wide

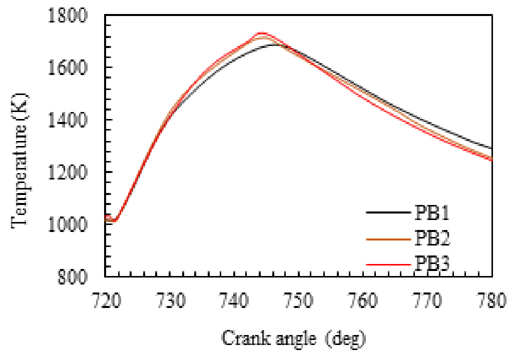


Fig. 13. The mean temperature in cylinder under 1400 rpm engine speed

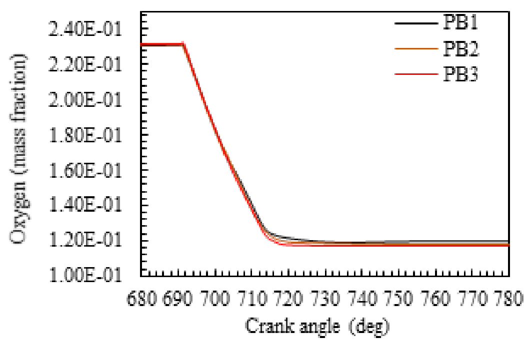


Fig. 14. The oxygen mass fraction under 1400 rpm engine speed

high temperature from 730 to 750 CAs, which means more fuel consumption. Fig. 15 shows that for the PB2 unburned DME emission is largely distributed onto the cylinder wall, which results from the quenching influences.

C. The combustion effects

In this section, the effects of engine speed and piston bowl geometry on the thermal efficiency, power and BSFC have been investigated and influences of changing the piston bowl geometry on the turbulence kinetic energy (TKE) and swirl are further discussed. Fig. 16 shows the combustion performance of the engine when piston bowl geometry is changed. As determined, the piston bowl and engine speed affected combustion characteristics and the PB1 caused lower thermal efficiency and power. BSFC is an economic parameter and increase of its value means more fuel has to consume to meet the same power, and this issue is costly. As seen, the PB1 had higher BSFC in comparison to other piston bowls. Fig.16 also shows the effects of engine speed on combustion characteristics. As observed, increasing the engine speed led to more fuel consumption and caused power and thermal efficiency to be reducing. However, the PB1 structure made higher the BSFC compared to other PBs in all of the intended engine speeds and only using PB2 geometry caused a moderate combustion performance.

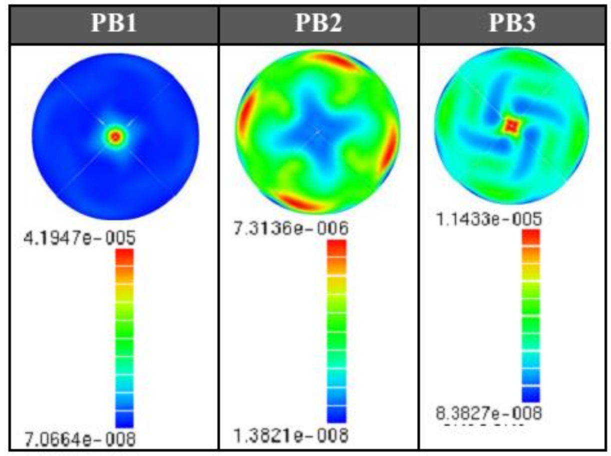


Fig. 15. Unburned DME mass fraction in combustion chamber almost when the exhaust valve is opening under 1400 rpm engine speed

The air-fuel motion into the cylinder of a diesel engine is generally determined by the swirl and turbulence. The swirl and TKE directly affect air-fuel mixing in the combustion chamber. Swirl motion of the air is created because of the type of intake port. Choosing an appropriate intake port and piston bowl geometry will create a good swirl and improve combustion [29]. 2D and 3D CFD simulations involving the flow motion pattern were considered to investigate the effect of swirl induced by PB1, PB2, and PB3 on the exhaust emissions and engine performance for the ISM 370 engine. Fig. 17 shows the swirl and TKE distribution versus crank angle induced by different piston bowl structures. As seen, due to limiting the space for the air motion in the combustion chamber at TDC, the swirl and TKE are in the highest values, and the PB2 geometry has higher TKE and swirl in TDC crank angle (720 CA) compared to other the PBs. The PB1 induced weak air swirl and TKE, which can be improper for the combustion due to making lowest power, thermal efficiency, higher the BSFC, and increasing soot and CO emissions. However, the PB3 structure created proper swirl and turbulence, which leads to higher performance and more NO emissions in comparison with other PBs, but the PB2 structure made the balanced conditions between combustion and emission performance. As illustrated, the manner of air swirl and turbulences was varied by changing the engine speed and increasing its value led to lower swirl and higher turbulence. The PB3 due to its lower bowl volume compared to other PBs improved air motion, and this case is the main reason of higher the swirl number and TKE.

Fig. 18 denotes the in-cylinder temperature, soot, NO and TKE distributions in different crank angles for the PB2. As seen, a higher temperature is the main reason for increasing NO emissions. At 740 CA (after the start of combustion), all fuel has been injected for a cycle and cylinder temperature is higher than 720 and 780 CAs. Conversely, in higher temperature, soot oxidation occurs better and soot emissions had lower values in higher temperature area. So, in 780 CA due to the passage of a wide high temperatures (from 730 to 780 CAs) zone, soot emissions are reduced.

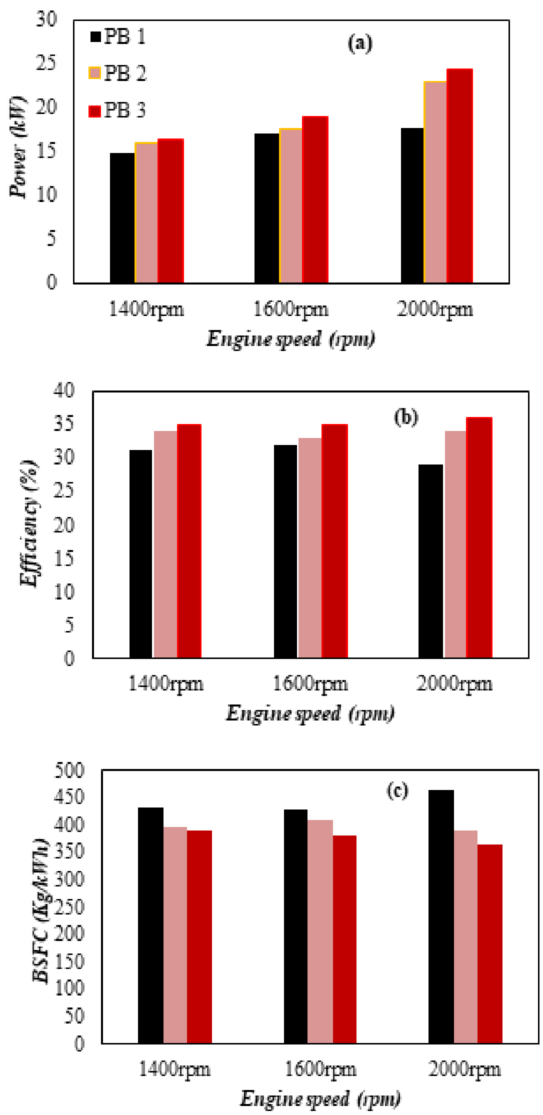


Fig. 16. Comparison of power, thermal efficiency and BSFC under engine speeds of (a):1400 rpm, (b): 1600rpm, and (c): 2000 rpm

At first, the TKE in the combustion chamber is in the lowest values before fuel injection but when fuel is injected into the cylinder, after a short time TKE is increased and at near 740 CA, this parameter is higher than other crank angles.

D. Effects of CR

The of an engine is a value that shows the ratio of the volume of its cylinder from its largest capacity to its smallest capacity. In this section, environmental and combustion effects of the PB1 with different CR values of 14, 16, and 25 have been investigated. The method to change CR was to reduce the piston bowl depth, center bowl depth, etc. and the purpose was to investigate the effects of CR, individually. Fig. **??** show that the increase of CR from 14 to 16 can improve the fuel economy (BSFC) under 1400 rpm engine speed, but a significant decrease in the power and thermal efficiency of CR 16 occurs. High CR makes higher in-cylinder pressure at spark timing and lower turbulence and smaller turbulence length scale. Modification of piston bowl may have a negative effect on the combustion, but the high CR can enhance the combustion pressure, which is a beneficial effect. Thus, the best CR should be chosen regarding the balanced state between combustion and emission characteristics [30]. As observed in Fig. **??**, the CR has significantly effect on BSFC, power, and thermal efficiency. When the CR sets to 25, BSFC reduces and power and efficiency increase dramatically. Therefore, increasing the CR value is useful and can enhance the engine in the case of emissions, efficiency, performance and the fuel economy (lower BSFC means that less fuel is required for generating a constant power). The incomplete combustion and CO2 cracking due to the high temperature are the main reason for CO emissions. As seen in Fig. 19 (c), higher CR caused lower CO emissions, unexpectedly. It is determined that reducing exhaust CO concentration for CR of 25 is due to the wide lower temperature zone from 750 to 810 CAs (as seen in Fig. 20). Furthermore, soot oxidation in CR of 14 is reduced because of generating lower mean temperature (can be observed in Fig. 20) compared to other CRs and this case is the main reason of higher exhaust soot concentration for it. As seen in Figures **??** and 19, an increase of the CR affected emission and combustion parameters, and CR of 16 was known proper in the case of making the balanced state between them.

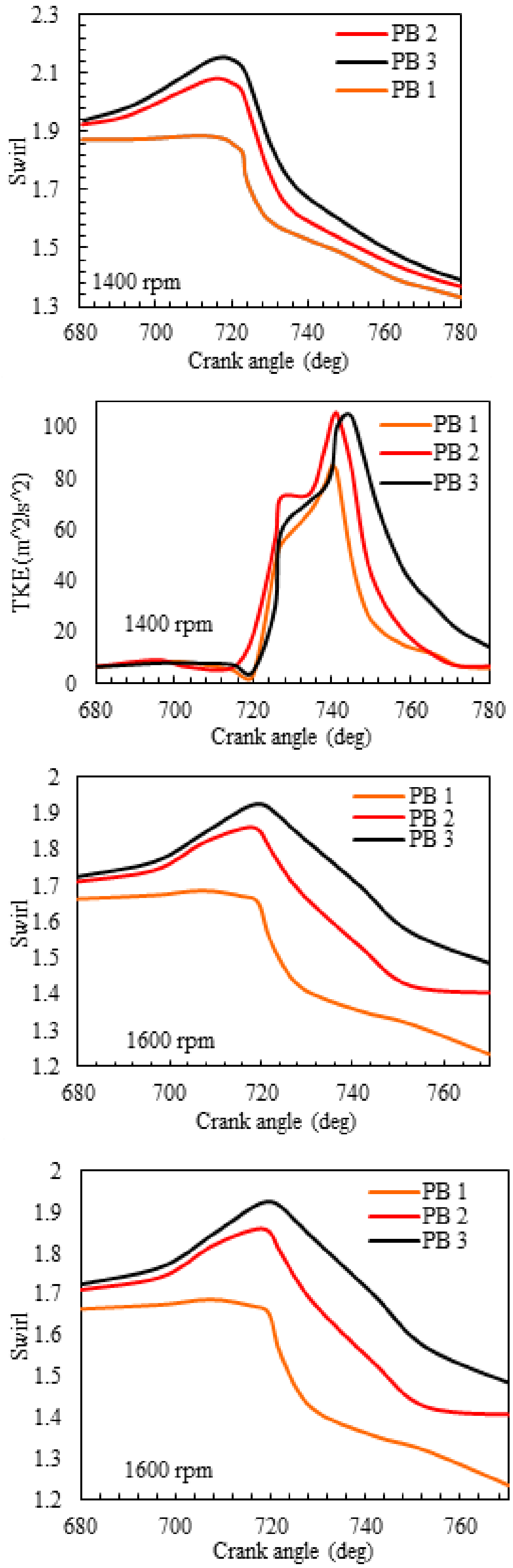


Fig. 17. Distribution of swirl and TKE versus crank angle for three piston bowl geometries under 1400 and 1600 rpm engine speeds

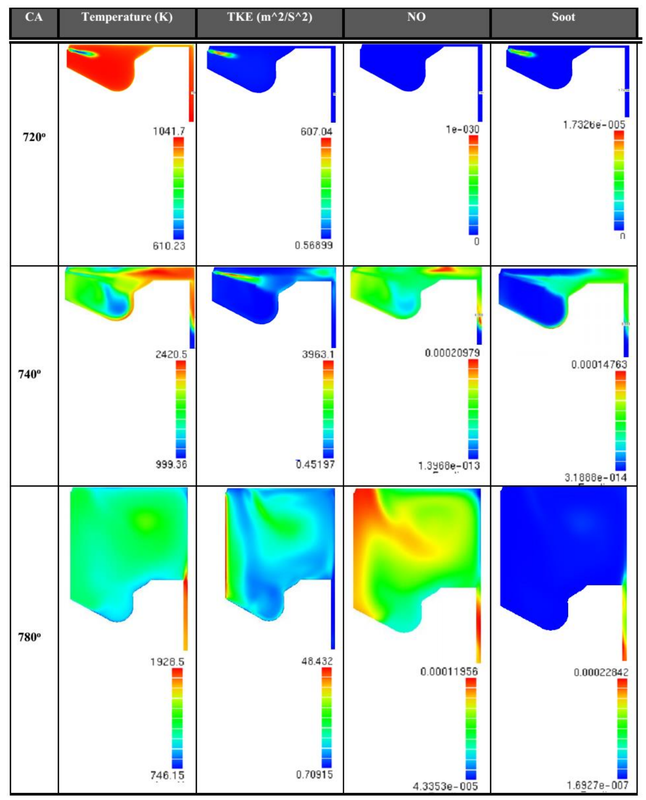


Fig. 18. Distribution of swirl and TKE versus crank angle for three piston bowl geometries under 1400 and 1600 rpm engine speeds

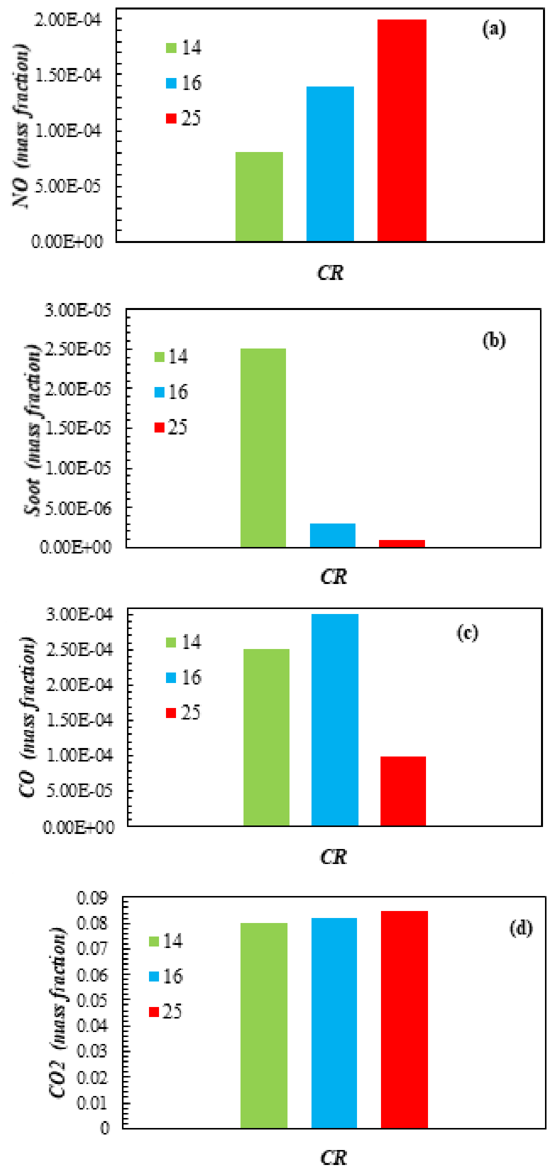


Fig. 19. Influences of changing the CR on emission characteristics under 1400 rpm engine speed (a): NO mass fraction (b): soot mass fraction (c): CO mass fraction (d): CO2 mass fraction

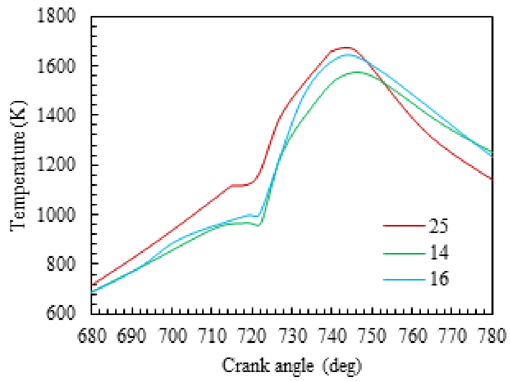


Fig. 20. Cylinder temperature curves under different compression ratios.

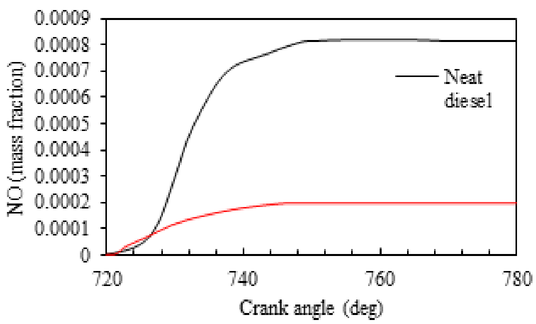


Fig. 21. NO emissions for DME and diesel fuel

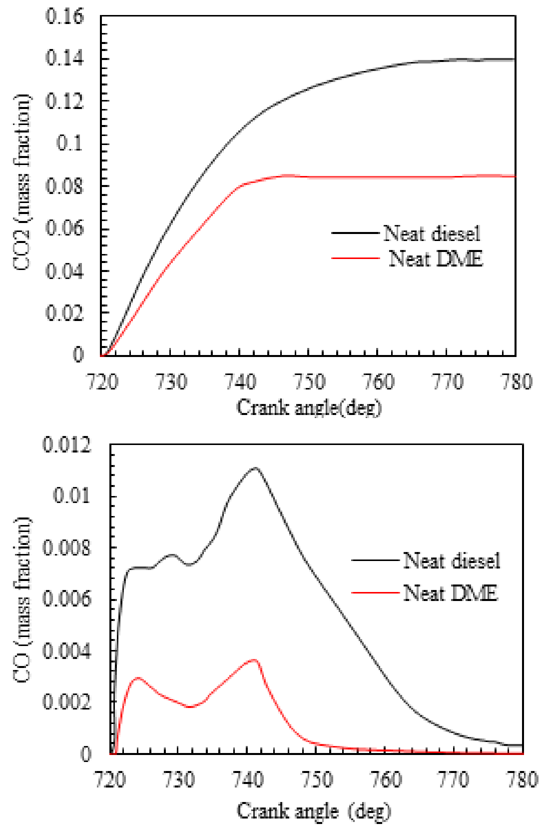


Fig. 22. CO2 and CO emissions for DME and diesel fuel (a): CO2 (b): CO

E. The effects of DME fuel as an alternative to pure diesel

As seen in Fig. 21, by using the DME amount of NO emission is reduced by 75%. The higher NO in diesel operation is reasonable because of its higher heating value, which leads to an increase of the cylinder temperature and so higher NO emission is formed. Moreover, DME operation has shorter ignition delay and fast combustion duration due to its higher cetane number, which results in lower NO emissions [27].

Figs. 22 show CO and CO2 emissions for DME and diesel fuel. As seen, DME operation displayed lower CO emissions compared to diesel-burning engine because the DME is an oxygenate fuel and the oxygen content in the chemical structure of DME fuel prevents formation of CO emissions and as previously mentioned, DME fuel does not have C-C bonds in its chemical structure and soot and CO2 emissions from an engine fueled by DME is also low [27]. As can be seen in Figs.22, the CO2 and CO of the ISM engine fueled by DME are reduced by 44.43% and 8%, respectively, compared to the engine fueled by diesel. Fig. 23 shows that the soot is reduced nearly 20% by fueling the engine with pure DME due to its oxygenate structure and lower heating value of DME compared to diesel fuel. As illustrated, the soot emissions had extreme variation in the case of production of the soot at the range of 720 to 780 crank angles. Using the DME fuel (especially when is obtained from biological sources) in combustion chamber of a HD diesel engine can achieve lower emissions compared to the engine fueled with pure diesel.

Fig. 24 shows a lower equivalence ratio is obtained for DME fuel, which results in improved air-fuel mixing compared to diesel fuel. As seen, oxygen mass fraction in the engine fueled by DME is lower than diesel-burning engine because of better air-fuel mixing, which led to increasing the fuel reaction. In addition, more equivalence ratio is a reason for lower soot emissions for the DME operation [29]. Fig. **??** shows that the diesel-burning engine created a higher temperature compared to the engine fueled by DME in various crank angles. The main factor is the fact that the lower heating value of DME fuel led to a reduction in the temperature after injection of fuel and this case is the main reason of producing higher NO emissions from engine fueled by the diesel. As seen in Fig. 21, for the DME operation almost all the quantity of injected DME was consumed due to having the ER below 1 (higher ER more than 1 means there is too much unburnt fuel) because the DME has a higher cetane number, which leads to advancing the combustion time and lower ER of DME operation is due to proper temperature generated in the engine.

Table 7 shows that engine fueled by the DME has higher thermal efficiency and lower torque and power. In this comparison, all upper mentioned parameters are true for diesel-burning en-

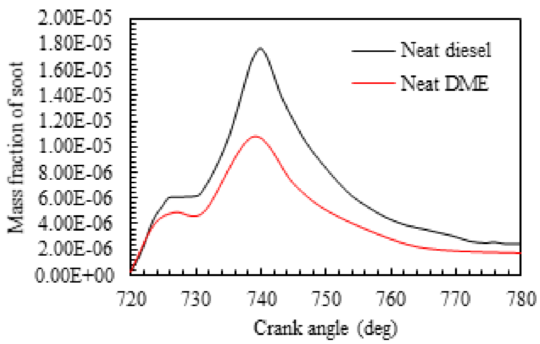


Fig. 23. Soot emissions for DME and diesel fuel

Table 7. Comparison of the power, thermal efficiency and torque between the DME and diesel

	DME	Diesel
Power (kW)	15.8	21.33
Thermal efficiency (%)	39	33
Torque (Nm)	125.75	162.33

gine and the comparison is performed at the same conditions. The main reason of reduction in the power and torque of DME operation is lower heating value of DME fuel compared to the diesel because for both operations all involved variables in simulation were identical and only fuel was changed.

4. CONCLUSION

Combustion and environmental effects of the engine speed and piston bowl geometry on the ISM 370 diesel engine fueled by the DME were numerically investigated. The following conclusions were obtained from the paper discussion:

- The PB2 and PB3 geometries because of having lower piston bowl volume cause lower emissions in comparison to the PB1 and when the engine is equipped with the PB3 or PB2, soot emission is dramatically reduced toward the PB1 by 91% under 1400 rpm engine speed. In addition, reduction of the bowl volume leads to lower CO2 emission.

- The obtained results show that the PB2 and PB3 had lower CO emissions because of producing higher in-cylinder temperature, which leads to more the CO conversion.

- PB3 structure produces lower NO emissions by around 9.3%.

- It was determined that with considering the baseline conditions, the amount of exhaust emissions is greatly reduced by injecting the DME instead of diesel fuel. Accordingly, by using the PB1 and DME fuel in comparison to diesel fuel the NO, soot, CO and CO2 are reduced 75%, 20%, 8%, and 44.43%, respectively.

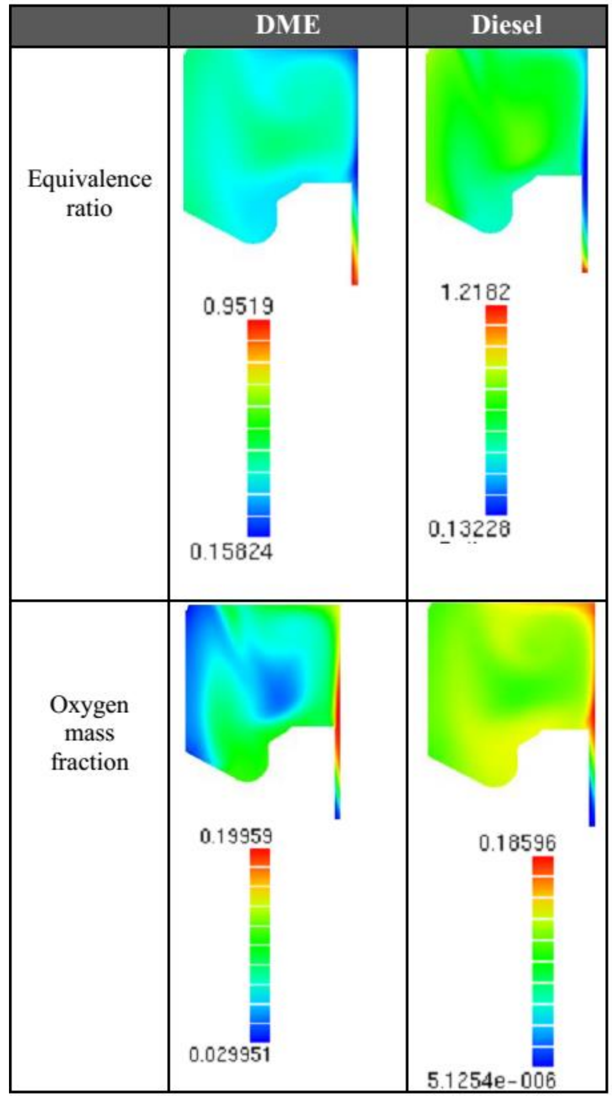


Fig. 24. Equivalence ratio (ER) and oxygen mass fraction distributions almost when exhaust valve is opening for diesel and DME fuel

- It was found that the most important reason of improving the combustion characteristics caused by PB3 geometry is better air-fuel mixing.

- Increasing led to enhancing the mean pressure and temperature in the cylinder. However, the moderate compression ratio, which made the balanced state between emission and combustion characteristics, was found 16 among considered compression ratios.

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