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Improving the combustion and emission characteristics of ISM 370 diesel engine by hydrogen addition and redesigning injection strategy

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ABSTRACT

Hydrogen fuel is the cleanest fuel available and it can be used as an additive in diesel engines. Diesel engines have a variety of advantages such as its power, high thermal efficiency and low fuel costs. There have been extensive studies on the use of hydrogen fuel in diesel engines in recent years. However, the effect of concurrently using gaseous hydrogen fuel and changing injection strategy needs further investigation, especially for the Cummins ISM370 engine. This work considers almost all the functional and emission parameters simultaneously. This procedure can effectively achieve balanced conditions when 6% H_2 (by volume) is injected into the Cummins ISM 370 diesel engine (under different engine speeds). In addition, due to a change in the fuel compound used in the engine, the injection timing and temperature of the engine should be redesigned for better operating. A computational fluid dynamics code is used for simulating the engine. In order to verify the validity of the simulation, the predicted mean pressure and the rate of heat release are compared to the experimental data, and the results are in accordance. The results show that most of the exhaust emissions such as NO, CO, etc. are dramatically reduced by using gaseous hydrogen under various engine speeds. With the addition of 6% H_2 to the engine, the thermal efficiency increases by around 39%, and the NO, soot, CO and CO_2 emissions are reduced by 5%, 75%, 70%, and 30%, respectively, under a 1600 rpm speed. Also, 4 deg BTDC under 2000 rpm is the best injection timing that makes a balance between the exhaust emissions and performance parameters. Moreover, the best injection temperature is 330 K among the three injection temperatures considered in this investigation

1. Introduction

Diesel engines have high thermal efficiency and preferable power performances compared to the spark engines and are widely utilized by automotive manufacturing companies [1]. Furthermore, there are many renewable energy sources such as butanol [2], DME [3], hydrogen [4], etc. which can be used in diesel engines to enhance their advantages. Clean and renewable fuels such as hydrogen are highly regarded by scientists because of their suitable environmental performance. Hydrogen is expected to be a very significant energy source in the near future for reducing pollutant emissions [4]. The use of hydrogen as a supplemental fuel in the combustion chamber can improve the mixture formation and enhance the ignition process [5-7]. The influence of added hydrogen on the combustion performance and tailpipe emissions of many engines such as high-speed direct injection and Volkswagen TDI engines have been investigated [8,9]. In these studies, it has been shown that the emission and performance feedback of different engines varied when hydrogen is added. Since the results are highly dependent on the engine type, it is necessary to investigate each engine individually. In addition, various means exist for the improvement of engine performance and exhaust emissions [10-17]. Injection timing plays an important role in engine

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performance. Soni and Gupta [10] investigated the effects of fuel injection timing on emission and combustion characteristics. They showed that changing the injection timing could vary the start of combustion, and then the in-cylinder pressure and temperature. In addition, they determined that retarding the injection timing led to lower NO emissions and advancing the injection timing caused increasing NO emissions. The brake specific fuel consumption (BSFC) is affected by changing the injection timing, and a lower BSFC means that a smaller amount of fuel is required to make the same power [11]. Zheng et al. [2] studied the Exhaust Gas Recirculation (EGR) effects on the combustion and emissions characteristics of a heavy-duty diesel engine fueled by n-butanol/diesel blend. They showed that by increasing the EGR rate, the amount of threshold level, CO and HC emissions increased but NO emissions were reduced significantly. Gowthaman and Sathiyagnanam [12] investigated the influence of intake air temperature and injection pressure on the performance and emission characteristics of a compression ignition engine fueled by pure diesel. They determined that an increasing temperature of inlet air limited the operation range of the engine because of the production of higher NOx emissions. Saei Moghaddam and Zarringhalam Moghaddam [13] investigated the performance and exhaust emission characteristics of a diesel engine fueled with diesel-nitrogenated additives. They stated that the use of additives decreased viscosity but increased the cetane index. In addition, they found that the smoke emission could be reduced at the maximum torque speed compared to the rated power speed. A change in injection pressure also limited the engine operation because of changes in some emissions such as HC emissions. Besides injection timing, combustion chamber geometry is a significant designing parameter that can change emissions and combustion parameters directly [14-17]. Yang et al. [18] investigated the influence of the addition of H_2 to the ISM370 diesel engine. They showed that the in-cylinder pressure and rate of heat released decreased with increasing of H_2 mass fraction, NO emissions increased, and soot emissions reduced. Moreover, they determined that using 17% H_2 led to a higher peak pressure and an increase in NO emissions. Yang et al. [18] studied the effects of the addition of H_2 only for a constant engine speed. In their work, the emissions of CO, CO_2 and HC and performance parameters like indicated power, thermal efficiency, BSFC, etc. were not considered altogether. In the present work, the ISM370 engine has been designed for both pure diesel fuel and when hydrogen fuel is added. Liu et al. [4] conducted an experimental investigation to evaluate the effects of hydrogen addition on the amount of NO_X emission of the ISM370 diesel engine under different operating conditions. They showed that increasing the addition of H_2 led to more NO_X emissions compared to a diesel-only operation. As mentioned previously, the examination of an emission is not sufficient, and the simultaneous investigation of most emissions and power parameters under various engine speeds is important. The addition of 6% hydrogen, volumetrically, can be a proper amount that responds to the possible economic problems and concerns about the stability of the combustion chamber (an unplanned increase in peak pressure may cause negative side effects on the engine). Furthermore, the ISM 370 engine shows good stability and performance with a 6% H_2 addition under 1200 rpm speed in a similar work [18] and in our study It is expected that the injection timing, injection temperature and engine speed will affect the exhaust emissions amount, fuel economy and engine performance significantly. Thus, various injection timings, injection temperatures and engine speeds have been chosen to evaluate their effects.

2. Simulation

The specifications of the engine considered in this work are presented in Table 1. As seen, a four-hole (0.35 mm in diameter) central injector nozzle was chosen. The movable meshes were designed via AVL FIRE ESE software. Figure 1 shows the details of the considered boundary conditions. The computations were done from the intake valve closed (IVC) to the exhaust valve opened (EVO). At first, the cells numbers were changed from 60,000 to 120000 for the simulated model and 80000 meshes were found suitable following the mesh analysis. The hydrogen fuel entered through the intake port with the presence of air before IVC, and the diesel fuel was injected through a four-hole injector.

Table 1. Characteristics of the ISM370 engine [5]

ISM 370 HD diesel engine	
Number of cylinders	4
Bore	0.125m
Stroke	0.147m
Compression ratio	16:1
IVC	570 ^oCA
EVO	810 ^oCA
Injector hole diameter	0.35mm
Number of injection holes	4

The present paper employed the 3-zones extended coherent flame (ECFM-3Z) combustion model for ignition modelling, and the WAVE breakup model was applied for modeling the created droplets. The coherent flame model (CFM) explains the fuel consumption amount per unit volume by the product of the flame surface density. In this model, the growth of an initial turbulence on a liquid face was related to its wavelength and to other physical parameters. The Dukowicz evaporation model was used. This model can assume a steady droplet temperature. The K-zeta-f model was used for turbulent wall heat transfer modeling because it is preferable for changeable boundaries and highly compressed flows. In the diesel engines, due to the high temperature conditions, the fuel and prompt NO is inconsiderable so the Extended Zeldowich mechanism was considered, which is appropriate for calculating thermal NO. The soot emission was predicted with the Lund Flamelet model. For turbulent combustion behavior, the ECFM combustion model was employed. This model offers two chemical mechanisms for fuel consumption as follows [19]:

$$C_nH_mO_k + \left[n + \frac{m}{4} - \frac{k}{2}\right] O_2 \xrightarrow{\text{yields}} n\,CO_2 + \frac{m}{2} H_2O \quad (1)$$

$$C_nH_mO_k + \left[\frac{n}{2} - \frac{k}{2}\right] O_2 \xrightarrow{\text{yields}} n\,CO + \frac{m}{2} H_2 \quad (2)$$

In formulas (1) and (2), n, m, and k are the number of carbon, hydrogen, and oxygen atoms of the hydrocarbon fuel, respectively. It is assumed that the unburned gas involves three unburned species of O_2, N_2 and H_2. The ECFM-3Z model is applicable for both the auto and spark ignition. In this model, the burnt gas is a combination of 11 species (N, N_2, O, O_2, H, CO, CO_2, H2, NO, OH, and H_2O). The ECFM-3Z model uses the transport equations, which solve for the mean quantities of NO, CO, N_2, etc., and the in-cylinder burnt gases involve the real burnt gases in the mixed region plus a zone of the unmixed fuel and airflow. The fuel considered for the engine is divided in two terms: the fuel that is present in the fresh airflow (y_{fu}^{u}), and the fuel that exists in the burnt gases (y_{fu}^{b}). The formula of $y_{fu} = y_{fu}^{u} + y_{fu}^{b}$ is the mean mass fraction of the fuel injected in the computational cell [19]. In AVL software, the reactions (in the case of emissions) are considered using the Meintjes/Morgan mechanism for burnt gas temperature that can be seen in the following chemical formula (3):

$$\begin{aligned} &N2 \leftrightarrow 2N \quad . \quad O2 \leftrightarrow 2O \\ &H2 \leftrightarrow 2H \quad . \quad O2 + H2 \leftrightarrow 2OH \\ &O2 + 2CO2 \leftrightarrow 2CO2 \\ &O2 + 2H2O \leftrightarrow 4OH \end{aligned} \quad (3)$$

The extended Zeldovich model presents the mechanism of thermal NO formation as follows:

$$\begin{aligned} &N_2 + O \overset{K}{\leftrightarrow} NO + N \\ &N + O2 \overset{k}{\leftrightarrow} NO + O \quad . \quad N + OH \overset{k}{\leftrightarrow} NO + H \end{aligned} \quad (4)$$

The amount of hydrogen added is introduced as the volume of hydrogen divided by the volume of hydrogen-air injected through the intake valve. The ratio of hydrogen can be set as the volume fraction of hydrogen in air-hydrogen at 810 °CA (EVO). The equivalence ratio is the ratio of the stoichiometry air-fuel mixture (AFRs) to the actual term of air-fuel mixture (AFRa). The equivalence ratio is defined in Formula (5):

$$\text{Equivalence ratio} = \frac{AFRs}{AFRa} \quad (5)$$

Figure 1 shows the different parts of the designed geometry and applied boundary conditions [5]. Linear is a variable parameter and depends to the crank angle. At top dead center (TDC), the liner is its minimum length. The swirl ratio can be introduced by a dimensionless parameter calculated to quantify the angular air motion inside the combustion chamber in various piston conditions [20].

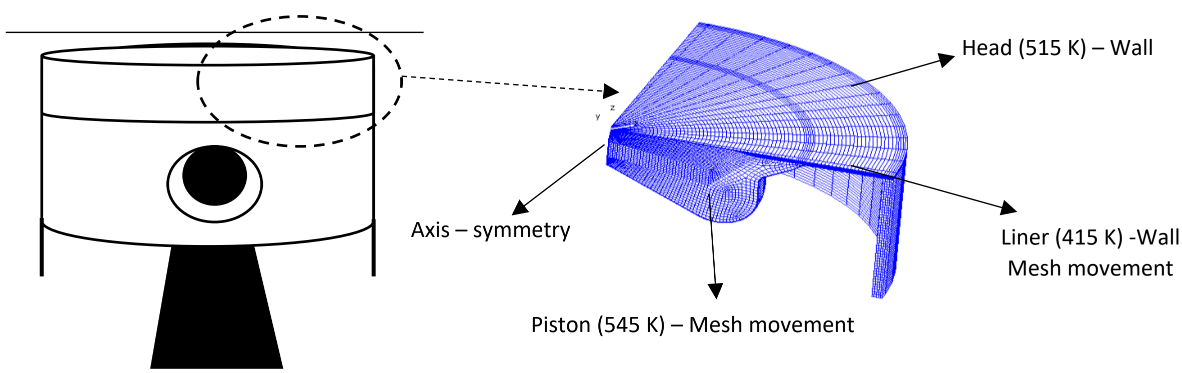


Fig. 1. Piston bowl geometry at TDC position and applied boundary conditions

3. Results and discussion

3.1. Validation

In order to validate the simulation with experimental data, boundary conditions were set as observed in Figure 1; also, Table 2 shows the initial parameters considered to yield suitable validation. In the AVL FIRE software the modules activated were species transport, combustion, emissions and spray.

Table 2. Initial parameters considered for validation part [5]

Input parameters	
Pressure	1900 Kpa
Temperature	386 K
Swirl (in initialization mode)	2880 (1/min)
Injection temperature	330 K
Mass of fuel in computational domain (pure diesel)	35.634 mg
Mass of fuel in computational domain (dual fuel)	24.021 mg

In this paper, for the validation of the numerical simulation, the mean pressure and the rate of heat released traces are compared with the experimental data [21] (pure diesel and dual fuel), and the results show a good agreement. As observed in Figure 2, the computational error is inconsiderable for the mean pressure (error under 3%). However, the computational errors of RHR (rate of heat released) are more than 5%, which are common in calculating the RHR parameter. This occurs because in the process of engine simulation, the heat transfer, thermal radiation, aperture impact, etc. are not considered and researchers [21,22] have approved these errors. However, the predicted curves for the RHR are in general accord with the experimental data and the software performed well. When compared to pure diesel, hydrogen-diesel has a higher peak pressure, longer ignition delay, and shorter combustion duration, as shown in Figure 2. A longer ignition delay is mainly due to a lower cetane number of hydrogen fuel. The EGR rate has greatly effects combustion, so a constant value of the EGR rate is considered for whole simulations.

3.2. Engine performance

The addition of hydrogen varied the fuel distribution, combustion process and amount of some parameters such as pressure and consequently temperature, which led to various amounts of tailpipe emissions. Figures 3 and 4 show the CO and CO_2 emissions under various speeds. As seen, faster engine speed led to lower CO_2 and higher CO production for both fuel types (diesel and diesel-H_2). The high engine speeds reduced the in-cylinder temperature, and this led to an increase in more incomplete fuel reaction and higher production of CO emissions; moreover, poor combining can result in more exhaust CO emissions.

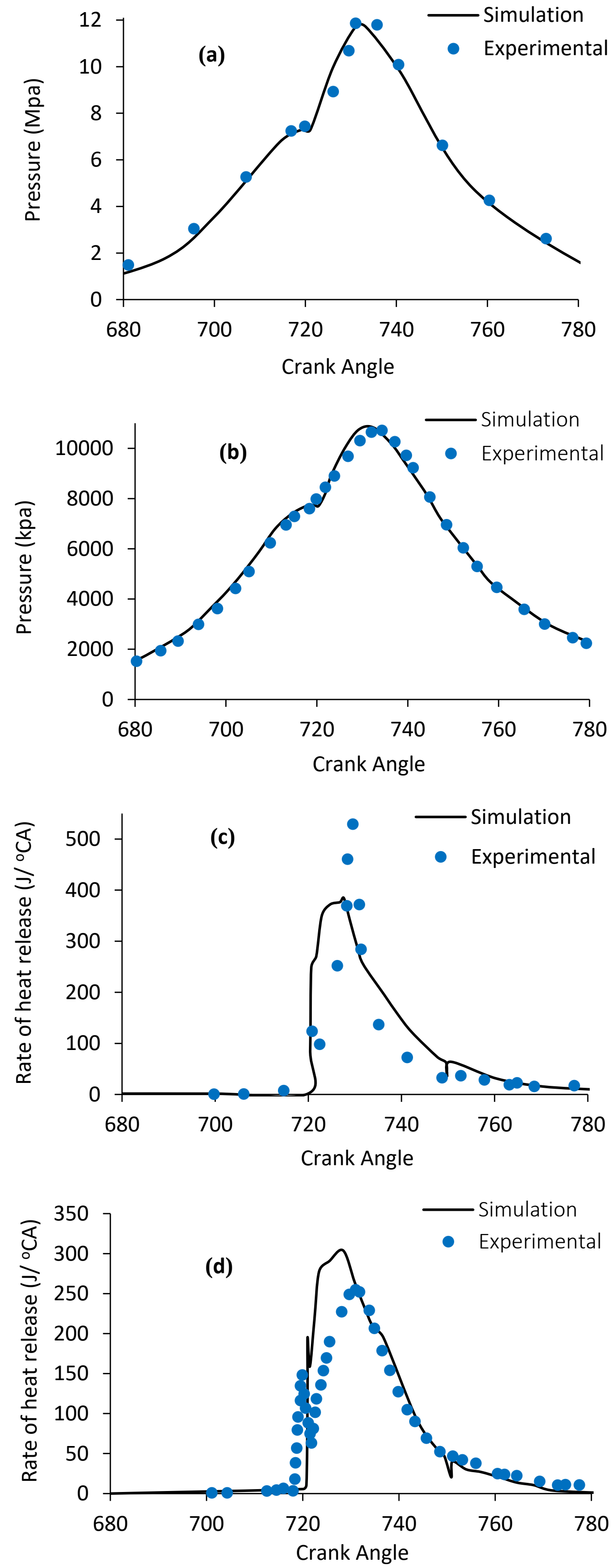


Fig. 2. Comparison of mean pressure and rate of heat released between experimental [18] and predicted data versus crank angle. (a) and (c): diesel + hydrogen (6%), (b) and (d): pure diesel

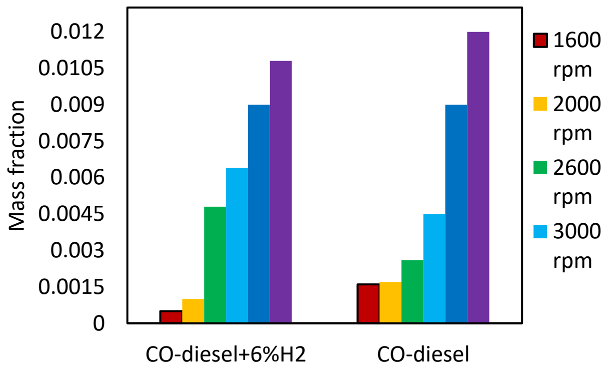


Fig. 3. Comparison of CO emissions under six engine speeds between pure diesel and dual fuel

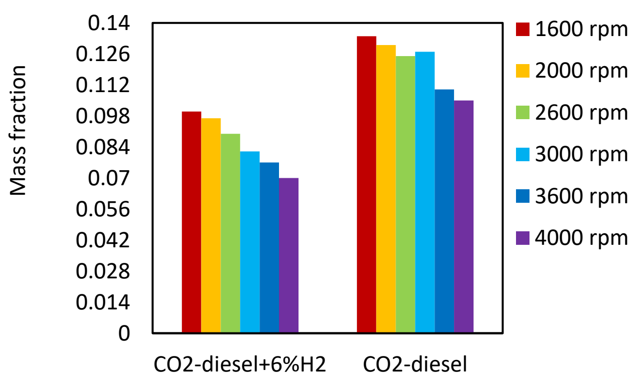


Fig. 4. Comparison of CO_2 emissions between pure diesel and dual fuel under six engine speeds

At low temperatures, N_2 exists as a very steady condition of the diatomic molecule. When the temperature is high in the combustion chamber, diatomic nitrogen converts into mono nitrogen, and NO is formed due to the existence of N and oxygen [10]. Figure 5 illustrates the comparisons between the calculated NO under various engine speeds. The NO emissions decrease by increasing engine speed because of the reduced residence time for both of the considered fuels. When the engine speed is reduced from 1200 to 1600 rpm, the tailpipe NO mass fraction is reduced by 6%. As denoted in Figure 6, the engine fueled by pure diesel produces a higher temperature after a 760 crank angle compared to the dual fuel engine. This high temperature zone leads to more NO emissions, according to the extended Zeldovich model. It should be noted that H_2 has a higher adiabatic flame temperature than diesel, according to thermal NOx formation (Zeldovich mechanism). The NO emissions should increase when H_2 is supplied into the engine (due to higher temperature areas), but in the simulation process, to make the same load around 70%, the amount of diesel fuel mass in the computational domain decreases from 35.634 to 24.021 mg [5]. This can be a reason for reduced NO emissions.

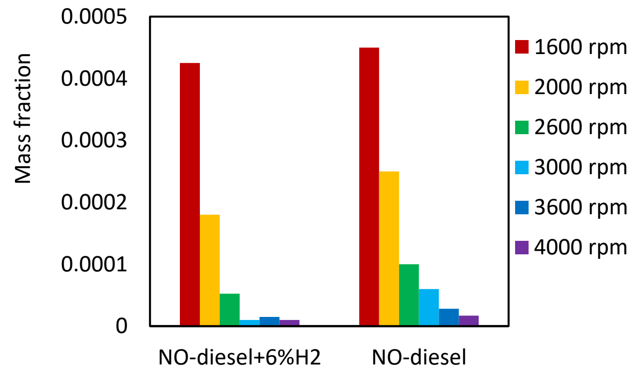


Fig. 5. Comparison of NO emissions between pure diesel and dual fuel under six engine speeds

As shown in Figure 6, the mean oxygen content for a diesel-burning engine is lower than the dual fuel engine at various crank angles under a constant engine speed. However, the temperature caused by pure diesel combustion is higher than dual fuel (due to reducing the diesel fuel mass in the computational domain from 35.623 to 24.021 mg to meet a 70% load [5]) from 720 to 760 CAs. This leads to higher NO emissions in the tailpipe of the diesel-burning engine because of decreasing injected diesel fuel mass in the dual fuel operation.

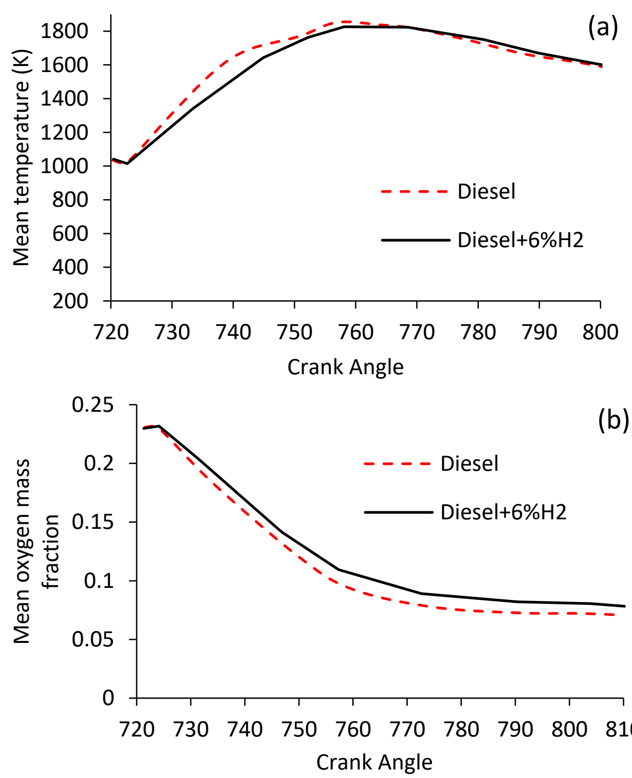


Fig. 6. Mean O_2 mass fraction and temperature traces for diesel and dual fuel under 2000 rpm engine speed

As can be seen in Figure 7, by increasing engine speed, the soot mass fraction dramatically increases. Nevertheless, when the engine works with dual fuel, the amount of exhaust soot is reduced compared to the diesel fuel under various engine speeds. The lower soot formation of the dual fuel can be related to the equivalence ratio [19].

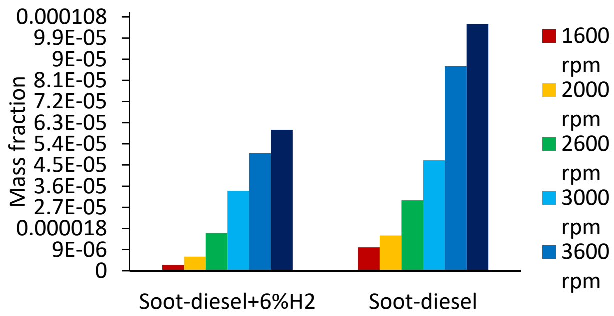


Fig. 7. Comparison of soot emissions under six engine speeds between pure diesel and dual fuel

When the equivalence ratio (ER) is below 1, there is too little airflow (stoichiometric mixture); an ER above 1 shows that there is too much unburnt fuel and not all the injected fuel can be utilized, which leads to incomplete combustion [19]. Table 3 shows a comparison of in-cylinder equivalence ratio distribution for diesel and dual fuel. As seen, the dual fuel has a smaller equivalence ratio in different crank angles, which means better equivalence ratio distribution. This can be explained by the higher oxygen content of the dual fuel, which leads to a lower equivalence ratio, as shown in Table 3. Furthermore, the higher heat from the evaporation of the hydrogen fuel leads to a lower ambient temperature near the spray zone before fuel ignition around the TDC point, which improves the air entrainment in the injected fuel into the cylinder. Moreover, the longer the ignition delay due to a lower cetane number of dual fuel makes more time to air-fuel mixing. Hydrogen-diesel dual fuel has a lower viscosity compared with pure diesel, which cause better spray atomization, followed by air-fuel mixing and the spray evaporation. The lower CO formation of hydrogen-diesel dual fuel can be caused because its higher temperature area. Lower equivalence ratio is because of advancing combustion stage, due to high temperature of gaseous air-fuel and oxygen content are both lead to the process of CO converts to CO_2. Moreover, a faster burning rate for the hydrogen-diesel fuel leads to a shorter combustion phase. Therefore, the final CO content in the cylinder of the dual fuel is greater than the pure diesel. A comparison of CO emissions between the fuels can be observed in Figure 1. These effects have been addressed in reference [12].

Table 3. Comparison of equivalence ratio distribution for pure diesel and hydrogen-diesel dual fuel under 2000 rpm engine speed

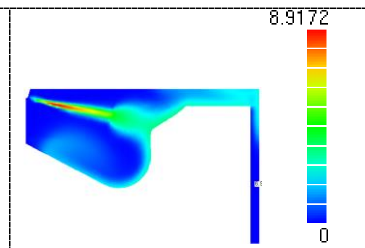
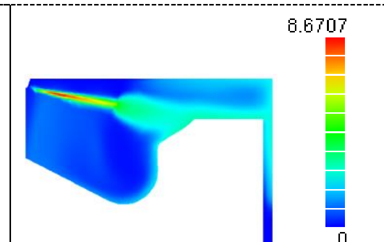
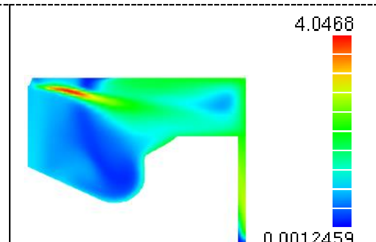
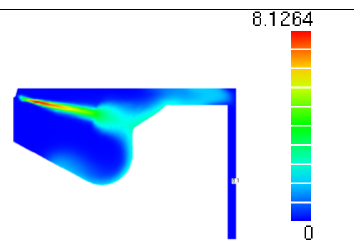
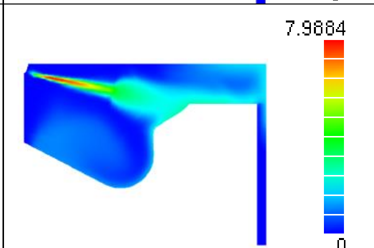
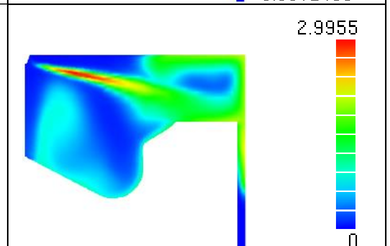
Equivalence ratio distribution			
Diesel			
Hydrogen -diesel			
	735 CA	745 CA	755 CA

Table 4 shows the distributions of oxygen, temperature, and equivalence ratio for the dual fuel engine at 730, 735 and EVO °CAs under a 2000 rpm engine speed. As seen, the equivalence ratio is under 1 and most of the injected diesel fuel has been burnt with $H_{2\ presence}$ (at EVO). As observed in Table 4, the ignition of the dual fuel occurs at both sides due to the piston wall and spray interaction. The ignition in the two sides of the piston can be explained by the lower cetane number and the higher heat of evaporation of the dual fuel, which produces more time for spray penetration into the cylinder. Furthermore, the lower density of the dual fuel creates in higher injection velocity than pure diesel in same injection conditions. Therefore, the spray penetration length of the hydrogen-diesel dual fuel is longer than the pure diesel under the same conditions. These cases have also been seen in other research [12].

Table 4. Oxygen, temperature and equivalence ratio distributions for dual fuel engine under 2000 rpm engine speed

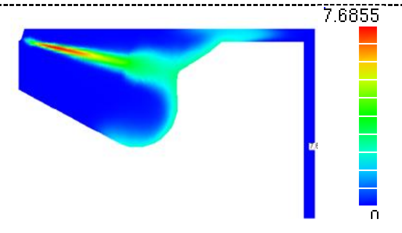
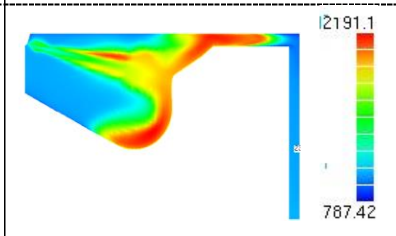
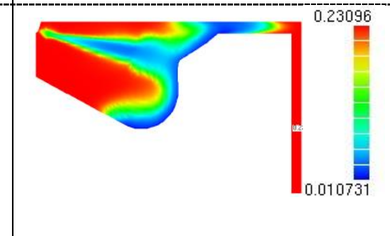
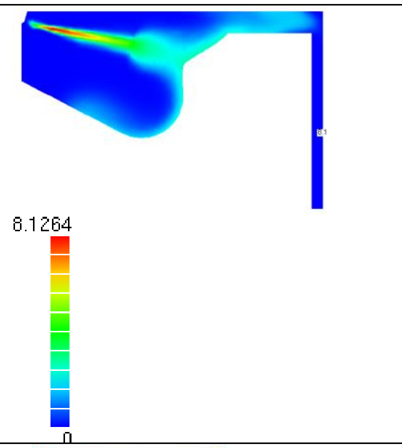
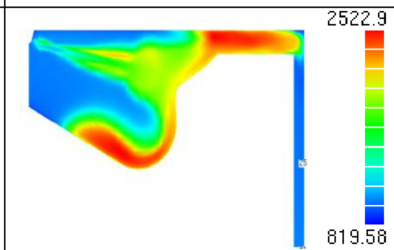
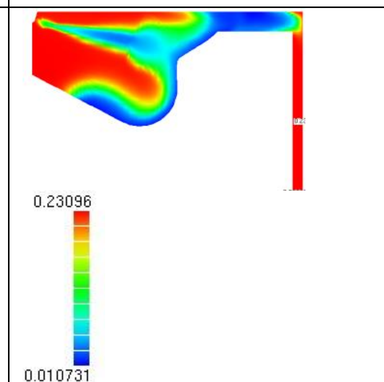
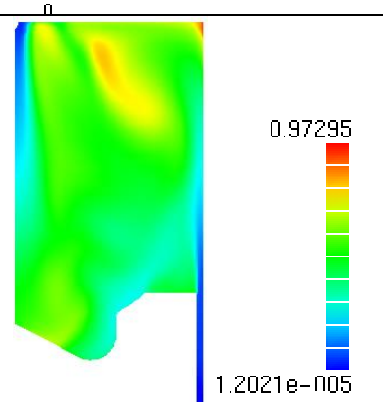
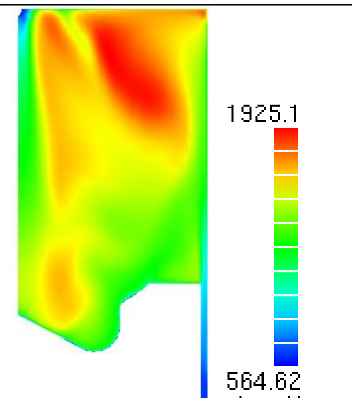
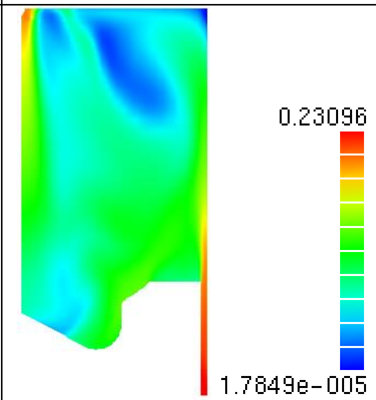
	Equivalence ratio	Temperature (K)	Oxygen (mass fraction)
730 °CA (Dual fuel)			
735 °CA (Dual fuel)			
EVO (810 °CA) (Dual fuel)			

Figure 8 shows that the indicated thermal efficiency is enhanced with increasing the engine speed to around 2000 rpm. Then, it drops after this engine speed and attains an optimum value. Increasing the engine speed enhances the turbulence levels significantly, which makes for better air-fuel mixing and improvement of the combustion process. In the optimum condition, increasing the engine speed causes a reduction in the efficiency only by limitation of the breathing ability. Figure 9 denotes a lower indicated power with increasing engine speed above 2600 rpm for both fuels. However, the combustion with the hydrogen supplemental fuel leads to better indicated thermal efficiency when compared to the pure diesel fuel. As seen, the thermal efficiency increases by 25% under a 2000 rpm engine speed, whereas the indicated power is reduced approximately 7% due to reducing the injected diesel fuel mass to meet a 70% load. However, the reduction of 7% in the indicated power is inconsiderable regarding the thermal efficiency improvement 27%, approximately.

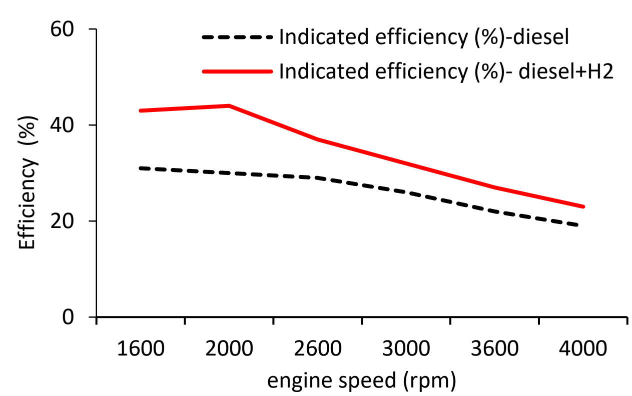


Fig. 8. Efficiency traces under six engine speeds and 70% load

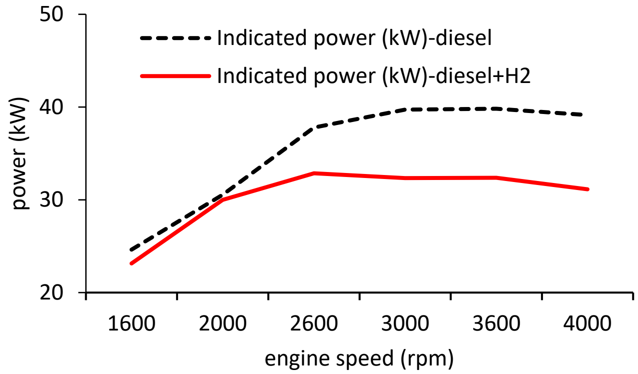


Fig. 9. Indicated power traces under six engine speeds and 70% load

3.3. Injection timing

The NO emissions also depend on the pressure, temperature, and duration of combustion time. The burnt gas takes form in the combustion process, increasingly, which are compressed in near the peak pressure after fuel combustion. This cause burnt gases stand in high temperatures for widely range of engine operation so this can lead to produce more NO emissions [23]. Retarding the injection timing creates a lower temperature and causes a reduction in the NO emissions, and the consumption of fuel increases in the combustion chamber due to the later ignition [10]. Our studies have shown that increasing the temperature and having enough oxygen makes the conversions of CO and soot emissions easier. As known, advanced injection timing increases the in-cylinder temperature. It should be noted that with addition of H_2, the amount of oxygen and nitrogen content in the intake air is reduced and lower oxygen content leads to lower NO and CO_2 emissions as well as higher soot and CO formation. In this section, the effects of injection timing on emissions and engine performance are numerically evaluated. As seen in Figure 10, the injection timing of 6 deg BTDC is not suggested since it generates a large amount of NO emissions. The best injection timing with considering a balance between CO, CO_2, NO, and soot emissions is 3 deg BTDC.

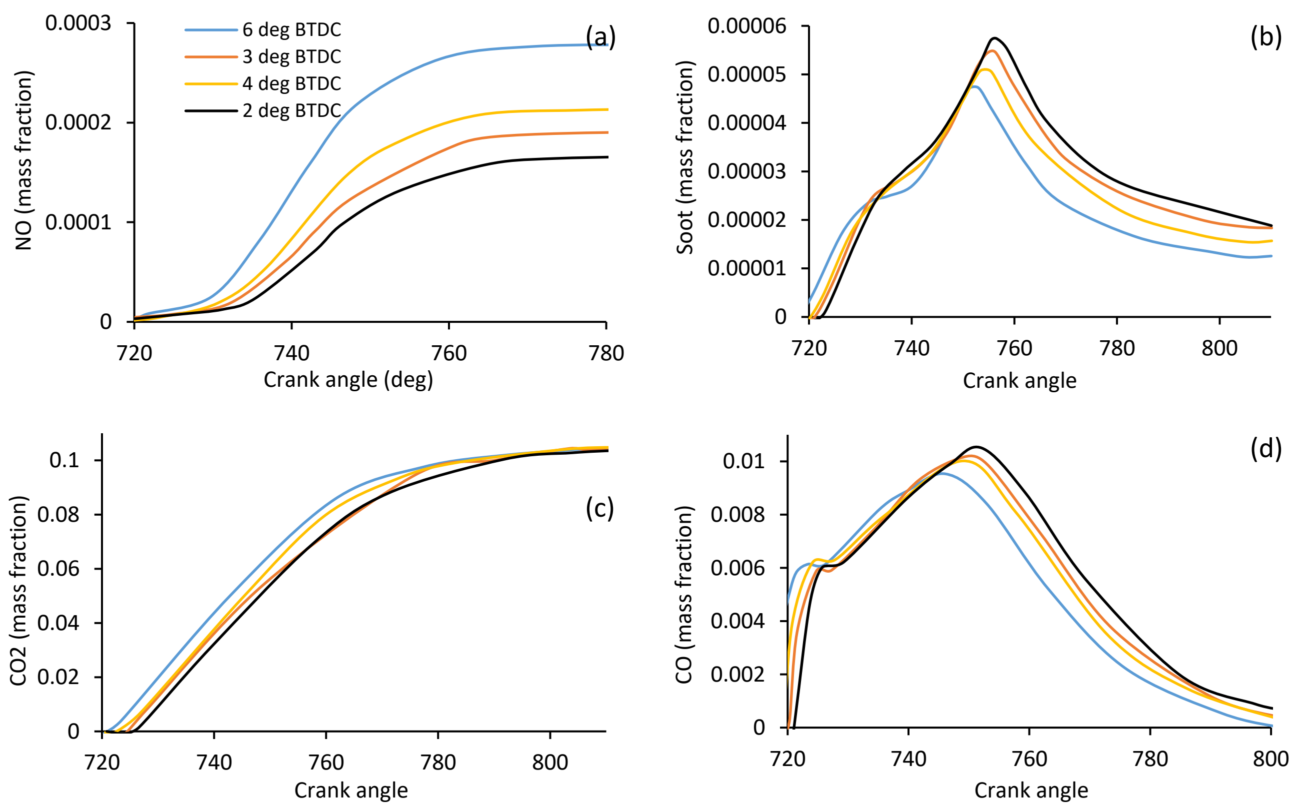
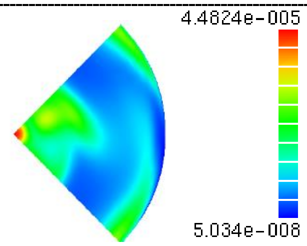
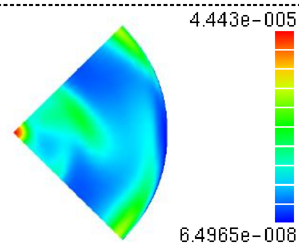
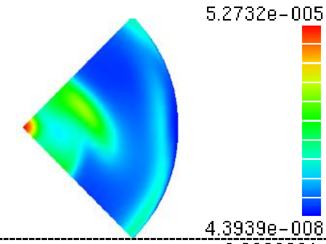
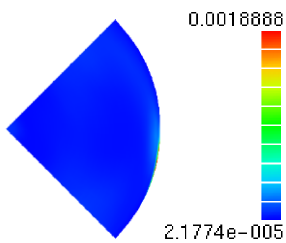
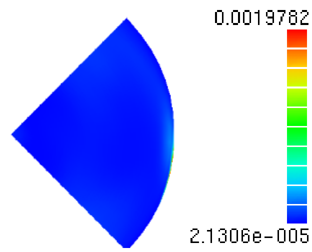
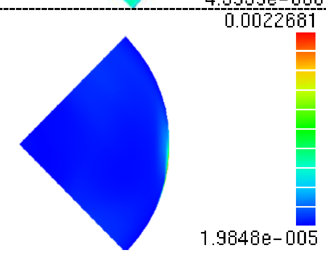


Fig. 10. Amount of exhaust emissions at four injection timings under 2000 rpm speed

One of the advantages of adding hydrogen to diesel fuel is that it does not produce major emissions such as HC, CO, SO_2, soot, PM, lead, and other pollutant gases. Hydrogen is a carbon-free fuel and only water is released from complete hydrogen combustion in the cylinder when the engine runs. Moreover, using the proper injection timing for the dual fuel engine can further help to reduce pollutant gases. Table 5 denotes the mass fraction of the residual unburned hydrogen and diesel fuel when the exhaust valve is open. As seen, the mass fraction of unburned diesel and hydrogen fuels does not change with advancing the injection timing. It can be concluded that changing the injection timing does not significantly affect the hydrocarbon emissions.

Table 5. Comparison of unburnt hydrogen and diesel fuel under 2000 rpm engine speed

Injection timing	**3 deg BTDC**	**4 deg BTDC**	**6 deg BTDC**
Mass fraction of Diesel			
Mass fraction of H_2			

3.3.1. Infuences of injection timing on BSFC

The brake specific fuel consumption (BSFC) of a diesel engine shows the relation between the fuel injection system and fuel specifications such as specific gravity and heating value. A lower BSFC means that a smaller fuel mass is needed to make the same amount of power (fuel economy) [23]**.** Figure 11 illustrates the variation of BSFC with injection timing under a constant engine speed (2000 rpm) for dual fuel. As indicated, with an advancing injection timing to 6 deg BTDC, the BSFC is in the lower value. However, with regard to the observance of moderate emissions and BSFC values, 4 deg BTDC is the best injection timing for the dual fuel engine in comparison to pure diesel. The obtained results show that the dual fuel engine has a lower BSFC under a constant injection timing, and the BSFC is reduced by 30% at the injection timing of 2 deg BTDC. Thus, in a constant condition, the addition of 6% hydrogen can reduce the rate of fuel consumption and yield more power.

3.3.2. Influences of Injection timing on indicated fuel consumption (IFC)

The specific fuel consumption (IFC) is supported on the torque caused by the engine concerning the mass flow of fuel achieved to the engine. It is calculated after all engine process losses are BSFC and calculating specific fuel consumption supported on the pressure of combustion chamber is IFC. As seen in Figure 12, it is obvious that when the injection timing is set at 4 deg BTDC, the amount of fuel consumption is reduced by around 2.5% in comparison to baseline injection timing (2 deg BTDC).

3.3.3. Influences of Injection timing on IMPE

The mean effective pressure (IMEP) is a value relating to the action of an engine (reciprocating) and is a noteworthy measure of an engine's displacement to process that is independent of diesel engine capacity. It is found that the IMEP is reduced by retarding injection timing, and as the timing advances, it increases. Investigations [23] show that in late injection timing, the in-cylinder mean pressure and temperature are lower than the baseline condition, and this can lead to lower peak pressure and IMEP. As seen in Fig 13, at a 6 deg BTDC, the injection timing IMEP increases about 3% in comparison to baseline injection timing.

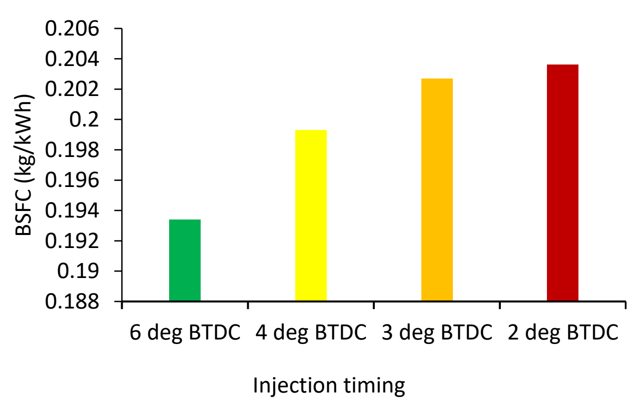


Fig. 11. Influences of injection timing on the BSFC

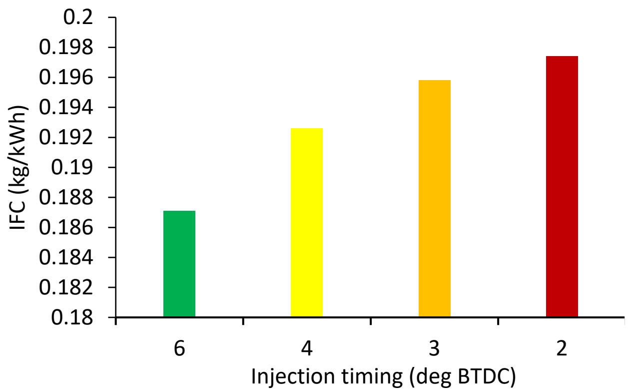


Fig. 12. Influences of injection timing on the IFC

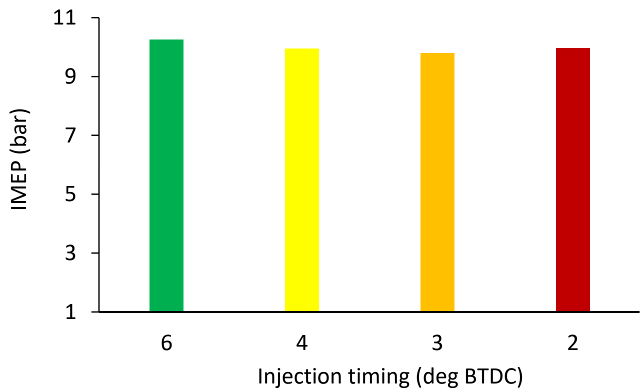


Fig. 13. Effects of injection timing on the IMEP

3.4. Injection Temperature

The injection temperature and pressure have direct effects on the exhaust emissions. In this paper, three injection temperatures (320, 330, and 350 K) are selected and investigated. Choosing an injection temperature under 320 K prevents combustion, and it is not suggested. The results show that an injection temperature from 320K to 330K has no effect on the CO and CO_2 emissions. As can be seen in Figure 14**,** the best injection temperature with regard to a balanced state is 330K for a diesel-H_2 dual fuel engine with an engine speed under 2000. Also, CO, CO_2, HC emissions are in a proper balance when the injection temperature is 330 K.

4. Conclusions

In the present study, an ISM370 heavy-duty engine fueled by pure diesel and hydrogen-diesel was simulated and evaluated under various engine speeds (1600, 2000, 2600, 3000, 36000, and 4000 rpm). The effects of the changes in the injection timing and temperature on the emissions and performance parameters were investigated. The results of the numerical simulation showed that the addition of 6% H_2 reduced the NO, soot, CO, and CO_2 emissions to about 5%, 75%, 70%, and 30%, respectively, under a 1600 engine speed. This supplemental fuel resulted in higher efficiency (more than 20%) under the same speed in comparison to pure diesel. It was found that advanced injection timing led to lower fuel consumption and higher mean effective pressure. The best injection timing in regard to the balanced state between the exhaust emissions (NO, CO, CO_2, and soot) and the engine performance parameters (BSFC, efficiency, etc.) was 4 deg BTDC under a 2000 rpm engine speed when the ISM 370 engine ran with hydrogen (6%) - diesel dual fuel. It was determined that the injection temperature of 320K to 350K had an effect on NO and soot emissions, and a 330K injection temperature was preferable for balancing the soot and NO emissions.

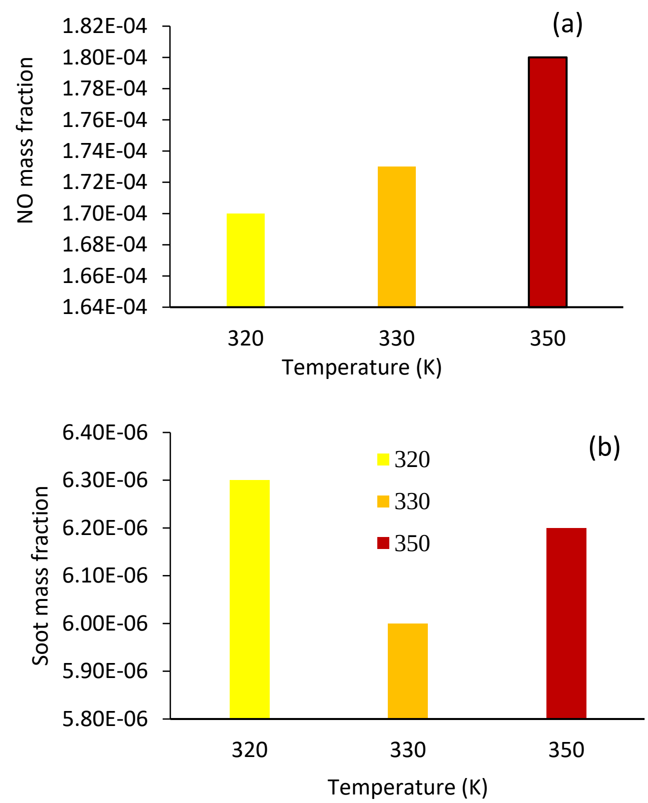


Fig. 14. Effects of injection temperature on the tailpipe soot and NO emissions

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