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Numerical Analysis of Combustion Characteristics and Emission of Dual and Tri-Fuel Diesel Engine under Two Engine Speeds

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Abstract

The numerical simulations were performed on a single-cylinder diesel engine that operates using the direct injection technique. In this study, a two-dimensional CFD code was used in order to evaluate the emissions and combustion characteristics of a dual-fuel operation (diesel-H2, diesel-NG), tri-fuel operation (diesel-NG-H2), and normal operation of a diesel engine under different engine speeds. The percentage of diesel fuel was 100% and 50% with the remaining fraction of different mixtures of NG-H2 (100%–0%, 50%–50%, and 0%–100%). The results showed an increase in peak temperature and pressure when gaseous fuels were added and influenced directly by H2 percentage. With diesel-H2, peak in-cylinder temperature and pressure are found. The higher temperature of combustion as a result of a rising fraction of H2 in the fuel blend proves the formation of NO, whereas increasing the fraction of diesel fuel limits the increase of NO emission, and rising percentage of H2 linearly increases NO. CO emission is mostly effected by NG fraction, but the rising fraction of H2 decreases CO closer to normal diesel operation. The mixture of 50% NG and 50% H2 produces optimum stability between combustion characteristics and emissions. However, high diesel fraction content is preferable for sustaining low combustion temperature, high thermal efficiency by avoiding excessive heat loss, reduces ignition delay, and peak in-cylinder pressure.

Keywords: Diesel engine; Dual fuel engine; NG; Hydrogen;

Abbreviations

2D	2-Dimensional	CH_4	Methane
AFR_{st}	Stoichiometric Air to Fuel Ratio	LPG	Liquefied Petroleum Gas
BDC	Bottom Dead Center	CO_2	Carbon Dioxide
CFD	Computational Fluid Dynamics	CNG	Compressed Natural Gas
CAD	Crank Angle Degree	CO	Carbon Monoxide
EVC	Exhaust Valve Close	PM	Particulate Matter
HC	Hydrocarbons	H_2	Hydrogen
NO	Nitric Oxide	$\dot{m}$	Mass Flow Rate
TDC	Top Dead Center	X	Mass Fraction of Fuels
NOx	Nitrogen Oxides	NG	Natural Gas
IVC	Intake Valve Close	IVO	Intake Valve Open

1. Introduction

Pollution is greatly felt in large urban cities where heavy-duty engines and cars largely contribute to total pollutant emissions. Most of the energy consumed globally is spent on automobiles, usually supplied by diesel and gasoline because of crude oil, a source of energy with limited resources. Rigorous oil consumption has increased its risk of rapid depletion. Oil prices become unsteady year by year [1]. Thus, scientists and researchers have shifted their focus to finding alternatives to the petroleum fuels that we normally utilize. Most of these research efforts aim to achieve high energy efficiency and to lower hazardous emissions [2]. The use of gaseous fuels for internal combustion engines has long been suggested to potentially maintain engine efficiency and performance whilst reducing emissions [3]. Current studies on various alternative fuels for diesel engines have been conducted to reduce diesel fuel consumption as well as particulate and nitrogen oxide NO emissions [4]. Natural gas (NG) has been suggested as one of the most suitable alternative fuels that work not only as a petroleum fuel replacement, but also as an alternative that can lower smoke, NO, and particulate matter emissions. Moreover, high resistance to knock permits engines to work at a high compression ratio, producing high thermal efficiency. Modification of any diesel or gasoline engine to dual fuel using NG operation involves slight mechanical changes as shown in the steady increase of vehicles that use NG resulting from alteration of NG engines [5, 6]. Revisions on the features operational of diesel engine dual-fuel (diesel-NG) single-cylinder revealed the dual-fuel mode (diesel-NG) consequences in longer ignition delay compared with normal diesel engines [7, 8]. In-cylinder pressure is enhanced at high load, but they are lowered at part loads as the content of NG increases. Hence lean mixture, long ignition delay, and low flame speed at part loads. The use of dual fuel (diesel-NG) engine revealed a

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significant increase of PM, HC, and CO emissions as a result of incomplete combustion, poor mixture, and increasing the zone of flame extinction at part load [9]. Conversely, increased quantity of diesel fuel resulting in a slight reduction of NO production and can reduce CO emission via the intake of pre-heated air [10]. When a (diesel-NG) dual-fuel engine works at several engine speeds and compression ratio, it decreases the emission of CO, and thermal efficiency increases at high engine speed and compression ratio, nevertheless, poor thermal efficiency appeared at low engine load [11]. Additionally, H_2 was suggested as another good alternative fuel for internal combustion engines hence, H_2 has a high-octane number, and high auto-ignition temperature exhibition that H_2 is a more appropriate fuel for Otto engines than diesel engines. H_2 can improve engine efficiency whilst reducing emissions. It is odorless, non-toxic and can be combusted completely. It is considered a perfect medium for energy storage that can be sourced from fossil and non-fossil sources [12]. However, mixing H_2 with other fuels appears useful to achieve its benefits. An examination of the combustion behavior of a diesel-H_2 engine showed optimum peak in-cylinder pressure improvement at a 70% load that is essential for durable and safe engine performance [13]. Combustion efficiency is noticeably low with high H_2 fraction. H_2 combustion efficiency depends on engine load. Hence, when operating under high load states, H_2 must be added to achieve high conversion efficiencies for H_2 and diesel fuels. As well as, increase the stability of the engine and decreased unburned hydrogen. conversely, hydrogen's rapid burning rate led to the reduced ignition energy and increased diffusivity that made the combustion unstable at part load [14]. When H_2 fraction is increased, HC/CO/CO_2/PM emissions decrease almost linearly. This behavior indicates that decreases in particle and carbon-based gaseous emissions are affected by the amount of H_2 being added [12,15]. However, under low to middle load states, NO emission decreases. At high load state, NO emission increases as a result of H_2's fast-burning level that results in high combustion temperatures and better NO formation. Besides, engine efficiency is decided by its speed, load, and the H_2 fraction [16]. Increasing H_2 fraction increases the consumption of fuel whilst decreasing thermal efficiency [17]. The processes of ignition of the spark-ignition engine with a mixture of NG-H_2 as fuel are widely examined, and sure limitations related to the efficiency and removed emissions [18]. Ignition delay decrease has been attained with alteration of partial O_2 pressure in the air [19], which influences the combustion of normal diesel. H_2 is used to improve the rate of combustion and expand the limit of NG lean [20]. An increasing fraction of H_2 linearly in an NG-H_2 engine increases the mixture's laminar flame speed [21]. The benefit of H_2 mixing was found to reduce ignition delay as well [22].

The works mentioned above only conducted studies on the use of either NG or H_2 as a secondary fuel. Given its lean operational capacity, H_2 can help maintain efficient and stable combustion. It also produces low amounts of hydrocarbon and greenhouse gas [23]. Thus, an engine can perform better if H_2 is added to NG to create a secondary fuel. Lata et al. [24,25] recently conducted several experimental and theoretical investigations to evaluate the performance of a dual-fuel engine that used a mixture of LPG-H_2 as the main fuel and diesel fuel as pilot fuel. These studies revealed that efficiency could be improved at low load condition states in a dual-fuel operation when H_2 and LPG are mixed to serve as the secondary fuel. Thus, NG and H_2 complement each other in flame stability, emissions, and heating value. NG mixed with H2 may be useful in reducing CO2/HC emissions and extending the flammability range. The addition of H_2 to NG could result in a stable flame and improve the fuel's volumetric burning velocity. Therefore, NG and H_2 mixture could achieve high energy and low emission levels [26]. Nevertheless, the presence of NG decreases the combustion temperature of H_2 and suppresses the emission of NO. Hence, NG has low flame propagation speed, and narrow flammability can make combustion of H_2 smoother and steadier and can thus help prevent insufficient combustion [25]. This study will compare the emission features and combustion characteristics of normal diesel fuel, diesel-NG, diesel-H_2 dual fuel, and diesel-NG-H_2 tri-fuel engine under different gas substations and different engine speeds.

2. Studied Engine

The engine specs [27] and gaseous fuel properties are presented in Tables 1 and 2, respectively. In the current study, the engine was operated under diesel engine, dual and tri-fuel modes under two engine speeds 1500, and 2000 rpm. Additionally, at atmospheric pressure, intake temperature (298 K) and torque (20,18 Nm) were normally taken. Natural gas was presumed to be 100% methane according to the literature [28-29]. Diesel-NG and diesel-H2, under the dual-fuel mode, and diesel-NG-H2 under tri-fuel mode were selected in this study. Thus, the diesel quantity was 50% of the total fuel and the other 50% was substituted by hydrogen and methane. The input power is utilized to be similar for diesel, dual, and tri-operations since the engine we are working under the same processes. Adding the gaseous fuel into a diesel engine demands different techniques in order to identify the dual or tri-fuel operation. These techniques include the carburetion, direct injection, continuous manifold induction, and timing-controlled manifold/port injection. In this study, the diesel was directly injected into the cylinder, and the gaseous fuel was noticeably aspirated into the intake port in combination with air. Hence, the gaseous fuels were recognized as a mass fraction of air in FLUENT. The fresh air in the cylinder consisted of N2 and O2 with a mass fraction of 76.8% and 23.2% respectively, then to present the mass fraction of air, methane, and hydrogen, the subsequent expression is utilized:

$$X_{h2=}\ \dot{m}_{h2}/\dot{m}_{h2} + \dot{m}_{NG} + \dot{m}_{air} \quad (1)$$

Where, $\dot{m}$ was the gaseous mass flow rate and $\dot{m}_{air}$ can calculate by:

$$\dot{m_{air}} = \lambda * AFR_{Dst} * \dot{m}_D + \lambda * AFR_{st.gaseous} * \dot{m}_{gaseous} \quad (2)$$

Where λ was the exceeding air and AFR_{Dst} matching to the stoichiometric air/fuel ratio for diesel (by mass), and AFR $_{st.gaseous}$ matching to the stoichiometric air/fuel ratio (by mass) for gaseous fuels which can be achieved from Table 2.

Table 1. Engine specifications [37]

Mode	Lister–Petter TS1
General details	Single cylinder, 4-stroke, air-cooled, direct injection, compression Ignition
Bore _ stroke	95.3 mm * 88.9 mm
Connecting rod length	165.3 mm
Compression ratio	18:1
Fuel injection timing	13º BTDC
Fuel injection pressure	250 bar
Rated power output	4.5 kW at1500 rpm
Orifices _ diameter	4 * 0.25 mm
Piston type	Cylindrical bowl (diameter: 45 mm and depth: 15 mm)
IVO	36º CA before TDC
IVC	69º CA after BDC
EVO	76º CA before BDC
EVC	32º CA after TDC

Table 2. Properties of various gaseous fuels (20 -C, 1 bar)

	Diesel-NG	NG50-$H_2$50	Diesel-H_2
Density/(kg/m3)	0.65	0.37	0.0837
LHV/(MJ/kg)	50.02	57.79	119.93
Stoichiometric AFR/(kg air/kg fuel)	17.25	25.82	34.39
Flammability limits/(Vol.% in air)	5.0-15.0	---	4.0-75.0
Flammability limits (AFR)	11.2-37.7		5.1-369.9
Thermal conductivity (10^{-2}W/mk)	2.42		4.97
Laminar burning velocity/(m/s)	0.27	---	28
Auto ignition temperature/(K)	813	---	858

3. Computational Models and Numerical Setup

CFD simulations were implemented on a diesel engine, single-cylinder, and direct injection. The GAMBIT software is utilized to generate the entire computational domain of the engine which includes only part of expansion, strokes and compression were also considered. The computational domain consists of the geometry of the combustion chamber without any ports or valves. Commercial fluid dynamics software FLUENT model 15 is used to manage governing equations and post-processing findings. The model of RNG k-ε was implemented for physical modeling according to [30,31]. The program depends on the method of pressure correction and the PISO algorithm is used. Furthermore, applied for energy, turbulence, and momentum equations, the second-order upwind differencing scheme based on Abdullah et al. [29] then in the same order for the rest of the equations are selected since it yields better results, especially for complex flows and tri/tet grids [32] temporary discretization. Initially, the implicit time integration equation can be commonly solved at an all-time level and thus has the benefit of being stable concerning the time size step. Besides, the implicit time equation is most appropriate in the density-based explicit solver situation [43]. In agreement with earlier studies, this study used the implicit method and the pressure-based solver [29,33]. Based on Jafarmadar [34] the wave method was chosen for the primary breakup method (liquid atomization) and the secondary breakup method (drop breakup), for the spray model. The emissions of NO are modeled through using extended Zeldovich mechanism. There are two acknowledged mechanisms that Jayashankara and Ganesan use here [35]. That usage nitric oxide through the combustion process: Thermal NO and Prompt NO.

3.1. Chemical kinetic mechanism

The eddy breaks up is used as the foundation for the modeling of combustion. The combustion of diesel includes a large number of reaction steps and chemical species, and for which mechanism was developed, it is based on the decreased alkanes combustion mechanism of Jones and Lindstedt [36]. Diesel reaction ($C_{12}H_{23}$) includes four global sequences of reactions defined by Eqs. (3-6).

$$C_{12}H_{23} + 6O_2 \longrightarrow 12CO + 11.5H_2 \quad (3)$$

$$C12H23\ 12 + H2 \longrightarrow 12CO + 23.5\ H2 \quad (4)$$

$$H2\ 0.5 + O2 \longleftrightarrow H2O \quad (5)$$

$$CO + H2O \longleftrightarrow CO2 + H2 \quad (6)$$

The mechanism reaction and Methane involve the three steps of global reaction

$$H2 + 0.5\ O2 \longrightarrow H2O \quad (7)$$

$$CH4\ 0.5 + O2 \longrightarrow CO + 2\ H2 \quad (8)$$

$$CO + 0.5\ O2 \longrightarrow CO2 \quad (9)$$

One reaction step involves hydrogen and reaction mechanisms [37].

$$H2 + 0.5\ O2 \longrightarrow H2O \quad (10)$$

The mixtures are supposed to obey the law of ideal gas. The computation of the specific heat, thermal conductivity, and viscosity of the mixture as temperature functions are all depending on the characteristics of each species. It is also suggested that the combustion products of mass fractions followed both the instantaneous and the local values of thermodynamic equilibrium. The equivalence ratio, temperature, and pressure are strongly affected by the cylinder's equilibrium composition [26]. The chemical species proposed for the calculation of CFD are CO_2, O_2, N_2, H_2O, H_2, NO, and CO, in which the natural gas is 100% methane (CH_4).

3.2. Grid generation

Firstly, the independence mesh examination was prepared for the Omega combustion chamber (OCC) bowl. Two measurement meshes had been created, specifically medium and fine. At an engine speed of 2000 rpm, the simulation was done for each mesh to illustration independence. Figure 2 indicates the expected in-cylinder pressure against the crank angle graph. The results were apparent that there is no marked difference between the medium and fine meshes. Indeed, the refined mesh raises the CPU costs by three times and meanwhile was not given beneficially for the CFD simulation. Hence, considering the computational time, the medium mesh was selected as the most suitable for this research.

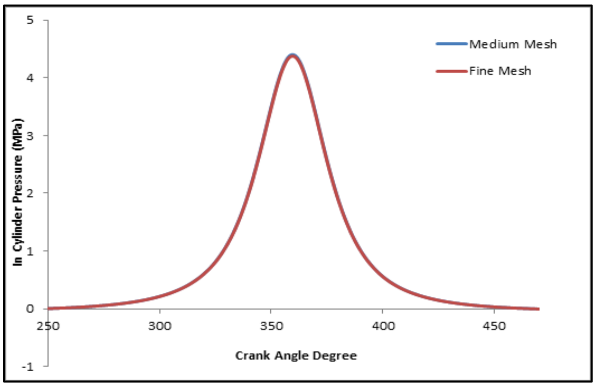

Figure 1. Calculated in-cylinder pressure results using mesh fine and medium.

3.3. Validation model

For validation, an engine model single cylinder was constructed and simulated. Engine description and further engine information utilized from the engine specifications of [27] were imported under specific conditions the speed engine (1500 rpm), temperature (330 k), and 100% diesel fuel. A 2D engine model was built and validated by comparing it with a set of experimental data for diesel operation. Moreover, to validate the simulation, the in-cylinder pressure rate was compared with the experimental results. The in-cylinder pressure validation in this study is presented in Figure 2. Generally, at most points, simulation data appear to achieve a good agreement with experimental data.

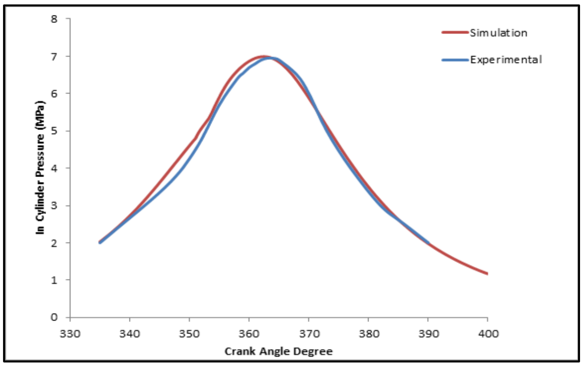

Figure 2. Validation of 2D simulation at 1500-rpm engine operation mode for In-cylinder pressure

4. Results and discussions

This section shows the numerical results of diesel, dual-fuel, and tri-fuel modes. Fuels were mixed with 100% NG and 100% H_2 for dual-fuel mode, and these percentages were alternated with 50% NG and 50% H2 for the tri-fuel mode under 1,500 and 2,000-rpm engine speeds. Using gaseous fuel will negatively affect the volumetric efficiency of the studied engine, hence, produced power reduced. In the study in hand, the effect of volumetric efficiency when using gaseous fuels with diesel ignored since the main aim is to investigate the influence on exhaust emissions.

4.1. In-cylinder pressure analysis

Figure 3 presents the effect of two selected engine speeds (1500 rpm-2000 rpm) on the peak in-cylinder pressure under pure diesel fuel, (diesel-NG, diesel-H_2) dual-fuel operations, and (diesel-NG-H_2) tri-fuel operations. The data in Figure 3 show there is no significant difference in peak in-cylinder pressure from comparing the two selected engine speeds at different types of fuel that used. However, the data also show an improvement in the peak pressure with gaseous addition to diesel fuel at two engine speeds. Furthermore, for 1500 rpm engine speed, peak pressure was improved from 7.12 MPa to 11.34 MPa (59% increase), 12.27 (72% increase) and 12.75 MPa (79% increase) for diesel fuel mode and adding NG in the following amounts, diesel-NG, diesel-NG-H_2, and diesel-H_2, respectively. Therefore, an increasing at fraction of H2 in the blend of fuel led to an increase in-cylinder pressure and advanced the occurrence of peak pressure due to the high content of specific energy and considerably faster flame as stated by Ref [37]. While the existence of H_2 increased in the blend, delays in ignition were shortened to reduced values than those of pure diesel. By contrast, the peak in-cylinder pressure decreased as the diesel fuel fraction increased due to the low flammability of the diesel fuel according to Ref [38]. The addition of pure NG to diesel enhanced ignition due to greater NG ignition power compared with diesel fuel.

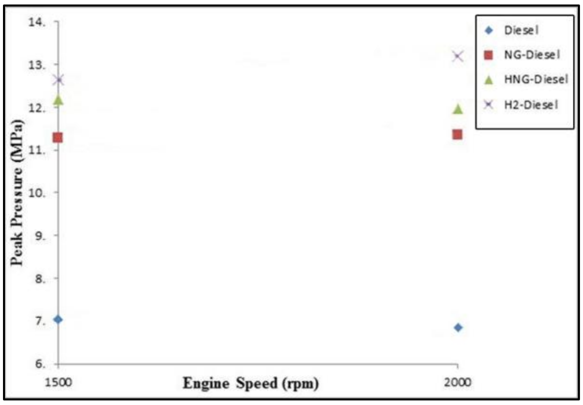

Figure 3. Effect of different ratio of gaseous addition on peak in-cylinder pressure under deferent engine speed.

Figure 4 summarizes the effects of altered mixtures of gasses fuel on peak in-cylinder pressure under two engine speeds. In the existence of H_2, the peak in-cylinder pressure was enhanced as previously demonstrated as a result of the high content of specific energy and increasingly fast flame. The combustion of H_2 increases the burning rate of diesel when combined [39]. The extensive premixed combustion phase indicates that the combustion of H_2 occurred through the premixed combustion phase of diesel fuel. Consequently, a long-premixed combustion period and a brief combustion stage of diffusion were accomplished. A fast decrease in the ignition delay was noted for diesel-H_2 combustion, which is in line with the preceding study Szwaja and Grab-Rogalinski [40]. In the current research, the shortened ignition delay is explained through two potential causes. Firstly, H_2 auto-ignition via the hot spot surface may happen under extreme circumstances or H_2 pre-ignition [41]. Secondly, a significant quantity of H_2 addition is likely to modify the chemistry of diesel oxidation and ignition delay, thus increasing at high temperatures as earlier described by [42]. As noted in the combustion of diesel-NG, the increase of peak pressure was much smaller than that of the diesel-H_2 blend. In this case, the source of ignition was imparted by diesel, and the subsequent propagation of the flame was based on the combustion of the pre-mixed gaseous fuel-air blend. The additional significant reason for this finding is the alteration between H_2 and NG in flame propagation speeds. NG with low-speed flame propagation reduced the impact on cylinder pressure and subsequently heat release rate. Long ignition delay of NG influenced the initial phase of pressure increase in all cases.

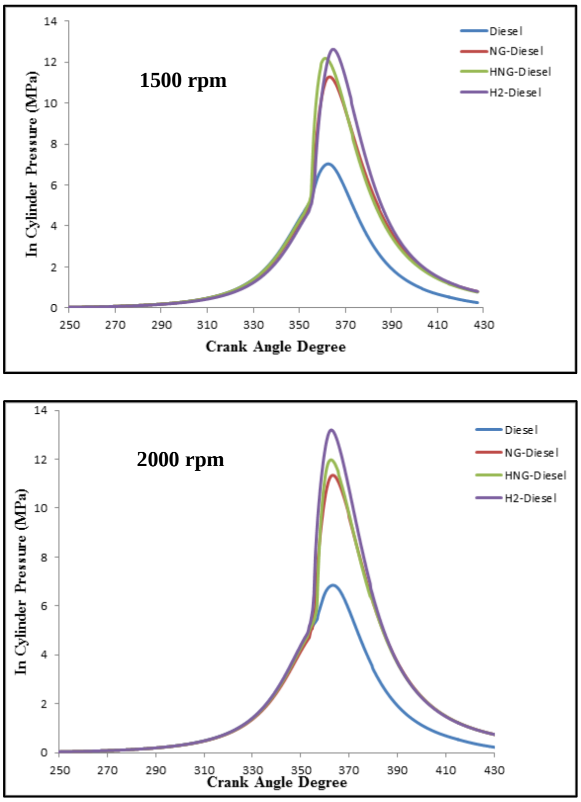

Figure 4. In-cylinder pressure curves under the different ratio of gaseous addition and engine speed

4.2. In-cylinder Temperature analysis

Figure 5 illustrates the peak in-cylinder temperature for different gases fuel substations and the different values of selected engine speeds (1500, and 2000 rpm). The data indicate there is no significant difference in peak temperature from comparing the two engine speeds at different types of fuel used. For 1500 rpm engine speed, the peak in-cylinder temperature was increased with the existence of each fuel gaseous and was obvious with an elevated H_2 percentage because hydrogen's fast-burning level led to the high combustion temperature as stated previously. The peak in-cylinder temperature enhanced from 1490 K to 2365 K (58.7% increase), 2591 K (73.9% increase), and 2787.7 K (87.1 increase) for diesel fuel mode and adding NG in the following amounts, diesel-NG, diesel-NG-H_2, and diesel-H_2, respectively. On the other hand, the peak in-cylinder temperature at diesel fuel mode decreased because of the low flammability of diesel combustion as previously confirmed.

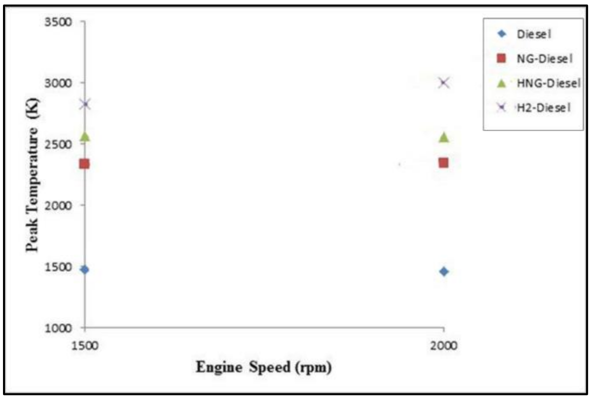

Figure 5. Effect of different ratio of gaseous addition on peak in-cylinder temperature under deferent engine speed.

Figure 6 shows the effect of altered blends mixture on the in-cylinder temperature. The variation in diesel-H_2 and diesel-NG combustion under the similar substitution ratio of energy was largely due to distinct H_2 and NG flaming propagation speeds. The NG flame propagation speed was considerably lower than that of H_2, leading to its slighter impact on temperature rate and in-cylinder pressure [34]. Furthermore, a significant ignition delay for diesel-NG was noted in all engine operating conditions. The NG presence advantage is the enhancement combustion of H_2 by preventing uncontrolled combustion, for example, a significant increase of peak pressure and temperature as illustrated in the literature.

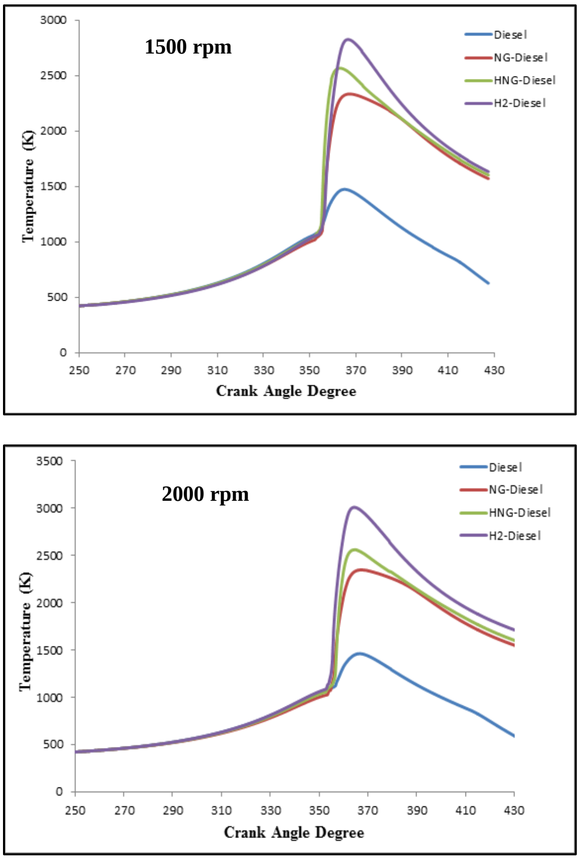

Figure 6. Temperature curves under the different ratio of gaseous addition and engine speed

Figure 7 displays the temperature contours of altered mixtures under an engine speed of 2,000 rpm. High H_2 content resulted in a large fraction of the high concentration red zone of temperature, which expanded further to the cylinder squish area. Similarly, but with decreased magnitude, diesel-NG combustion reached greater temperatures relative to the diesel fuel combustion engine.

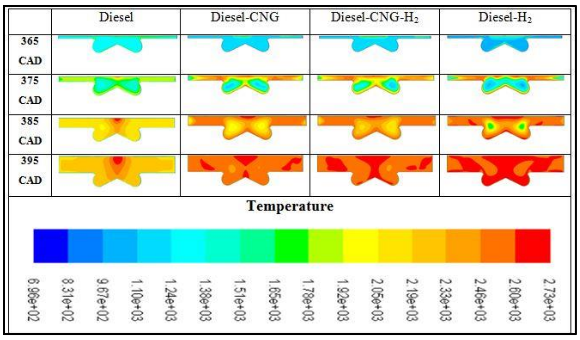

Figure 7. The development of average temperature under the different ratio of gaseous addition and 2000 rpm

5. Emissions

5.1. NO emissions

Fig. 8 demonstrates the variation of NO emissions with a different value of engine speed and gases fuel substations. The data shows there is no significant difference in NO emissions at two selected engine speeds at altered types of fuel used. For diesel-NG operation, the emission of NO slightly increased compared with the operation of diesel fuel for two engine speeds as a result of the low temperature of diesel fuel combustion according to Ref [38]. The addition of H_2 to NG increased the NO mass fraction compared with diesel and NG, with values rising from 0.00000516 to 0.00122, 0.00298, and 0.00425 when using diesel, diesel-NG, diesel-NG-H_2, and diesel-H_2, respectively under 1500 rpm engine speed. Hence, the NO formation was enhanced, and the mixture's burning rate increased as the H_2 fraction in NG increased. According to Choi et al. [43], the rise in NO emissions when H_2 is added is dependent on H_2 fuel's flame temperature, which is higher than that of the NG fuel and diesel fuel. Moreover, the emission of NO was increased in diesel-H_2 and diesel-NG-H_2 compared with pure diesel and diesel-NG under two engine speeds. In contrast, NO emission decrease is a consequence of NG fraction increasing in the blend, as discovered by Ref [37]. The mixture has the maximum amount, average, and the lowest proportion of NG in H_2 shown the best result towards the reduction of NO emission. Consequently, NG has low flame propagation speed, and narrow flammability can make combustion of H_2 smoother and steadier and can thus help prevent insufficient combustion. Besides, NO emissions in diesel-NG-H_2 operations were smaller than in diesel-H_2 operations.

Figure 9 summarises the influences of different gaseous fuel blend and diesel on the in-cylinder of NO development. Increasing diesel fraction has eliminated the formation of NO due to low temperature of diesel combustion, whereas the best for the decrease in NO emission was studied using the highest NG or the lowest H_2 values as illustrated in the previous paragraph.

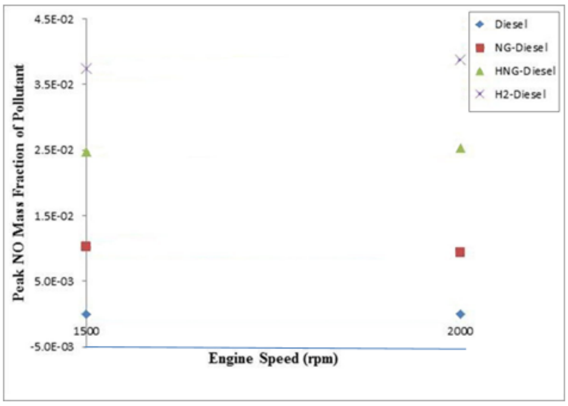

Figure 8. Effect of different ratio of gaseous addition on peak NO emissions under deferent engine speed.

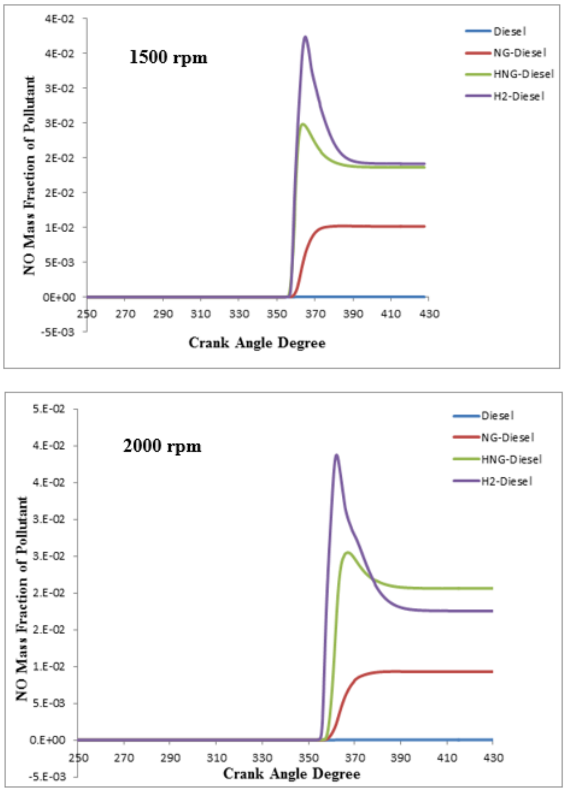

Figure 9. NO emissions curve under the different ratio of gaseous addition and engine speed

Figure 10 illustrates the contours formation of NO mass fraction at 2000-rpm engine speed. The operation of diesel-H_2 dual fuel also showed a high concentration red zone of NO distribution. The amount of formation of NO in the mode of diesel-NG dual fuel exceeded that of the normal operation of diesel fuel due to the diesel-NG combustion attained higher temperature than the normal diesel fuel. The combustion of diesel fuel engine extended to a cylinder's squish area, and the high distribution developed of NO formation at squish and swirl zones for 385° CA. However, rising diesel fraction reduced the NO formation. Another significant contributor to NO development is the equivalence ratio, where NO is maximum at stoichiometric value. However, the results indicate that while the maximum equivalence ratios were found with 50% NG and 50% H_2, temperature affects seemed to be the dominating factor.

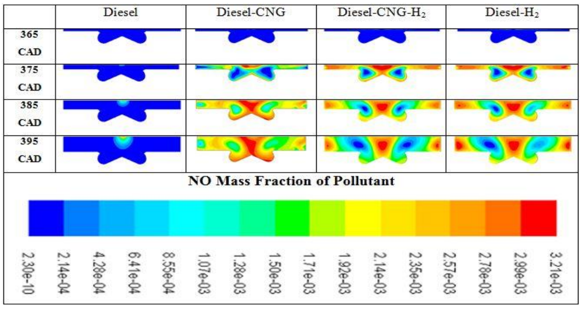

Figure 10. The development of NO mass fraction of pollutant under the different ratio of gaseous addition and 2000 rpm

5.2. CO Emissions

Incomplete combustion is the main cause of most CO emissions. Incomplete combustion occurs because of low gas temperatures, and the lack of oxidants and is overseen by the air-fuel equivalence ratio [37]. Figure 11 demonstrates the CO emission variations for operations of the diesel, dual, and tri-fuel engines under 1500 rpm, and 2000 rpm engine speeds. The figure shows there is no significant difference in CO emissions at two selected engine speeds at altered types of fuel that used. For diesel-NG operation, CO is one of the by-products of incomplete combustion of NG. CO emission increases significantly when fuels gaseous present with NG content but reductions with H2 fraction, especially compared with the addition of diesel fuel and H_2 to NG. The values rise from 0.004651 to 0.006533, 0.0325, and 0.04837 for operations of diesel fuel and with the addition of diesel-H_2, diesel-NG-H_2, and diesel-NG, respectively under 1500-rpm engine speed. Additionally, the high carbon ratio of NG could have led to an increase in CO emission. These outcomes are closer to the experimental study by Gatts et al. [44], which studied the NG combustion completion of the dual-fuel (diesel-NG) operation. Unburned CO emissions were used in their research to determine NG's combustion efficiency, which is determined by NG content and engine speed.

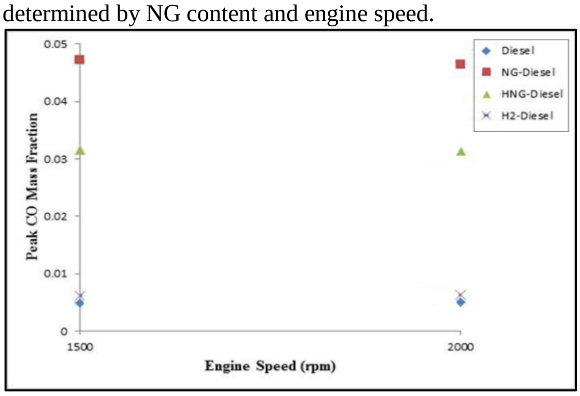

Figure 11. Effect of different ratio of gaseous addition on peak CO emissions under deferent engine speed.

Figure 12 shows the impacts of various fuel mixtures on the CO peak level with varying engine speeds. No considerable difference in CO emission was observed when the speed engine was increased. That high NG presence increases the CO peak rate, which can be related to its significantly slower flammability and the high carbon ratio of NG resulting in incomplete combustion [38]. The addition of H_2 in NG enhances NG's flammability as stated by Ref [45]. An enhancing concentration of O, H, and OH radicals was detected, rising key reaction rates to enhance NG-lean combustion efficiency with decreased CO emissions. Additionally, H_2 has a high flame velocity resulting in complete combustion that causes to reduce CO emissions. The reduction will be further decreased with the addition of H_2 in diesel. However, reducing CO emissions by increasing the diesel fraction may indicate that raised H_2 content causes CO_2 dissociation to CO. On the other hand, CO is additionally decreased at a high fraction of diesel.

Figure 13 showed the region formation of carbon monoxide at 2000 rpm under different ratios of addition gaseous. These contours illustration at 375, 385, and 395 CAD for addition H_2, as compared with diesel and NG. The formation region of CO reductions when the H_2 mass fraction addition to the gaseous and this seems obvious at diesel-H_2 dual-fuel operation and 50NG-50H_2 tri-fuel operation tables. This is due to the high flame velocity of hydrogen and resulting in complete combustion. In contrast, the formation region of CO emissions was increased as illustrated at 375, 385, and 395 CAD for diesel-NG tables. Thus, the NG has a higher carbon ratio and helps to produce CO emission.

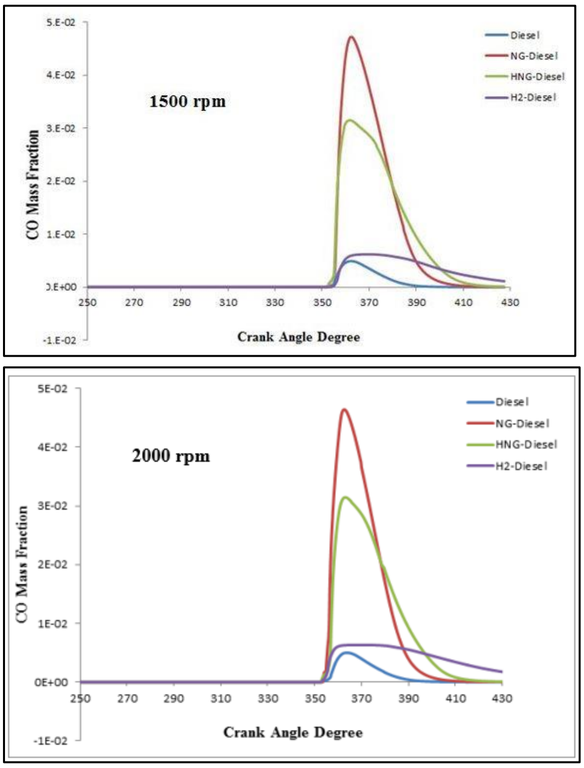

Figure 12. CO emissions curve under the different ratio of gaseous addition and engine speed

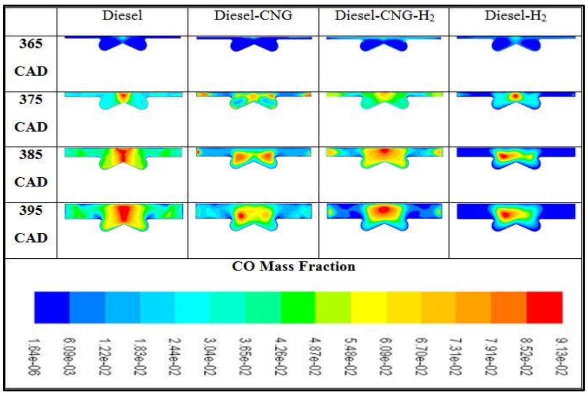

Figure 13. The development of CO mass fraction under the different ratio of gaseous addition and 2000 rpm

6. Conclusion

The present study used a direct injection method with a single cylinder to investigate the effect of using diesel with NG and H_2 at various mixture ratios under two engine speeds on a diesel engine by using a Computational Fluid Dynamics. The following conclusions are drawn from the simulation results.

- For all gaseous fuel additions under two engine speeds, in-cylinder pressure and temperature are increased as compared to normal diesel fuel operation. The improvements are related to the increase of H_2 fraction.
- Combustion is affected significantly by H_2 addition and leads to the high temperature of combustion, which can decrease the efficiency of the dual-fuel engine.
- Dual-fuel operations have a considerable influence on engine emissions. Diesel-NG mode, however, raises CO emissions, and diesel-H_2 operations raise NO emissions compared to normal diesel fuel operations.
- Tri-fuel operation involving diesel-NG-H_2 decreased CO emissions comparative to diesel-NG. It also lowered the emission of NO for all engine speed when compared to the diesel-H_2 operation. The high fraction of H_2 in NG cause to lower CO emissions. In contrast, Lower concentration of H2 in NG minimizes unwanted H_2 combustion and restricts NO emission rises.

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