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Numerical analysis of diesel injection strategies on emissions and performance in CH_4/diesel powered RCCI diesel engine with high ratio EGR

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KEYWORDS

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Post injection;
EGR;
Diesel engine

Abstract In this study, the effects of intake port injection of methane and direct of injection diesel on emissions and the combustion of a light-duty RCCI engine were numerically researched. In this way, AVL Boost software was used for 1-dimensional simulation of the combustion process and emission estimation. Higher octane number methane gas was mixed with air from the intake port, while lower octane number diesel fuel was injected directly into the combustion chamber during the compression stroke. Methane gas was injected at a rate of 65% and diesel fuel at a rate of 35%. The diesel injected directly into the combustion chamber was sprayed at a rate of 5% between 0 and 35 °CA with 8 different injection timings after the main injection. In the model engine, 50% high EGR rate was applied in all combustion modes. The results showed that these parameters have significant effects on performance and emissions. The results in summary; NOx emission reduced at all engine speeds with delayed diesel post-injection timing. The maximum drop of NOx emission was 57.34% with Post15. Although the addition of methane slightly increased the soot emission, it was significantly reduced by the simultaneous addition of methane and application of post-injection strategies of diesel fuel. The soot emission was reduced by 58.85% with the Post35 injection strategy.

Abbreviations: bTDC, Before Top Dead Center; BMEP, Brake Mean Effective Pressure; BSFC, Brake Specific Fuel Consumption; CO2, Carbon Dioxide; CER, Carbon Energy Ratio; CO, Carbon Monoxide; °CA, Crank Angles; DPF, Diesel Particulate Filter; DFM, Dual-Fuel Combustion Mode; eEGR, External Exhaust Gas Recirculation; EGR, Exhaust Gas Recirculation; GHG, Greenhouse Gas; HRR, Heat Release Rate; HC, Hydrocarbon; LTC, Low-Temperature Combustion; CH4, Methane; MCC, Mixing Controlled Combustion; NG, Natural Gas; NOx, Oxides of Nitrogen; PM, Particulate Matter; PC, Post Combustion; PF, Primary Fuel; RCCI, Reactive Controlled Combustion; SCR, Selective Catalytic Reduction

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1. Introduction

Diesel and gasoline still have a essential place in the transportation industry. However, the usage of these fuels is limited due to the shrinking of reserves around the world and their very dreadful damage to environmental health [1]. The expansion in the rates of pollutants originating from conventional fuels has increased the search for cleaner and more reliable fuels from these fuels [2]. Diesel engines are particularly used in off-road vehicles due to high of their thermal efficiency. However, NOx emissions, which is one of the most dangerous emissions from diesel [3], pose a significant problem due to the high thermal efficiency of the diesel engine. To decrease particulate matter (PM) and NOx emissions, it is recommended to use dual fuels with various strategies, apart from diesel particulate filter (DPF) and the use of selective catalytic reduction (SCR) [4,5]. The use of clean gas fuels such as hydrogen, natural gas, etane and methane is advantageous to reduce the use of harmful fossil fuels such as diesel and gasoline, which are sources of polluting emissions [6]. Methane (CH_4), one of the most important gaseous fuels, is the main component of natural gas (NG), attracting attention with its low flame temperature and the lowest carbon energy ratio (CER) between fossil fuels [7,8]. Methane is a neat-burning fuel that produces low greenhouse gas (GHG) emissions [9]. Methane fuels reduce CO_2 emissions more than conventional fuels such as gasoline and diesel. Additionally, since methane is a gas, it provides a better homogeneous mixture to liquid fuels and reduces particulates and pollutant emissions [10,11]. The low-temperature combustion (LTC) application that enables methane to be used in existing diesel engines is dual-fuel combustion mode (DFM) technology [12]. Since the diesel-methane dual-fuel engine is suitable to run on both diesel and methane, it effectively reduces CO, HC, and greenhouse gas emissions [13]. Reactive controlled combustion (RCCI) is a DFM approach in which at least two low reactive (low cetane number) fuels are injected into the port to provide a homogeneous blend with the air, and the highly reactive fuel is directly sprayed into the cylinder [12,14,15]. Fig. 1 shows the schematic of RCCI combustion mode.

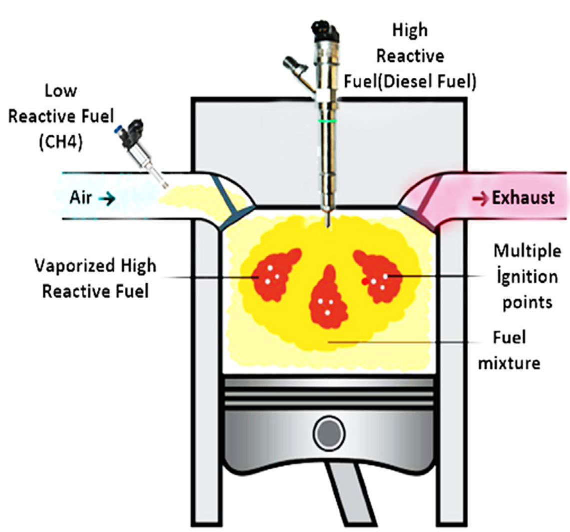

Fig. 1 Schematic of RCCI combustion model.

This study aims to reduce the amount of emission by high EGR rate and post-injection, as well as RCCI combustion mode in a diesel engine. AVL Boost software was used to the effect of injection timing and high EGR ratio on emissions and combustion analysis. The study showed that the effect of RCCI combustion on NOx and soot emissions can be further improved with the high EGR ratio and post-injection. The study will contribute to the limited literature as an innovation in terms of performing reactive controlled combustion analysis of high-rate methane and EGR in light and heavy commercial vehicle engines and investigating the effects of diesel post injections on these two analysis.

2. The combustion process of methane and diesel in RCCI diesel engine

Since dual fuel combustion works with a poor air/methane ratio at low engine load conditions, the flame propagation speed does not reach distant areas with a small amount of injected diesel. Therefore, this situation causes a slowdown in the combustion speed and an increase in CO and HC emissions with incomplete combustion [16]. However, since DFM operates with rich air–fuel rate at high engine load, high NOx emissions with high cylinder temperature occur in the diesel engine. To overcome these combustion and emission problems in engines, many methods such as dual fuel mode (RCCI), gas-diesel injection, split pilot-post injection times [17,18], intake air temperature, Exhaust Gas Recirculation (EGR) [19], divided fuel amount [20] can be applied. However, there are many difficulties in the use of natural gas/methane as fuel in engines. Since natural gas/methane is a gas fuel, when it is used as a port fuel, the decrease in volumetric efficiency, high ignition temperature, and the lack of a constant ratio of air-methane fuel mixture are among these difficulties [21,22]. The RCCI engine module makes a significant contribution toward solving these problems [23]. Defined as the primary fuel (PF), methane is taken in together with air during the intake stroke in the RCCI engine. In the compression stroke, the pressure and the temperature of the primary fuel are increased. As soon as the compression stroke is very close to the end, the secondary fuel diesel is sprayed directly into the cylinder for ignition purposes. Diesel fuel forms a flame front in many areas within the pressurized air-methane mixture. In this way, the combustion of the mixture is smoother and faster [24–26]. Due to the low reaction temperature in the RCCI engine, NOx emission decreases, while CO and HC emissions are higher than those of diesel [27]. In addition, the fact that natural gas (methane) has a longer ignition delay due to high auto-ignition temperature than conventional fuels causes a rise in CO and HC emissions [28,29].

3. The combustion process of post-injection of secondary fuel in diesel engine

A Natural Gas/methane engine produces lower greenhouse gas (GHG) emissions at high engine loads than a diesel engine [30]. However, at low and medium loads, it creates higher unburned CH_4 than a diesel engine. Because the lean CH_4-air mixture has a weak flame propagation speed. As a result, unburned CH_4 is the main cause of unburned hydrocarbons [31]. Post injection timing is one of the best ways to deal with unburned methane in a gas DFM engine at low engine load conditions. Post injection is the injection of some quantity of main fuel in the late expansion stroke after the main injection [31]. Pollutant emissions can be controlled by phasing diesel combustion. While pre-injection reduces NOx emissions and combustion noise, early post-injection helps to reduce soot emissions and late post-injection helps to renew the diesel particulate filter [32–34]. Some post-injection (PI) applied after the main injection can reduce the soot emission and increase the temperature of the exhaust gas with the effect of the fuel that is subsequently included in the combustion [35]. Post-combustion (PC) is very effective in fuel oxidation due to the rise in-cylinder temperature.

4. Effect on the combustion process of exhaust gas recirculation (EGR) in methane-diesel engine

NOx emission resulting from combustion in diesel engines generally consists of NO and NO_2 emissions. The main process of NO emission is the oxidation of nitrogen in atmospheric air under high temperatures. Therefore, NO emission occurs during combustion in the high-temperature zone of the combustion chamber [36]. Nitrogen dioxide (NO_2) is normally formed by the oxidation of nitrogen monoxide (NO) with oxygen in the air at high temperatures [37]. When some of the exhaust gas is returned to the cylinder, it plays a role as a diluent for the fuel/air mixture. EGR dilutes the fuel–air mixture in the cylinder and reduces the oxygen concentration. In addition, since the specific heat of the exhaust gas is much higher than the specific heat capacity of the air taken into the cylinder, it reduces the increase in heat release rate. [36]. As the EGR ratio increases, NOx formation decreases [38]. Because the external EGR (eEGR) can be circulated up to 50% without extreme affecting the stability of combustion in a diesel engine, it is a very effective method in reducing NOx emissions [39,40]. However, although NOx emission decreases with EGR, a slight increase can be observed in specific fuel consumption and soot emission [41]. Also, EGR amount and EGR temperature (cool, hot) are important factors in the formation of these emissions [42]. Significant reductions in NOx emissions can be observed when the eEGR gas is cooled before entering the cylinder [40]. When RCCI and EGR applications are used together in a methane/diesel DF engine, NOx emissions are considerably decreased. In addition, if the amount of EGR is kept small, a slight decrease in HC and CO emissions is detected [43].

In summary, there are many studies on reducing soot and NOx emissions. However, the impact of the combined application of methane, post-injection, and EGR parameters on the performance and emissions in the RCCI engine is unclear. In this respect, in the current study, the effect of RCCI application on engine emissions and performance in a CH_4/diesel DF diesel engine using EGR was numerically investigated in AVL Boost simulation software. A three-cylinder turbocharged diesel engine was built and simulated. EGR ratio used in the engine model was set to 50%. The effects of fixed-rate methane gas injected into the intake port and post-injections of diesel fuel injected directly at different times on emissions and performance were analyzed.

5. Methodology, conditions, and validation of the numerical model

In this study, first of all, the 3-cylinder experiment engine used in our previous studies was modeled and calibrated in the AVL BOOST software. The characteristics of the model engine are shown in Table 1.

The validity of the model engine using the mixing controlled combustion (MCC) model was provided with the data obtained from the experiments. The validation graph is shown in Fig. 2 with heat release rate (HRR) and the pressure curves of the test and modeled engine. A good harmony was seen between the experimentally measured and the numerically calculated pressure and HRR data. The model appears to be capable of computationally estimating the start of combustion and engine maximum pressure curves with acceptable accuracy compared to the experimental test data. Later, this model was rearranged by adding the EGR system. The EGR ratio was kept constant at 50%. The 1-dimensional diagram of the model engine is shown in Fig. 3. The numerical model consists of system boundary SB1, turbo compressor TC1, air cooler CO1, throttle valve R1, plenum PL1, system boundary SB2 from the cylinder in the exhaust gas line, turbo turbine TC1, and Plenum PL2. In addition, there are Port injector I1 on the intake line, cylinders C1-C4, External Exhaust Gas Recirculation line, and exhaust cooler (CO2) elements. In addition

Table 1 Model engine features.

Parameters	Characteristics
Number of Cylinder	3
Aspiration	Turbocharge
Engine Volume	2.9 L
Compression Ratio	17:1
Bore × Stroke	104 × 115 mm
Rated Speed	2450 rpm
Maximum Output	36 kW
Cooling System	Water Cooled

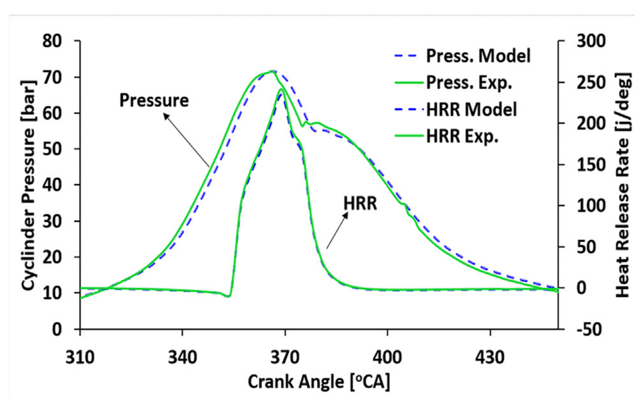

Fig. 2 HRR and pressure curves of experiment and model engines.

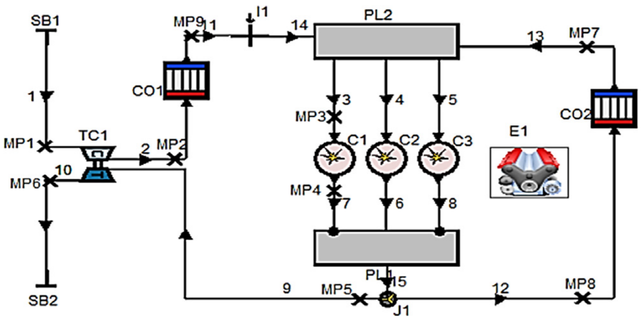

Fig. 3 One-dimensional (1D) engine model.

to these, M1-10 measurement points have been added to monitor the simulation measurements.

MCC (mixing controlled combustion) combustion model was used in the engine modeled in the AVL Boost software. Because the MCC combustion model is very successful in predicting the heat release, production of NOx, and soot emission, which can be used for multiple and single injections depending on the amount of liquid/gas fuel and turbulent kinetic energy in diesel engines. The AVL Boost MCC model facilitates the modeling of complex burning processes of single and dual fuel combustion modes (DFM) of gaseous and liquid fuels such as diesel, gasoline, hydrogen, and methane, using detailed attributes and high-order equations [44–46]. In addition, this combustion model can predict the effect of EGR on NOx emission, soot emission production, and combustion. The engine model was adapted for methane/diesel blend fuel using modified parameters with quenching distance and various equivalence ratios of methane gas fuel.

The AVL MCC model is quite capable of accurately predicting the occurrence of soot and nitrogen oxide emissions [47]. NOx was estimated by a three-stage extended Zeldovich mechanism in the combustion model. In the model, the total HRR is calculated by Eq. (1) [44]. The AVL Boost MCC model uses the Vibe function shown in Eq. (2) when calculating the total HRR during combustion based on the crank angle. In the MCC model, Vibe's law (Eq. (2)) is used to approximate the actual heat release value of the internal combustion engine [48,49]. Also, the total heat release rate is expressed by Eq. (3) [47]. The Vibe heat release function is practical and simple.

Woschni uses the heat transfer model to calculate the heat loss from the cylinder walls [50,51]. In the MCC combustion model, the Woschni model in Eq. (4) is used to calculate the heat transfer losses in the cylinder wall. The Woschni combustion correlation performs complex heat transfer processes by simply and accurately converting them into heat transfer coefficients [51].

$$\frac{dQ_{total}}{d\alpha} = \frac{dQ_{MCC}}{d\alpha} + \frac{dQ_{PMC}}{d\alpha} \tag{1}$$

$$\frac{\frac{dQ_{PMC}}{Q_{PMC}}}{d\alpha} = \frac{a}{\Delta\alpha_c}(m+1)\mathrm{y}^{\mathrm{m}}\mathrm{e}^{-a\mathrm{y}^{(\mathrm{m}+1)}} \tag{2}$$

$$\frac{dQ_{MCC}}{d\alpha} = C_{Comb}.\left(m_F - \frac{dQ_{MCC}}{Hu}\right).(w_{air})^{C_{EGR}}.C_{Rate}.\frac{\sqrt{k}}{\sqrt[3]{V_D}} \tag{3}$$

$$a_w = 130.\mathrm{D}^{-0.2}.p_c{}^{0.8}.T_C^{-0.53}.\left[C_1.C_m + C_2.\frac{V_D.T_{C,1}}{P_{C,1}.V_{C,1}}.(P_C - P_{C,0})\right]^{0.8} \tag{4}$$

where the variables in the equations represent:

Q_{PMC}, the heat released throughout premixed combustion, Q_{MCC}, the heat released during mixing controlled combustion. m is the combustion intensity shape parameter, $d\alpha$ is the crank angle, C_{Comb} is the combustion constant, C_{EGR} is the EGR influence constant, C_{Rate} is the mixing ratio constant. w_{air} is the effective air mass fraction, k is the turbulent energy density, $andH_u$ is the low calorific value of the fuel. m_F is the evaporative fuel mass, $C_1 = 2.28 + \frac{0.308\mathrm{Cu}}{\mathrm{Cm}}$, Cm is the average piston speed, and Cu is the circumferential speed. The coefficient is taken as 0.00324 in direct injection engines. V_D represents the displacement volume per cylinder, D indicates the diameter of the cylinder, $T_{C,1}$ represents the temperature value when the intake valve is closed, $P_{C,0}$ shows the pressure in the cold cycle, $P_{C,1}$ shows the pressure value when the intake valve is closed. y is the relative combustion duration: $\mathrm{y} = \frac{\alpha-\alpha_0}{\Delta\alpha_c}$, where $\Delta\alpha_c$ is the combustion duration, and α_0 is the start of combustion.

The model engine was first run with only diesel fuel as the reference fuel at an engine speed of 2000 rpm. Then, CH_4 gas was injected into the intake port, equaling 65% of the total energy introduced into the cylinder with diesel fuel. Therefore, the total energy amount entering the cylinder remained stable with energy from 65% CH_4 and from 35% diesel fuel. In this model engine, when the rate of methane injected into the intake port is kept higher than 65%, the volumetric efficiency drops too much due to the methane being in the gas phase. Therefore, the injected methane rate was not used more than 65% and was kept constant at this rate. The properties of diesel and methane fuels are shown in Table 2 [12,52]. In the final stage, 10% (Post fuel quantity) of diesel fuel with a total mass of the range of 23.65–25.95 mg was injected as post fuel at 0, 5, 10, 15, 20, 25, 30, and 35 crank angles (°CA) after the end of the main injection. Fig. 4 shows the injection rate profile of the main and post injections of diesel fuel. The operating parameters of the model are presented in Table 3.

In this study, minimum diesel conditions were determined to ensure flammability. Since the ignition start could not be achieved when the diesel fuel rate used in the tests was below 30%, studies were conducted with a diesel fuel rate of 35%. In addition, post-injection limits were determined so that BSFC was not affected much, fuel injection did not shift to late

Table 2 General properties of diesel and methane fuels.

Parameter	Methane	Diesel
Chemical formula	CH4	C12H22
Lower heating value, [MJ/kg]	50.04	42.83
Air/fuel ratio	17.23	15.17
Heat of evaporation, [J/kg]	5.13 × 105	2.75 × 105
Total carbon content of the fuel	0.748	0.839
Density, [kg/m^3, 20 °C]	0.66	820
Carbon content [wt. %]	74.86	85.94
Hydrogen content [wt. %]	25.14	14.06
Auto ignition [°C]	537	200–300

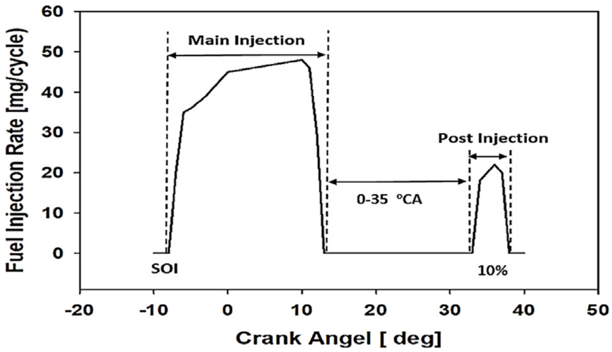

Fig. 4 Post injection rate strategy.

Table 3 The operating parameters of the model.

Parameter	Methane	Diesel
Port Fuel Ratio	65%,	–
Main Fuel Ratio	–	25%,
Post Injection Ratio	–	10%,
Post Injection Angle	–	0–35 °CA
Rail Pressure	–	1800 bar
EGR	Constant, 50%	Constant, 50%
Engine Speed	800, 1200, 1600, 2000 rpm	800, 1200, 1600, 2000 rpm
SOI	Intake Stroke	Constant, −8 CA

injection, and the heat release rate remained in the region we can control. In addition to these conditions, basic boundary conditions such as certain engine speeds, 65% methane ratio, 35% diesel ratio, and 50% EGR ratio were determined.

6. Results and discussion

In this division, the effects of diesel post-injection strategies on emissions and combustion characteristics of DF combustion RCCI engine mode were researched numerically using 65% methane gas energy ratio. These results were compared with those of methane/diesel combustion mode single diesel injection and main and post diesel strategies under constant engine load and different engine speed operating conditions. In the research, cylinder pressure, HRR, engine power, BMEP, BSFC, NOx, and soot emissions were investigated for each operating conditions. Emission analysis were limited to NOx and soot emissions, which are more problems in diesel engines.

Figs. 5-8 which is fuel energy input kept the same shows the heat release rate and the cylinder pressure of the methane/diesel DF mode for single and main-post diesel injection strategies at constant diesel main-injection timing of −8 ○CA bTDC (Before Top Dead Center) under the 2000 rpm, 1600 rpm, 1200 rpm, 800 rpm engine speed conditions. The simulation results showed that the in-cylinder pressure did not alter seriously when the engine was operating with methane-diesel mixture and post-injection strategies. All the resulting pressure graphs overlapped almost exactly. In all tests, the maximum pressure value was obtained with diesel fuel and was determined as 74 bar. Since the velocity of the flame front of methane is slower than that of diesel and the combustion time is longer, a decrease in in-cylinder pressure and less intense combustion occurs than in diesel [52–54]. Post-injection strategies showed lower peak pressure values than single injection strategies. In addition, post-injection HRRs reached a lower peak than single injection HRRs. Because EGR and the reduction of the amount of main fuel injected with post-injection are effective in decreasing the peak point of HRR [55]. The peak HRR of post-injection strategies was observed in the Post0 early post-injection strategy due to higher oxygen usage. The peak was lower in the expansion stroke as there was more oxygen consumption and lower in-cylinder temperature in post injections [18].

The engine power for the various injection applications of diesel fuel at different engine speeds is represented in Fig. 9. Engine power decreased slightly at each engine speed with methane added to diesel fuel at the rate of 65% of the total energy. This reduction was due to the reduction in volumetric efficiency because of methane injected into the intake port. Compared to diesel fuel, engine power was reduced by up to 16.66% under all engine speed conditions owing to the reduction in volumetric efficiency with 65% methane injected into the intake port.

BMEP (brake mean effective pressure) for various injection strategies at different engine speeds is presented in Fig. 10. BMEP decreased at all engine speeds with CH_4 and diesel sin-

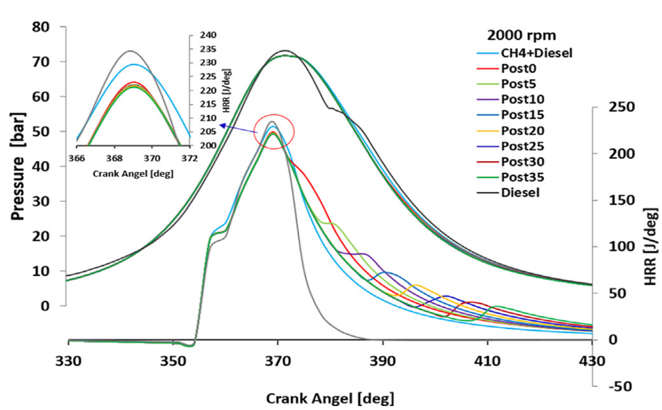

Fig. 5 HRR and in-cylinder pressure versus crank angle for various post-injection strategies at 2000 rpm.

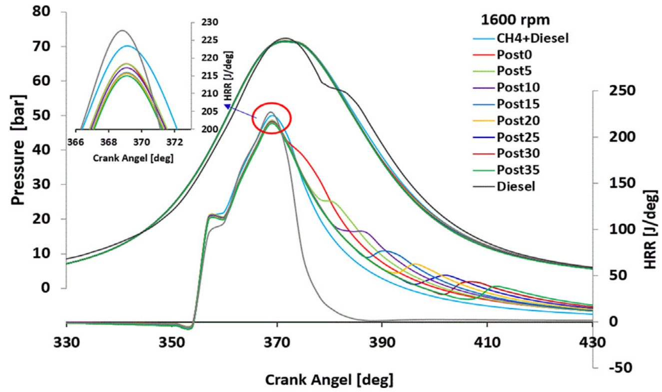

Fig. 6 HRR and in-cylinder pressure versus crank angle for various post-injection strategies at 1600 rpm.

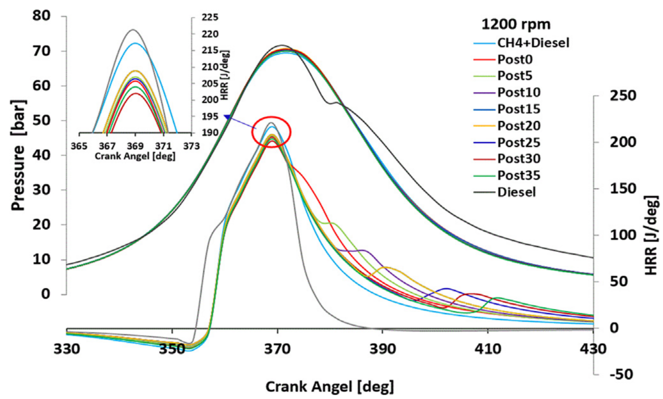

Fig. 7 HRR and in-cylinder pressure versus crank angle for various post-injection strategies at 1200 rpm.

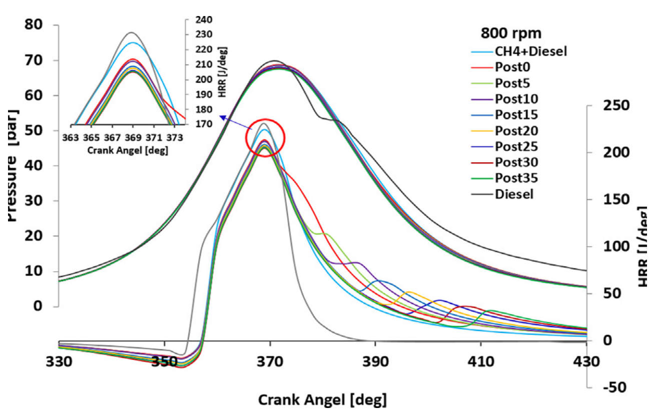

Fig. 8 HRR and in-cylinder pressure versus crank angle for various post-injection strategies at 800 rpm.

gle injection and CH_4 and diesel post-injection strategies compared to diesel fuel. The lowest BMEP was detected at a rate of 35.33% at 2000 rpm engine speed and with Post35 injection compared to methane and diesel fuel.

The brake-specific fuel consumption (BSFC) for the diverse injection applications at various engine speeds is presented in Fig. 11. The bar graphs indicate the same propensity for the DF and the diesel modes at all engine speeds examined. BSFC decreased at all engine speeds and in all operating modes. In addition, BSFC was lower under post-injection strategies and dual-fuel mode compared to diesel for single injection. While the highest reduction of brake-specific fuel consumption was obtained with methane/diesel single injection study at 47.05% rate, the highest reduction of BSFC in the split injection studies was provided with post0 at 45.44% rate. The least improvement in BSFC occurred with Post30 and Post35, 3.74% and 1.60%, respectively. The decrease in the improvement in BSFC at high engine speeds was owing to the fact that the high EGR ratio has a distorting effect on combustion. However, this distorting effect decreased as the post-spraying was performed late.

The NOx emissions for the various injection applications at different engine speeds are presented in Fig. 12. At all engine speeds, a lower NOx emission concentration was noticed in the DF mode compared to single injection diesel fuel due to PF methane, which reduces the oxygen concentration in the cylinder. Because the use of EGR in a dual-fuel engine greatly reduces NOx emissions [43]. At medium and high engine speeds (1600 and 2000 rpm), higher NOx emissions were produced for methane and diesel fuel operation compared to post-injection strategies, while lower NOx emissions from all fuel tests due to lower in-cylinder temperatures at low engine speeds (1200 and 800 rpm). Compared to pure diesel fuel at 2000 rpm engine speed, 27.97% NOx reduction was achieved when 65% methane was added. In addition, with post-injection timings, NOx emissions were greatly reduced compared to diesel and methane and diesel fuel runs. Nevertheless, with the improvement of combustion in post20-35 injections,

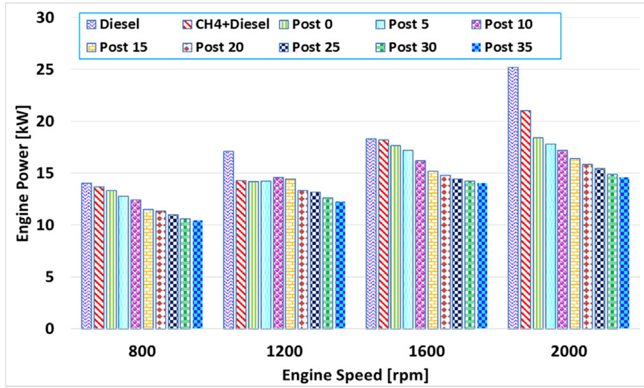

Fig. 9 Engine power variation of post injections at different engine speeds.

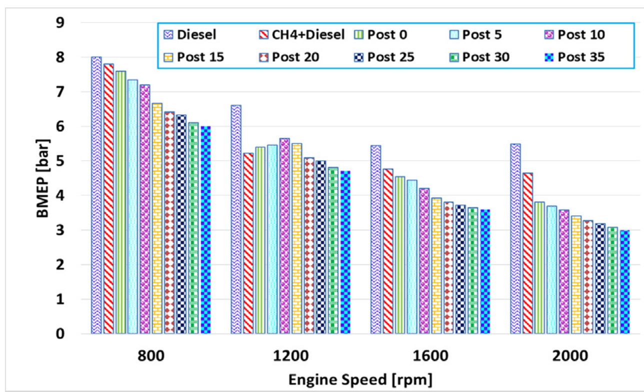

Fig. 10 Brake mean effective pressure variation of post-injection at different engine speeds.

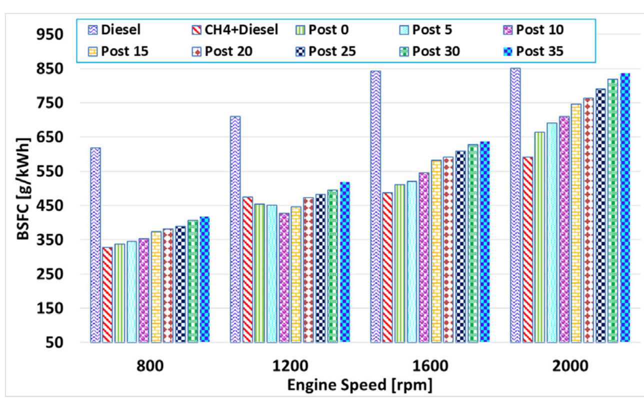

Fig. 11 BSFC variation of post-injection at different engine speeds.

NOx emissions slightly increased at 1600 rpm and 2000 rpm compared to other injection timings. The lowest NOx emission rate 57.34% compared to diesel fuel was obtained with post15 at 2000 rpm engine speed.

The soot emissions for the various injection applications at diverse engine speeds are presented in Fig. 13. In RCCI mode, soot emission slightly increased with the addition of methane to diesel fuel due to EGR. Significant reductions in soot emissions occurred in the post-injection strategies compared to the CH_4/diesel study. The best improvement in soot emissions with the Post35 strategy at 2000 rpm was 58.85% compared to diesel fuel and 62.63% compared to CH_4/diesel fuel. However, after significant reductions in soot emissions at 800 rpm engine speed, a slight increase occurred in the post30 and post35 injection strategies. Because at low engine speeds and lowering the average in-cylinder temperature owing to lower diesel in the main injection with late post injection strategies can exhibit incomplete combustion [18].

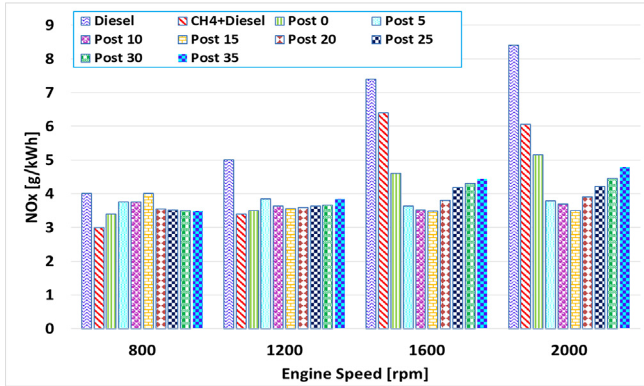

Fig. 12 NOx concentration variation of post-injection at different engine speeds.

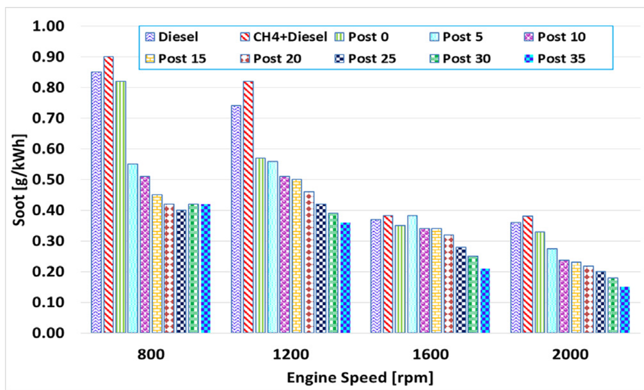

Fig. 13 Soot concentration variation of post-injection at various engine speeds.

7. Conclusions

The effects of the post-injection strategy on emissions, performance, and the combustion properties of a DF mode in an RCCI engine were examined at fixed CH_4 mass flow rates. The main conclusions were itemized as follows:

- The peak in-cylinder pressure of diesel fuel was higher than in methane/diesel DF mode. Besides, the position of the HRR curve replaced the expansion stroke with post-injection strategies compared to single injection strategies of diesel and CH_4/diesel.
- The NOx emissions indicate that NOx values reduced at all engine speeds with delayed diesel post-injection timing. NOx declined even more violently with EGR due to the rarefaction effect. The maximum reduction in NOx emission was 57.34% at 2000 rpm with the post15-injection.
- The soot emission increased slightly with the effect of both the dual fuel mode and the EGR. However, the Soot emission re-entered the downward trend with the delay of the post-injection. The soot emission decreased by 58.85% with the Post35 injection strategy at 2000-rpm engine speed compared to diesel one injection application.
- The combination of EGR, RCCI mode, and post-injection strategies concluded in significant improvements in emissions.
- Methane gas fuel can be used in diesel generators or long-range heavy-duty commercial diesel vehicles to gain advantages in terms of cost and compliance with new emissions regulations.
- The effects of high EGR and post-injection in the RCCI combustion mode can guide further studies on clean gas fuels such as methane, which have become more important in recent years.

Declaration of Competing Interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

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