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Integration of RCCI and CDF Combustion in Conventional Diesel Engine Using CNG-diesel Fuels: An Experimental Study

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ABSTRACT

An experimental investigation is carried out on conventional compression ignition engines' combustion, performance, and emission characteristics using Conventional Dual Fuel (CDF) combustion and reactivity-controlled compression ignition (RCCI) combustion strategies. The experiments are performed on a variable-speed production-grade diesel engine converted to a research engine. Comparative combustion analysis shows that RCCI combustion is more stable and shows a consistent ignition delay across all engine speeds. The exhaust gas temperature of RCCI combustion is lower than that of CDF combustion and is in the range of Conventional Diesel Combustion (CDC). CDC shows better brake thermal efficiency (BTE) than CDF and RCCI combustion across all engine speeds, followed by RCCI combustion. The lowest BTE is observed in CDF combustion. Emission results show that RCCI combustion produced significantly lower NO_x emissions than CDC at low engine speed without much HC and CO emissions increment. RCCI combustion does not effectively reduce NO_x emissions and produces higher HC and CO emissions at high engine speeds. A BTE-NO_x trade-off analysis is also carried out, demonstrating the suitability of RCCI combustion at low engine speed. CDC under high engine speed conditions and in the transition region of medium engine speed CDF combustion is more favorable to reduce NO_x emission.

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1. INTRODUCTION

To Facilitate the adoption of CNG retrofit kits in BS-VI diesel vehicles and substitute diesel engines with CNG counterparts in cars, SUVs, and small cargo vehicles, the ministry road transport (notification dated 11, August 2022) has issued standards for their type approval and emission regulations. These regulations pertain to all vehicles with a gross vehicle weight (GVW) under 3.5 tonnes. In line with this initiative, the Indian government aims to significantly increase the share of CNG in the India's' primary energy mix from the current 6.5% to 15% by 2030. These reforms are intended to facilitate the expansion of natural gas consumption and contribute to the government's emission reduction and net-zero targets.

Moreover, the Bureau of Indian Standards (BIS) has established specifications (IS 17314:2019) for Hydrogen-enriched CNG (H-CNG) as an automotive fuel. This move demonstrates the government's commitment to exploring CNG as an alternative fuel source. In September 2020, the Ministry published a notification (GSR 585 (E)) amending the Central Motor Vehicles Rules 1989 to include H-CNG as an automotive fuel. These proposed modifications aim to promote the adoption of cleaner fuels and reduce vehicle emissions. With these policies in mind, there is scope for diesel-CNG dual-fuel engines. Diesel engines are preferred for power generation and transportation applications due to their high fuel efficiency. They may likely remain in use for commercial transportation for many years. Diesel-CNG dual-fuel engines offer a potential solution by combining the advantages of diesel engines with the cleaner burning characteristics of CNG. This approach allows for a gradual shift towards cleaner technologies while capitalizing on the existing infrastructure and expertise in diesel-based applications. It is worth noting that adopting diesel-CNG dual-fuel engines required further research, development, and investment to ensure optimal performance, emissions control, and compatibility with existing diesel infrastructure.

In normal CI engine combustion, shown in Fig. 1, diesel is sprayed in a combustion chamber at high pressure

NOMENCLATURE

CNG	Compressed Natural Gas	*CI*	Compression Ignition
m_{CNG}	mass flow of CNG	NO_x	Nitrogen Oxides
m_{diesel}	mass flow of diesel	*RCCI*	Reactivity-Controlled Compression Ignition
E_{CNG}	energy share of CNG	*PCCI*	Premixed Charged Compression Ignition
BTE	Brake Thermal Efficiency	*SoI*	Start of Ignition
HRR	Heat Release Rate	*TDC*	Top Dead Center
HCCI	Homogeneous Charged Compression Ignition	*bTDC*	Before Top Dead Center
CDF	Conventional Dual Fuel	*BSFC*	Brake-Specific Fuel Consumption
CDC	Conventional Diesel Combustion		

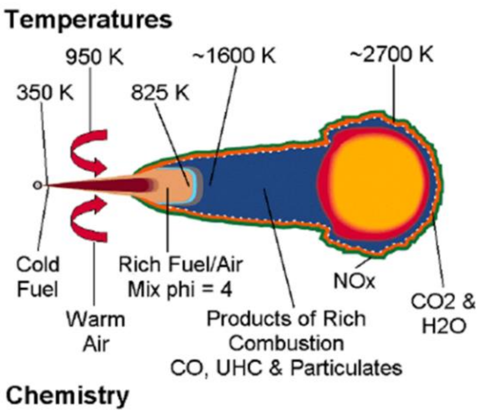


Fig. 1 Combustion of diesel spray inside engine (Pitsch et al., 1996)

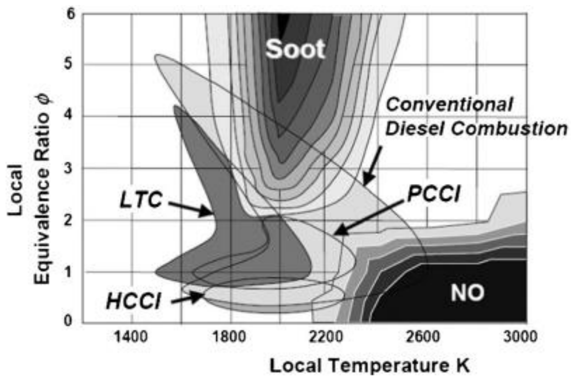
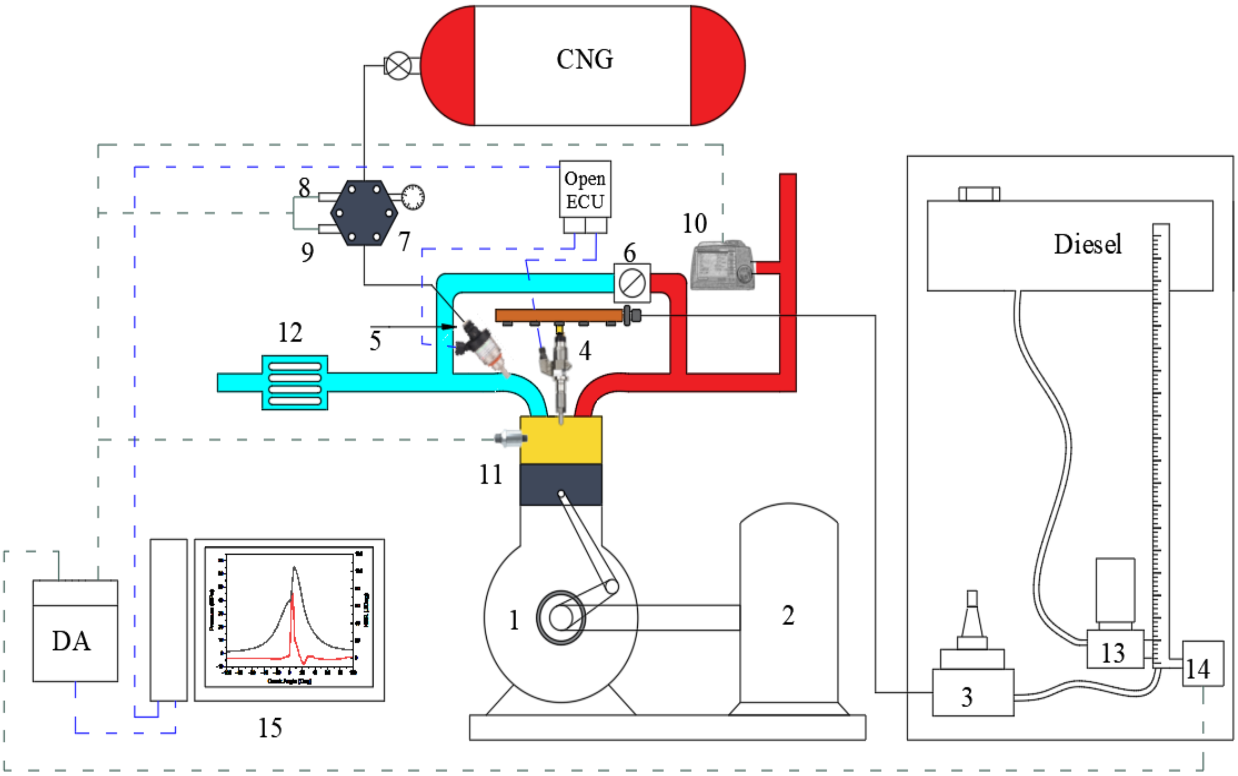


Fig. 2 Equivalence ratio vs. Temperature graph for different modes of combustion (Neely et al., 2005)

(100 to 1000 bar) (Musculus et al., 2013). During combustion in the local rich mixture zone, soot is formed. NO_x is formed when the soot mixes with air and starts oxidation in the outer lean zone. Due to this type of combustion occurring at both high and low equivalence ratios, a high amount of NO_x and soot are formed (Pitsch et al., 1996; Flynn et al., 1999). Suppose the entire combustion happens at a low equivalence ratio and low temperature, as shown in Fig. 2. In that case, the NO_x and soot emissions can be reduced. This phenomenon is the basis of Low-Temperature Combustion (LTC) in diesel engines (Zheng et al., 2009). RCCI stands out as one of the most notable LTC strategies, offering the capability to regulate the combustion process effectively. It holds the promise of curbing fuel consumption and emissions without relying on after-treatment methods. RCCI optimizes the combustion processes of Homogeneous Charge Compression Ignition (HCCI) and Premixed Charge Compression Ignition (PCCI) in diesel engines to decrease emissions. In RCCI, fuel types, relative fuel amounts, and injection timing are carefully selected to tailor combustion for optimal fuel efficiency at low temperatures (for NO_x reduction) while maintaining low equivalence ratios (for soot reduction). Optimized combustion phasing can be achieved by employing two fuels with varying reactivities and employing multiple fuel injections, thereby controlling fuel reactivity within the cylinder. Merts et al. (2022) emphasized the significance of implementing a unified control strategy for initiating combustion. In the context of CDF combustion, the ignitability island is approached from the rich side, whereas for RCCI, it must be approached from the lean side. These relationships are integrated into a single engine controller, accommodating both RCCI and CDF modes. A comparison study examining the combustion, performance, and emissions attributes of engines operating in LTC mode, specifically RCCI, demonstrated that both dual-fuel combustion and RCCI combustion with a higher fraction of low-reactivity fuel exhibit relatively stable combustion and reduced NOx and soot emissions. (Kokjohn et al., 2011; Duraisamy et al., 2020; Pan et al., 2020; Singh et al., 2020; Wategave et al., 2021a). Fajri et al. (2017) considered several control parameters like engine load and speed, varying fuel ratio, equivalence ratio, injection timing, EGR, boost pressure and injection pressure to discover the most influential parameter on performance and emission of the RCCI engine.

Researchers used a combination of various alternative fuels such as CNG, ethanol, methanol, and dimethyl ether as the low-reactive fuels and diesel or biodiesel as the high-reactive fuels, giving an excellent opportunity to operate the engine in the RCCI combustion range (Dahodwala et al., 2014; Dempsey et al., 2013; Kavuri et al., 2016; Jia & Denbratt, 2018). Martin et al. (2018) used propane along with dimethyl ether as low-reactive fuel and ultra-low-sulphur diesel as high-reactive fuel for RCCI operation. The findings of the experiment are that RCCI can improve thermal efficiency, reduce NO_x, and soot emissions at the expense of degradations in peak pressure, max heat release rate (HRR), and HC emissions. conducted a numerical investigation into RCCI combustion using methanol/diesel blends. The findings indicated that RCCI combustion effectively decreases HC and soot emissions while enhancing fuel economy compared to dual-fuel combustion. Işık & Aydın (2016)



1. Diesle engine
2. Eddy current dynamometer
3. Fuel pump
4. Diesel injector
5. CNG injector
6. EGR valve
7. CNG regulator with cylinder pressure gauge
8. CNG low-pressure sensor
9. CNG temperature sensor
10. Emission analyser
11. Combustion pressure sensor
12. Air flow sensor
13. Solenoid valve
14. Fuel pressure head sensor
15. PC for data collection and analysis

Fig. 3 Schematic diagram of the set-up

found that RCCI operation with ethanol and safflower biodiesel blend increased brake specific fuel consumption (BSFC) and decreased NO_x emissions. Bhagatwala et al. (2015) conducted numerical simulations under RCCI conditions using a primary reference fuel mixture of n-heptane and iso-octane. The research revealed that a higher concentration of n-heptane promotes superior flame propagation. Conversely, a lower concentration of n-heptane led to the formation of spontaneous ignition fronts. Li et al. (2015) explored the effect of reactivity gradient in RCCI combustion with gasoline and diesel as primary and secondary fuels. The findings indicated that an increased reactivity gradient can delay ignition timing, decrease the HRR, and mitigate the rate of peak pressure rise. Additionally, it contributes to reduced emissions of NOx and soot in dual fuel combustion. Among these fuels, CNG has emerged as an option that is sustainable and can be easily integrated with diesel for RCCI combustion (Aydın, 2021; Biswas et al., 2022; Singh et al., 2023; Wategave et al., 2021b). CNG can be easily integrated into RCCI and conventional dual-fuel combustion techniques as an established alternative fuel.

This study is the first to compare the CDF and RCCI combustion scenarios against the CDC at different engine speeds. Unlike prior research, which has mainly focused on studying individual combustion modes in isolation, our work demonstrates the comparison between different modes of combustion phasing across different engine speeds. This focus on comparison between both CDF and RCCI combustion modes will mark significant advancements in understanding the performance of alternative fuel-based IC engines.

2. Experimental Setup and Methodology

2.1 Test Setup

The experiments are carried out on a variable-speed Compression Ignition (CI) engine test bench, as illustrated in Fig. 3. Diesel is introduced through a Common Rail Direct Injection injector. At the same time, CNG is introduced into the port, entering the cylinder as a homogeneous mixture with air. The engine utilized in the present research is a production-grade automotive engine; its specifications are outlined in Table 1. The engine is connected to a Technomech TME-10 eddy current dynamometer to measure torque and speed. The uncertainty in the dynamometer measurement is 0.25% of full scale. Combustion analysis is performed using EngineScan, a data acquisition system based on NI LabVIEW. The setup included a PCB 113B22 combustion pressure sensor (accuracy 0.8% of full scale), which is a dynamic pressure sensor designed with naturally piezoelectric, stable quartz sensing elements. These sensors are specifically tailored for measuring rapidly

Table 1 Engine specifications

Description	Specifications
Make	Mahindra and Mahindra
Engine Capacity (cc)	625, single cylinder
Type	Multi-speed, 4-stroke
Compression Ratio	18:1
Max. Power @ RPM	9 HP @ 3000 RPM
Max. Torque @ RPM	30 Nm @ 1800 RPM

fluctuating pressures across a broad range of amplitudes and frequencies. The rotary encoder Autonics E50S8 provides crankshaft angle measurement for combustion analysis having measurement accuracy of 1%. An instrumented panel with a Dawyer 628 fuel pressure transmitter (accuracy 1% of full scale), a Uflow DAN14Z solenoid valve, and a burette arrangement are employed to quantify physical fuel consumption. The intake airflow measurement system comprises of an HFM Type T-MAF sensor, to measure the intake air flow rate in kg/hr with an accuracy of 4% of full scale. Exhaust gas analysis uses an ARAI-approved Mars multi-gas analyzer to measure CO, CO_2, HC, NO_x, and O_2 concentrations. The measurement accuracy of the multi gas analyzer device is 4% of the full scale.

2.2 Experimental Procedure

Figure 4 demonstrates different fuel injection strategies used for various combustion modes in the present study. For baseline CI combustion, diesel serves as the primary fuel. However, for dual fuel and RCCI combustion, CNG is used as the low-reactive fuel, and diesel is used as the high-reactive fuel. RCCI combustion tends to provide best result at low load conditions. In present study all experiments are conducted under a constant engine load of 25% over five different engine speeds ranging from 1000 to 3000 RPM. For each combustion mode, the fuel injection pressure and the start of injection timing are maintained constant at 500 bar and 10 bTDC, respectively. The proportion of CNG energy varies with engine speed to substitute for the quantity of diesel fuel. The test conditions are outlined in Table 2. The calculation of the CNG energy share is determined by the equation 1 (Jamrozik et al., 2019).

$$E_{CNG} = \frac{m_{CNG} * LHV_{CNG}}{m_{CNG} * LHV_{CNG} + m_{diesel} * LHV_{diesel}} * 100 \quad (1)$$

Table 2 CNG energy share for dual fuel and RCCI combustion at different engine speeds

Engine Speed	CNG energy share
1000	65%
1500	80%
2000	85%
2500	75%
3000	75%

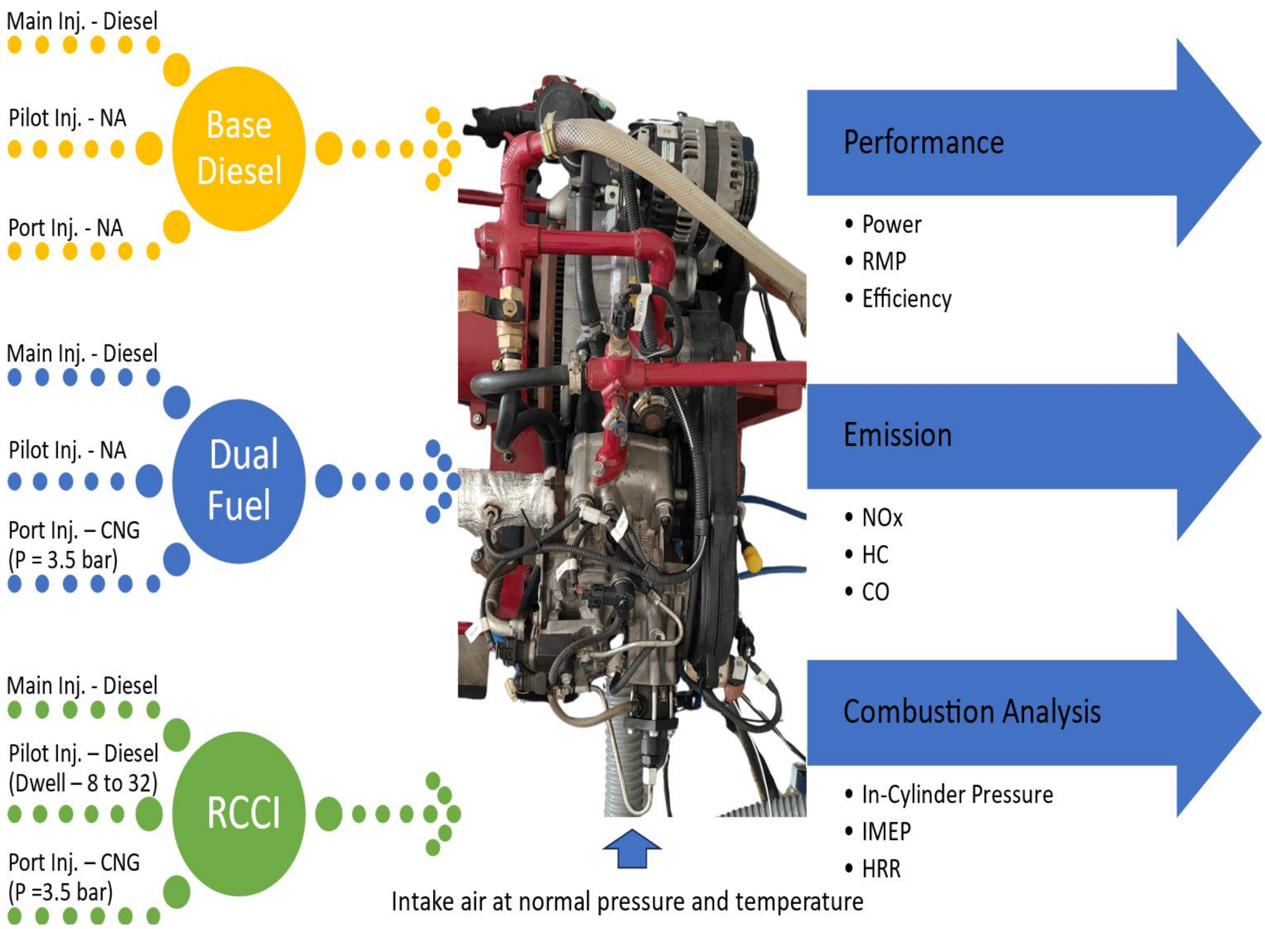


Fig. 4 Fuel injection strategy for different combustion modes

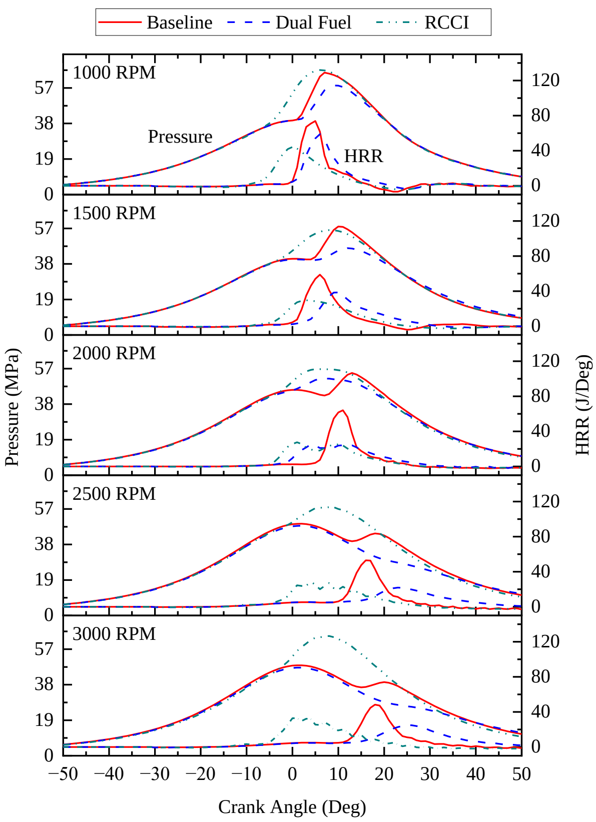


Fig. 5 Variation of pressure and HRR with crank angle for different combustion modes

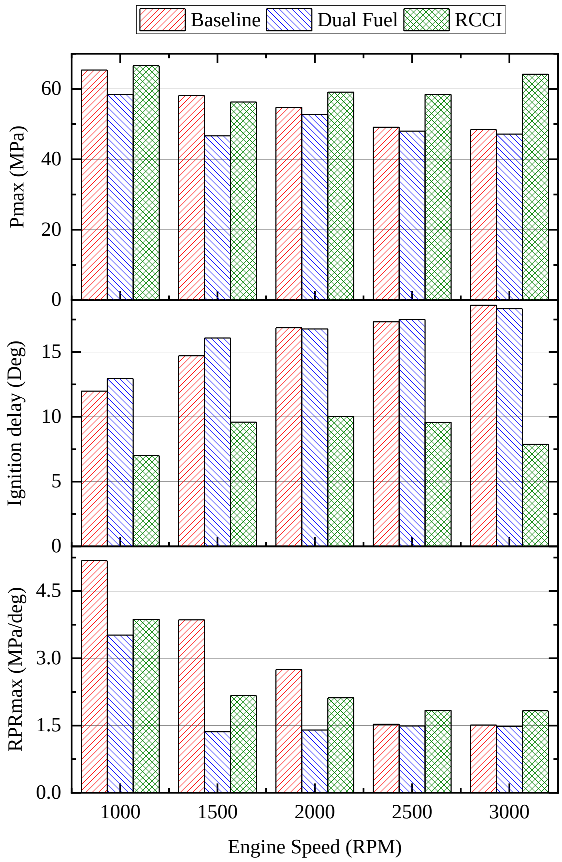


Fig. 6 Variation of maximum pressure, ignition delay and maximum rate of pressure rise with engine speed for different combustion modes

3. Results and Discussion

This section provides an in-depth discussion on the performance of CDC, dual-fuel combustion, and RCCI combustion. The present section has discussed the effects of combustion strategy on performance, such as BSFC and BTE. Engine emission performance and its sensitivity to combustion strategy are also discussed in this paper.

3.1 Combustion Analysis

Combustion performance is evaluated based on cylinder pressure and HRR. The heat release diagram allows for the estimation of combustion duration and intensity. This diagram proves particularly valuable in understanding the initial combustion phase during which most NO_x are typically generated. The calculation of the HRR involves applying the principles of the first law of thermodynamics to every recorded cylinder pressure point, as outlined in Equation 2 (Heywood, 2018).

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} - \frac{1}{\gamma-1} V \frac{dp}{d\theta} \tag{2}$$

Figure 5 shows the pressure and HRR variation with crank angle for all combustion modes. Compared to baseline combustion, dual-fuel combustion typically exhibits a delayed peak HRR. In RCCI combustion, the peak HRR is earlier than baseline CDC because of the higher reactivity of the air-fuel mixture at the start of the main injection. At lower engine speed, extended time allows for adequate diesel spray penetrating in the CNG-air mixture, resulting in favorable combustion across all modes. However, at 1500 RPM, dual fuel combustion experiences high ignition delay and lower peak pressure. RCCI combustion does not experience ignition delays as high as dual fuel combustion, even with CNG as the primary fuel (Merts et al., 2021). These trends emphasize the impact of injection timing and fuel characteristics on combustion behavior in different modes and engine speeds.

Above 2000 RPM, based on HRR graph it is observed both dual fuel and RCCI combustion modes exhibit a distinctive two-stage combustion phenomenon. It is observed that CNG initiates combustion before the main injection in dual fuel combustion. The early combustion phase indicates a pre-mixture of CNG with air, influencing the overall combustion process. The previous studies have shown that homogenization decrease the local equivalence ratio and fuel-rich regions, which would allow the mixture auto-ignition under lean mixture conditions (Pan et al., 2021). At higher engine speeds, the combustion starts late, and a rise in combustion pressure is observed at 15 degrees aTDC. This late combustion leads to incomplete combustion and higher unburnt hydrocarbon emissions.

Figure 6 shows a variation of maximum pressure, ignition delay and maximum rate of pressure rise for different engine speeds. At high RPM, high peak pressure is maintained in RCCI combustion. Mixing diesel with CNG-air mixture before the main diesel injection increases the overall reactivity of the fuel. In baseline CI

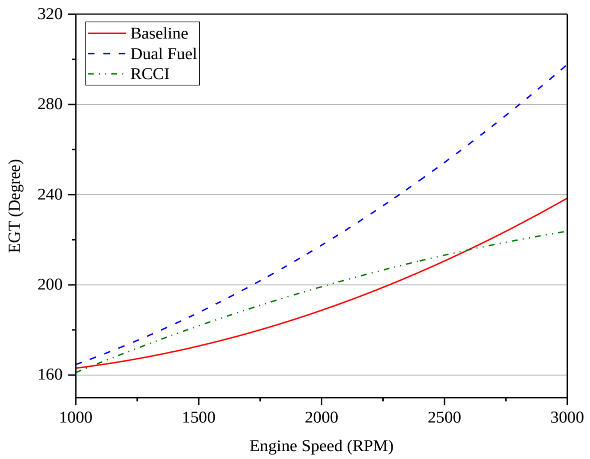


Fig. 7 Effect of different combustion modes on exhaust gas temperature

and dual fuel combustion, peak pressure decreases with increasing RPM due to higher ignition delay, resulting in delayed peak pressure. However, for RCCI combustion, peak pressure variation is minimal across different engine speeds, with RCCI exhibiting the highest P_{max} at all speeds. Ignition delay increases with engine speed for CI and dual fuel combustion. Due to elevated in-cylinder temperatures, ignition delay is generally reduced. However, when measured in crank angle, ignition delay increases with higher engine speed. For RCCI combustion, the ignition delay is reduced at higher engine speeds. The maximum rate of pressure rise serves as an indicator of diffused combustion. The lower values of RPR_{max} indicate a higher percentage of fuel burned under diffused combustion. Dual fuel combustion consistently displays the lowest RPR_{max} across all engine speeds.

Figure 7 demonstrates the variation of exhaust gas temperature (EGT) at different engine speeds for various combustion modes considered in the present study. The EGT trends offer valuable insights into the combustion characteristics of each mode, highlighting differences in temperature profiles and combustion efficiency across various engine speeds. EGT serves as a reliable indicator of average in-cylinder temperature. EGT in dual fuel combustion with CNG is higher than diesel combustion through all engine speeds. This is because CNG has a higher heating value than diesel. As the flame speed of CNG is lower than diesel, the combustion could happen till the end of the power stroke.(Yusaf et al., 2010) This also could lead to higher EGT. The EGT in RCCI combustion is lower in the diesel combustion with CNG. Interestingly, at maximum engine speed, RCCI combustion showcases the lowest temperature, signifying successful low-temperature combustion.

Figure 8 shows the variation of CA50 (representing the crank angle position corresponding to 50% of cumulative heat release) with engine speed. CA50 is a crucial parameter for combustion analysis. Lower CA50 values indicate rapid combustion leading to knocking. In comparison, higher CA50 values suggest incomplete combustion, resulting in elevated HC and CO emissions (Husted et al., 2007). Notably, CA50 is maximized for

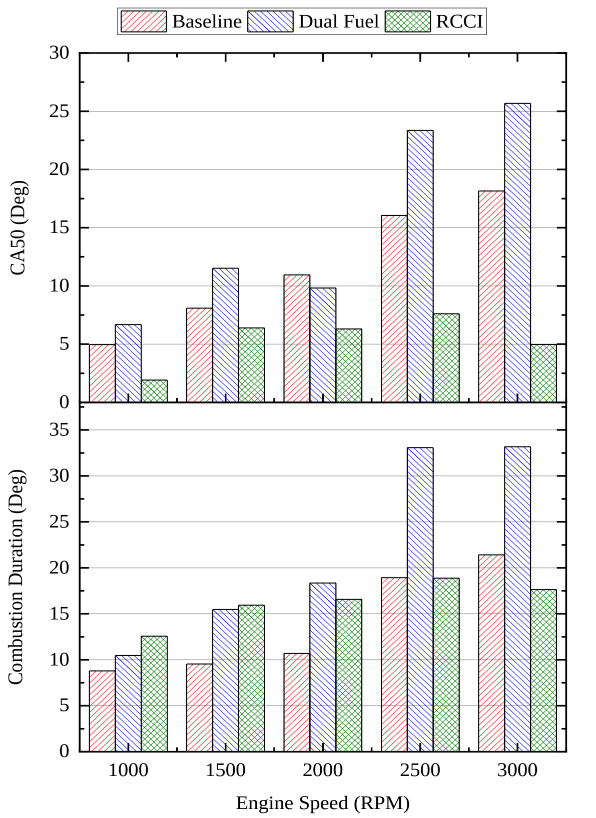


Fig. 8 CA50 and combustion duration at different engine speeds

dual fuel combustion across all engine speeds, except at 2000 RPM, where baseline CI combustion exhibits slightly higher CA50 than dual fuel combustion. In contrast to CDC and dual fuel combustion, it is observed that CA50 for RCCI combustion initially increases at lower to medium engine speeds and then decreases at higher speeds. At low speed the temperature in the combustion chamber is lower, in RCCI combustion earlier pilot injection in a cooler cylinder, have a longer mixing time resulting in a late CA50 but as engine speed increases the temperature in the combustion chamber increases resulting in reduction in CA50 angle (Merts et al., 2021). Compared to other combustion modes, the variation in CA50 is less pronounced in RCCI combustion, indicating stable and optimized combustion in RCCI mode. Most of the studies show that CA50 increases with RCCI combustion, this is because the blending of LRF could increase the ignition delay (Hanson et al., 2010; Benajes et al., 2014). But there are some studies that shows that CA50 reduces with RCCI combustion (Li et al., 2013). This contradiction can be due to the different engine specifications, SoI timings, and the properties of fuels.

Combustion duration, measured as the crank angle difference between the start of ignition estimated by CA10 and the end of combustion estimated by CA90 (Bai-Gang et al., 2014), demonstrates notable trends across different combustion modes and engine speeds. Combustion duration consistently increases with speed in all combustion modes. At low speeds, RCCI combustion exhibits the highest combustion duration. However, as engine speed increases, dual fuel combustion experiences

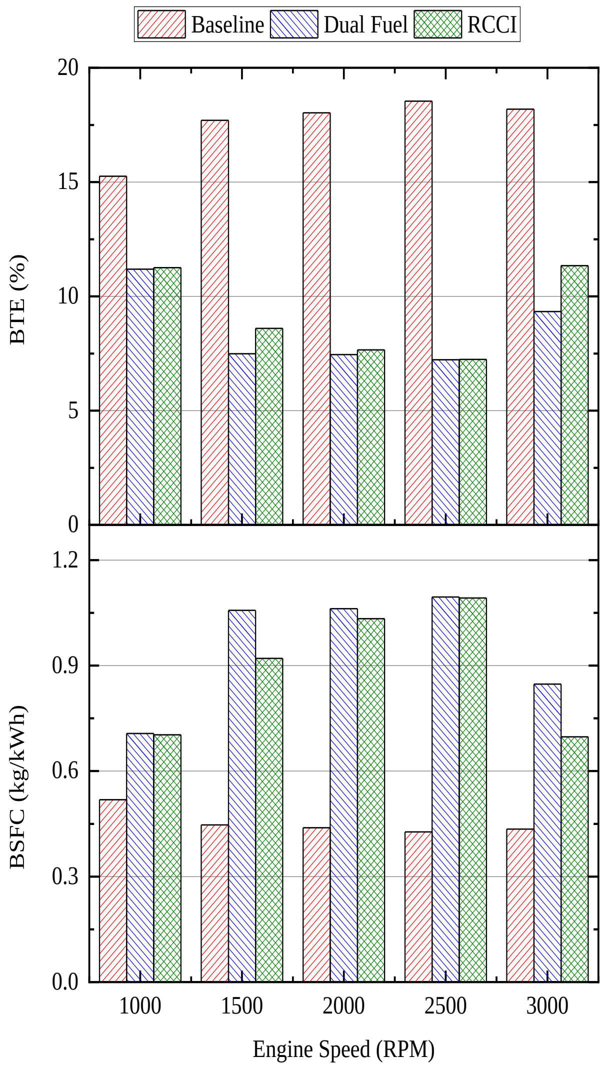


Fig. 9 BTE and BSFC at different engine speeds

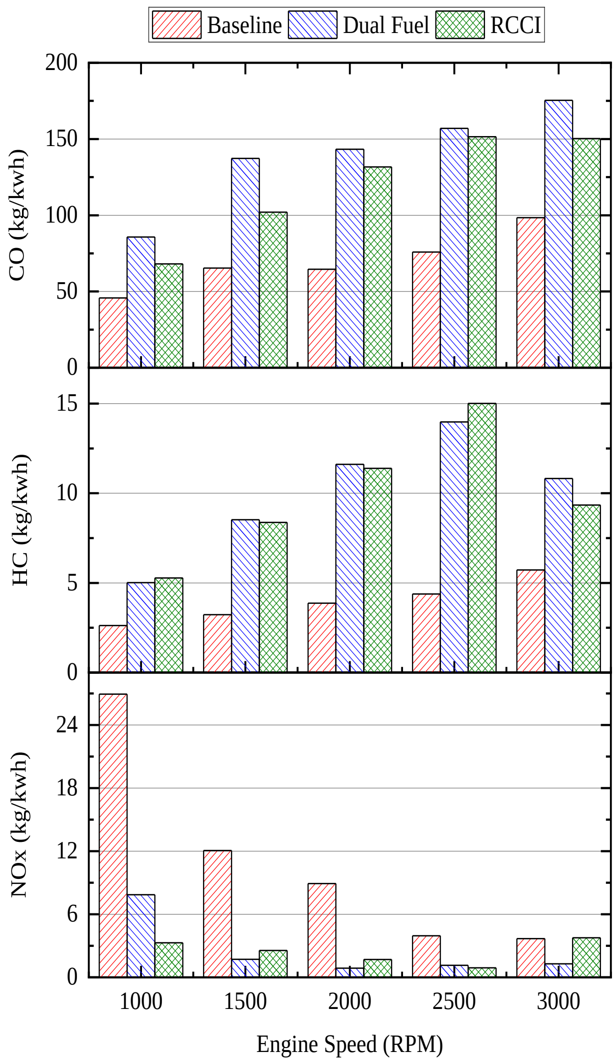


Fig. 10 Concentrations of CO, HC and NO_x emissions at different engine speeds for different combustion modes

a rapid increase in combustion duration due to the low reactivity of CNG. Mixing pilot diesel with CNG-enriched air in RCCI combustion enhances combustion kinetics, preventing a significant increase in combustion duration compared to dual fuel mode. At maximum engine speed, RCCI combustion duration is even less than baseline combustion, attributed to more intense in-cylinder conditions resulting from pilot diesel injection. These findings emphasize the impact of low reactivity fuel CNG on combustion duration.

3.2 Engine Performance

The BTE and BSFC are compared for different combustion modes in performance analysis. For dual fuel and RCCI combustion, BTE and BSFC are calculated using equations 3 and 4, respectively (Chen et al., 2021).

$$BTE = \frac{BP}{m_{CNG}*LHV_{CNG}+m_{diesel}*LHV_{diesel}} \quad (3)$$

$$BSFC = \frac{(m_{CNG}*LHV_{CNG}+m_{diesel}*LHV_{diesel})/LHV_{diesel}}{BP} \quad (4)$$

Figure 9 shows the results of BTE and BSFC obtained experimentally at different engine speeds for different combustion modes. Figure 9 reveals that BTE is higher at all engine speeds for the baseline combustion case than the dual fuel and RCCI combustion modes. Higher magnitudes of BTE in baseline combustions are due to better combustion, resulting in higher BTE. In dual fuel, the supply of a low-reactive CNG fuel reduces neat heat release, resulting in more unburnt fuel in exhaust gases. Consequently, the BTE is low. However, RCCI combustion improves BTE compared to conventional dual-fuel combustion mode at all engine speeds. It is observed from Fig. 9 that as the engine speed increases, the difference in BTE between baseline CI combustion and dual-fuel combustion increases further, which indicates that dual-fuel combustion mode is not suitable for high engine speeds.

It is depicted in Fig. 9 that BSFC is low for baseline combustion and not altered much by engine speed. Lower BSFC in baseline combustion is due to the complete combustion of fuel. In dual-fuel combustion, BSFC is higher and increases with engine speed. The BSFC reduces for dual fuel and RCCI combustion at maximum engine speed. The RCCI combustion mode shows better

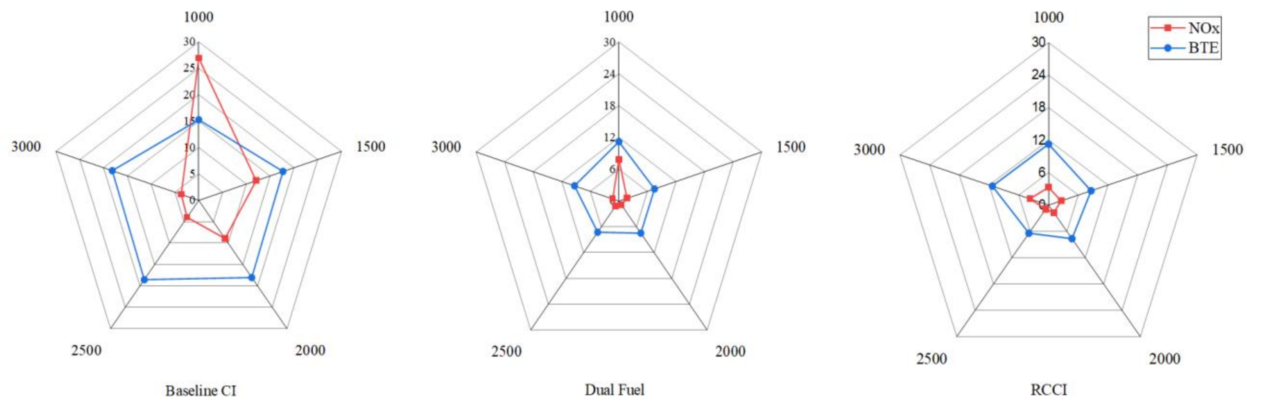


Fig. 11 Trade-off for BTE and NO_x with engine speed

in BTE and BSFC than conventional dual-fuel combustion at all engine speeds.

3.3 Emission

Figure 10 shows experimental results of CO, HC and NO_x emissions obtained at different engine speeds for baseline CI, dual fuel and RCCI combustion modes. The emission values are calculated from the readings obtained from gas analyzer with equation 5.

$$Emission\left(\frac{kg}{kWh}\right) = \frac{u * Emission * (m_{air} + m_{fuel})}{BP} \quad (5)$$

where:

Emission	Coefficient (u)	Measured unit
NO_x	0.001587	ppm
CO	0.000966	ppm
HC	0.000479	ppm

Figure 8 depicts that CO and HC emissions increase with engine speed increase across all combustion modes. At high speed, the time available for combustion is less, and combustion duration also increases, leading to incomplete combustion. Compared to baseline CI combustion, dual fuel and RCCI combustion produce significantly higher HC and CO emissions. As CNG reduces the reactivity of the charge inside the combustion chamber, the fuel remains unburnt at the end of combustion. When dual fuel and RCCI combustion are compared, the RCCI combustion produces fewer CO and HC emissions. In baseline CI combustion, the HC and CO emissions do not vary significantly with engine speed compared to dual fuel and RCCI combustion.

On the other hand, NO_x concentrations reduce with engine speed across all combustion modes. The ignition delay is more significant at higher engine speeds, leading to more defused combustion. In diffused combustion, there are fewer hot spots. This leads to a reduction in thermal NO_x formation. Dual fuel and RCCI combustion significantly reduce the NO_x emission compared to baseline CI combustion at lower and medium engine speeds. RCCI combustion produces the lowest NO_x emission at 2500 RPM. Overall, RCCI combustion does not show significant variation in NO_x emission with engine speed.

RCCI combustion has low operation range with higher ratio of LRF/HRF. To extend the operation range the LRF/HRF ratio needs to be substantially reduced (Wang et al., 2016). This initiates the transition of combustion from RCCI to CDC. The operation range of engine for different modes of combustion can be further explored with trad off between BTE and NO_x. Figure 11 shows the trade-off for BTE and NO_x with engine speed for baseline CI, dual fuel combustion and RCCI combustion modes. As shown in figure 11, two engine operating zones are identified. They are low-efficiency high-emission zones and high-efficiency low-emission zones. At 1000 RPM in the baseline combustion, the NO_x polygon goes outside the BTE polygon, which can be considered a low-efficiency high-emission zone. Meanwhile, in dual fuel and RCCI combustion, the NOx polygon remains inside the BTE polygon, and at 1000 rpm, RCCI combustion shows high efficiency and low emission. As the speed increases, the efficiency of RCCI combustion reduces, and at medium engine speeds of 1500 to 2000 RPM, dual fuel combustion shows better BTE and NO_x trade-off than the RCCI and baseline combustion. At maximum engine speed, the engine operated best with baseline Conventional Diesel Combustion.

CONCLUSION

Here are the conclusions drawn from the current experimental investigation.

1. As CA50 is uniform at different engine speeds for RCCI combustion, it is more uniform and stable under all engine speeds than CI and dual fuel combustion. Dual fuel combustion has maximum ignition delay, which leads to incomplete combustion at all engine speeds. RPRmax is very high at low engine speed for baseline combustion, indicating rapid HRR and engine knock.

2. CO and HC emissions are low for baseline CI combustion. In contrast, for dual fuel and RCCI combustion modes, the CO and HC emissions are high and increase with engine speed. As compared to baseline combustion, NO_x concentration levels are lower for dual fuel and RCCI combustion. RCCI combustion shows a significant reduction in NO_x emission at low engine speeds.

3. BTE is higher for baseline CI combustion and increases slightly with engine speed. The RCCI combustion gives better thermal efficiency than dual fuel combustion across all engine speeds.
4. The trade-off analysis shows that only at low engine speeds, RCCI combustion mode performs better in terms of efficiency and emissions compared to dual fuel and baseline CI combustion modes.

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CONFLICT OF INTEREST

The authors declare that they have no conflict of interest.

AUTHORS CONTRIBUTION

Jay Chhatbar: Investigation, Methodology, Data curation, Writing – original draft, **Parag Rajpara**: Conceptualization, Supervision, Funding acquisition, Writing – review & editing, **Rahul Banerjee**: Conceptualization, Funding acquisition, Methodology, Supervision, **Srijit Biswas**: Conceptualization, Methodology, Writing – review & editing.

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