EFFECT OF VARIATION OF FUEL INJECTION TIMING ON PERFORMANCE OF A LIQUIFIED PETROLEUM GAS FUELLED REACTIVITY-CONTROLLED COMPRESSION IGNITION ENGINE

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Abstract: *This study report examines the utilization of a Liquefied Petroleum Gas-diesel dual-fuel Reactivity Controlled Compression Ignition engine, demonstrating significant enhancements in combustion efficiency and decreases in NOx and particulate emissions. The impact of liquefied petroleum gas on the operation, combustion, and exhaust pollutant attributes of a diesel-powered engine operating in reactivity-controlled compression ignition mode was investigated. Findings demonstrate improved thermal efficiency and reduced emissions relative to traditional diesel engines. This study's innovation resides in employing LPG as a low-reactivity fuel in an RCCI engine, a subject that has been less investigated relative to other gaseous fuels. This surpasses prior initiatives by enhancing dual-fuel techniques to attain reduced emissions and increased efficiency across diverse load scenarios. The eddy current dynamometer was employed for loading purposes. The Nira7i software was utilized to manage operations and gather statistics. The engine underwent testing at a steady speed of 1500 rpm under a load of 3.5 kW utilizing an eddy current dynamometer. The experimental results reveal that for the RCCI engine, the indicated power increases by 50.97%, brake power rises by 12.99%, and brake thermal efficiency improves by 63.21% compared to conventional diesel combustion. Carbon monoxide and hydrocarbon emissions rise by 262.50% and 606.25%, respectively, although Nitrogen oxide and smoke emissions decline by 95.66% and 73.29%. The cylinder pressure is reduced by 27.48%, Net heat release rate by 25.88% when compared to a conventional diesel combustion engine utilizing liquefied petroleum gas energy supply.*

Keywords: **IC engine; RCCI engine; Emission; LPG; Combustion.**

1. INTRODUCTION

Internal combustion engines find application across a range of domains such as road transport, rail transport, and sea transport. agriculture, and electricity generation. Internal combustion engines (ICEs) have served as a fundamental element in transportation and industrial applications for more than a hundred years. Even with progress in alternative propulsion systems such as battery-electric and hydrogen fuel cell technologies, IC engines remain prevalent in the automotive, aviation, and marine industries. This is largely attributed to their superior energy density, quick refueling options, and a robust existing infrastructure. Nonetheless, the rising global apprehensions regarding green-house gas emissions and the exhaustion of fossil fuel resources demand substantial advancements in the efficiency of IC engines and emissions management [1].

Recently, a lot of work has been put into developing new ways to burn fuel, like homogeneous charge compression ignition (HCCI), reactivity-controlled compression ignition (RCCI), and partially premixed combustion (PPC). The goal is to make combustion more efficient and reduce pollution [2]. Also, new types of fuels like biofuels, synthetic fuels, and hydrogen-enriched blends have shown promise in lowering emissions of carbon dioxide (CO_2) and nitrogen ox-ide (NOx) [3]. Adding hybrid powertrains and waste heat recovery technologies to IC engines makes them even better at meeting strict environmental rules while still providing high power output and operational flexibility [4].

Kokjohn S. et al. [5] explored the applications of gasoline and diesel in RCCI. The outcomes indicate that the RCCI engine is more efficient across a broad spectrum of engine loads, reaching a high point of 56% gross indicated efficiency at 9.3 bar IMEP. Kalsi S.S. and Subramanian K.A. [6] conducted experiments on the RCCI mode that was fuelled by biodiesel-CNG/HCNG. The efficiency improved from 33.2% with a 39% CNG fraction of energy to 34.6% with a 38% HCNG fraction of energy. The efficiency of HCNG is inferior to that of conventional biodiesel when HCNG's fraction of energy share exceeds 69%. Using HCNG reduces CO, HC, and smoke exhaust pollutants compared to traditional CNG-fueled engines. NOx emissions rise moderately for all HCNG energy proportions in comparison to a CNG-fueled engine. RCCI engines that use both CNG and hydrogen have significant issues with HC and CO pollutants. These can be greatly reduced by using hydrogen-blended CNG.

Hodges K.A. et al. [7] investigated the consequence of Propane Energy Fraction (PEF) on diesel-ignition low-temperature combustion. The heightened tendency to knock was noted when PEF was 53% or below. The engine-out UHC and CO pollutants diminished by approximately 75% when PEF de-creased from 90% to 60%. The ideal indicated fuel conversion efficiency (IFCE) was determined to be approximately 48% at 71% PEF. Li J. et al. [8] evaluated the optimal compression ratio (CR) in RCCI engines powered by gasoline and biodiesel. When the effective compression ratio goes up, the ignition delay goes down, the peak pressure goes up, the heat re-lease goes up, and the NO emissions go up. Hunicz J. et al. [9] performed an investigation on RCCI combustion utilizing diesel or high-reactivity fuels. The higher reactivity (cetane number) is a key factor in the benefits that HVO brings to the operation of RCCI. The best DF-NG RCCI has an impressive efficiency of 44.8% and releases 2.25 g/kWh of NOX, 2.8 g/kWh of CO, and 9.9 g/kWh of CH4. The NOX and CO pollutants from an HVO-activated RCCI are well below the EPA's Tier 4 emission standards for mobile equipment that isn't used on the road. The new CDC combustion gets 42.7% thermal efficiency and beats EPA Tier 4 standards by over 200% for PM emissions and over 1500% for NOX emissions.

Habineza et al. [10] conducted a research study on commercial engine generators utilizing syngas. The altered engine demonstrated notable performance, achieving a maxi-mum power output of 3,689 W. The quality of syngas in the gasifier influences the power output. The implementation of syngas engine generators can alleviate environ-mental concerns and reduce electricity expenses and fossil fuel consumption. Ramachandran E. et al. [11] conducted studies to examine the influence of varying energy proportions between CNG and microalgae biodiesel. The main findings show that 30% CNG shares produced a 4.35% higher BTE compared to CBC. HC, CO, CO_2, NOx, and smoke pollutants dropped by 2.2%, 4.7%, 7.2%, 25%, and 31%, respectively, compared to CBC when 30% CNG was used. Mohsen M.J. et al. [12] conducted an examination of the advantages of LPG in CI engines. This decreases expenses and facilitates a rapid transition back to conventional fuel. It enhances blending and ignition, decreases fuel utilization, and optimizes exhaust pollutants characteristics. The injection of LPG markedly enhances engine performance. This creates a uniformly lean combination of air and LPG, thereby decreasing NOx emissions. This facilitates operation at a high compression ratio beneath the knocking threshold, hence improving efficiency. Com-pared to conventional engines, it elevates the cylinder pressure and temperature. This decreases ignition delay in diesel and dual-fuel operations. Elevating the intake charge temperature lowers the amount of HC and CO that is released.

Aydin A. and Aydin H. [13] examined the efficiency, combustion characteristics, and emissions of a diesel engine utilizing LPG fumigation as a supplementary fuel. The results indicated that an elevated LPG ratio resulted in a rise in cylinder pressure, which caused knocking. Exhaust gas temperatures declined by up to 54% with LPG, accompanied by a rise in specific fuel consumption. Nevertheless, minimal utilization of

LPG as a secondary nonreactive fuel was determined to be feasible. Emission testing indicated a 2033% rise in CO and a 1088% rise in HC emissions with the use of LPG. The research indicates that LPG utilization may enhance exhaust emissions and efficiency; nevertheless, additional investigation is required to assess its long-term impacts on engine components and the influence of modern injection systems.
Al-kaabi M.M. et al [14] conducted research on the utilization of LPG, which is accessible at a reduced cost, in diesel engines. Experiments were performed on four fuel modalities, commencing with D-100, followed by LPG-25, LPG-50, and LPG-100. The results indicated that LPG-50 fuel enhanced thermal efficiency, decreased brake specific fuel consumption (BSFC), and lowered emissions across all running modes. The most effective emission reduction was noted with LPG-50.
Singh S. [15] examined the efficiency and emissions of a RCCI engine operating on LPG with 1500 rpm, yielding a maximum power output of 3.7 kW. The findings indicated that BSFC and brake power decreased by 83.14% and 34.65%, respectively. The rise in cooling water temperature was decreased by 5.88% at full load. The exhaust gas temperature decreased by 29.77%, while BTE improved by 19.17%. Smoke opacity decreased by 69.81% at full loading conditions. The results indicate that a RCCI engine may function effectively using LPG, including roughly 95% premixed energy, thereby decreasing the reliance on diesel fuel and minimizing engine emissions.
Previous studies have shown that RCCI combustion using dual fuels such as gasoline-diesel and natural gas-diesel can significantly reduce NOx and particulate emissions while improving thermal efficiency. LPG has also been identified as a clean-burning, readily available fuel with potential for use in dual-fuel systems. But a very Limited studies have explored the use of LPG as a low-reactivity fuel in RCCI mode, particularly in terms of combustion behaviour, engine performance under varied loads, and real-world applicability. Advanced injection strategies, combustion phasing, and fuel blend ratios should be explored to fully harness the benefits of LPG while maintaining engine stability and efficiency.

2. Experimental Setup

The research investigation was performed on a diesel engine with one cylinder transformed into an RCCI engine with a cubic capacity of 661 CC. The specifications of the equipment of the testing engine are presented in Table 1, and the visual representation is illustrated in Figure 3. The LPG gas maintains a constant outlet pressure of 1 kg/cm², with gas injection timings set at 4 ms and 5 ms. At 4 ms, the gas flow rate is 0.3 kg/hr, and at 5 ms, the gas flow rate is 0.5 kg/hr. The gas flow rate was measure with the help of a gas flow meter. The energy ratio for LPG at 4ms and 5ms gas injection time is 84% and 89% respectively. The direct injection fuel pressure was taken 600 bar and the direct injection timing for the diesel was taken as 20° bTDC, 23° bTDC, and 26° bTDC. The diesel fuel injection timing was changed with the help of Nira i7 open ECU control soft-ware with control the timing if CRDI system. The diesel fuel is directly introduced into the cylinder at the separate combinations of the aforementioned parameters. LPG possesses a high calorific value, facilitates combustion, and minimizes emissions. An innovative Electronic Control Unit (ECU) was developed to regulate LPG injection timing and duration, accompanied by the installation of a magnetic sensor on a single-cylinder diesel engine.

This study used diesel as a high-reactivity fuel and liquefied petroleum gas as a low-reactivity fuel. The chemical as well as physical specifications of the fuels employed in this study are delineated in Table 2. The fraction of energy of LPG in the total fuel can be calculated with the help of Equation 1.

$$E_{LPG} = \frac{m_{LPG}\,CV_{LPG}}{m_{LPG}\,CV_{LPG} + m_{diesel}\,CV_{diesel}} \times 100\% \tag{1}$$

where, the LPG mass flow rate is represented by m_{LPG} (kg/s), the diesel fuel mass flow rate is represented by m_{diesel} (kg/s) and the calorific value of LPG and diesel are represented by CV_{LPG}, CV_{diesel} respectively.

Table 1. Specifications of Experimental engine setup.

Description	Value	Unit
Rated speed	1500	rpm
Rated power output	3.7	kW
Compression ratio	17.5	-
Cylinder	1	-
Bore	87.5	cm
Stroke	110	cm
Cubic capacity	661	cm^3
Nozzle opening pressure	200	bar
Injector nozzle diameter	0.19	mm
Aspiration mode	Naturally	-
Cooling method	Water cooled	-

Table 2. Physical and chemical properties of diesel and LPG [16], [17], [18].

Properties	Diesel	LPG	
Molecular formula	$C_{12}H_{24}$	C_4H_{10}(Butane)	C_3H_8 (Propane)
Molecular Weight [g/mol]	150-250	58.12	44.09
Composition [%] Carbon Hydrogen Oxygen	 86.5 13.5 0	 82.66 17.33 0	 81.72 18.28 0
C/H ratio	6.41	4.77	4.47
Cetane Number	40-55	10	5
Octane number	20-30	92	105
Liquid Density (kg/m^3)	840	578	500
Specific gravity	0.83	0.58	0.51
Stoichiometric air fuel ratio	14.5	15.5	15.9
auto-ignition Temperature [K]	523-553	743-823	743-823
Lower Heating Value [KJ/kg]	42500	45700	46300

Table 3. Measurement range and resolution of analyzers.

Parameter	Measurement range	Units	Resolution	Uncertainty (%)
CO	0-10%	vol	0.01%	2.40
HC	0-20,000	ppm vol	1 ppm	2.45
NOx	0-5000	ppm vol	1 ppm	1.20
Smoke	0-100%	Opacity	±1% full scale	2.50
Air flow rate	4.8 -1116	Nm^3/h	±1.5% of flow	1.9
Mass flow of diesel	0.200 -300	Kg/h	±0.2% of rate	0.2
Gas (LPG) flow rate	0-10	Kg/h	±1% of flow	1.3
Brake thermal efficiency	-	-	-	0.58

Figure 3. Pictorial view of experimental setup

3. Results and Discussion

The experimental findings for this paper are presented within this segment with the important performance parameters, combustion parameters and engine pollutants.

3.1. Indicated power

The indicated power (IP) of an RCCI engine mostly relies on the pressure generated within the cylinder during combustion. It escalates when combustion is efficient and well-timed, resulting in elevated peak pressures and optimal work on the piston. It diminishes with delayed combustion, inadequate fuel-air mixing, or reduced in-cylinder temperatures, which lessen the pressure increase and energy conversion efficiency. Critical influencing aspects encompass combustion timing, fuel type and ratio, as well as overall combustion quality. Figure 4 illustrates the comparison of IP between the CDC and RCCI at port fuel injection timings of 4 ms and 5 ms. The results show that the IP has increased for RCCI engine. At 4 ms port injection time, the IP of the RCCI improved by 43.44%, 44.40%, and 50.97%, respectively at direct in-cylinder injection (DI) timing of 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the IP of RCCI increased by 42.26%, 43.71%, and 46.77%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the IP of the RCCI rises by 34.56%, 32.63%, and 35.52%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the RCCI engine's IP increases by 40.32%, 40.65%, and 40.48%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a peak increase in IP by 50.97% and 46.77% at 26˚ bTDC, at port injection timings of 4 ms under part load and maximum load conditions respectively.

3.2. Brake power

The brake power (BP) of an RCCI engine is primarily determined by the quantity of fuel combusted and the efficiency of the combustion process. It escalates when additional fuel is efficiently combusted at the optimal period, producing elevated in-cylinder pressure and torque output. Conversely, it diminishes with incomplete combustion, inadequate air-fuel mixing, or improper combustion timing, which impair the engine’s capacity to transform fuel energy into mechanical effort. Engine load, fuel energy content, and injection strategy substantially influence brake power. Figure 5 illustrates the comparison of BP between the CDC and RCCI at port fuel injection timings of 4 ms and 5 ms. The results show that the BP of RCCI is almost same as CDC engine. At 4 ms port injection time, the BP of the RCCI is marginally changed by 6.78%, 3.39%, and 12.99%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the BP of RCCI is changed by -1.45%, 0.00%, and -2.33%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the BP of the RCCI is changed by 3.39%, 3.39%, and 6.21%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the RCCI engine's BP changed by -2.33%, -0.58%, and -9.30%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a maximum increase in BP by 12.99% at 26˚ bTDC, at port injection timings of 4 ms under part load.

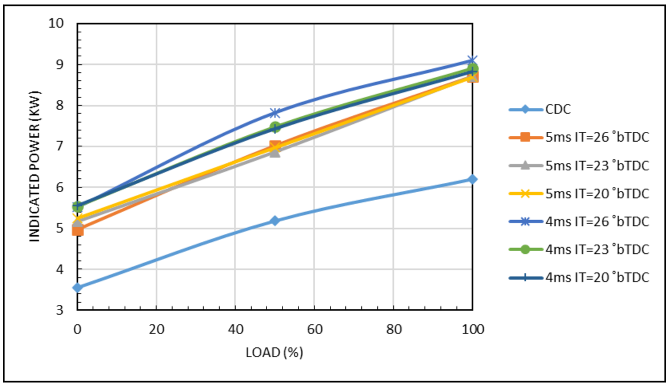

Figure 4. Indicated power vs load at different injection timing.

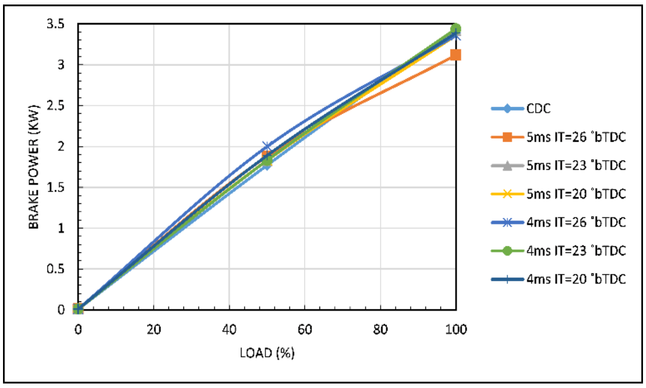

Figure 5. Brake power vs load at different injection timing.

3.3. Brake thermal efficiency

Brake Thermal Efficiency (BTE) of an RCCI engine is mostly influenced by the completeness and timing of combustion. BTE rises when combustion is precisely scheduled and more complete, facilitating the maximum conversion of fuel energy into productive work typically accomplished by efficient fuel premixing and combustion phasing. It diminishes due to incomplete combustion, excessive thermal losses, or delayed combustion, hence reducing the effective work output. Critical influencing aspects encompass injection timing, fuel reactivity ratio, and engine load conditions. Figure 6 illustrates the comparison of BTE between the CDC and RCCI engines at port fuel injection timings of 4 ms and 5 ms. The results show that the RCCI engine's BTE has increased compared to the CDC engine. At 4 ms port injection time, the BTE of the RCCl improved by 59.97%, 54.89%, and 54.19%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the BTE of RCCI increased by 33.33%, 27.44%, and 24.58%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the BTE of the RCCI rises by 63.21%, 58.67%, and 54.34%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the RCCI engine's BTE increases by 47.86%, 36.59%, and 20.97%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a maximum increase in BTE by 63.21% and 47.86% at 26˚ bTDC, at port injection timings of 5 ms under part load and maximum load respectively.

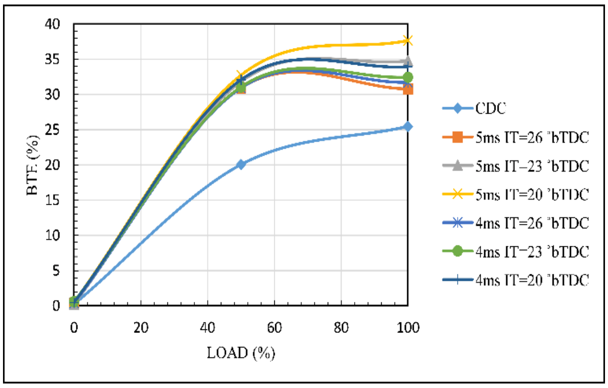

Figure 6. Brake thermal efficiency vs load at varying injection timing.

3.4. Cylinder pressure

The cylinder pressure in an RCCI engine is predominantly influenced by the combustion timing, combustion rate, and the fuel's energy content. It escalates with enhanced combustion phasing and augmented fuel energy release, particularly when a significant fraction of the charge is premixed and ignites swiftly. Conversely, cylinder pressure diminishes when combustion is postponed, extended over a protracted period, or when fuel is combusted inefficiently. Injection timing, fuel ratio, and in-cylinder temperature significantly affect pressure levels. Figure 7 presents a graph that compares the relationship between cylinder pressure and crank angle for both CDC and RCCI engines, using port fuel injection timings of 4 ms and 5 ms. The findings indicate that the cylinder pressure of the RCCI is lower than that of the CDC throughout all operational circumstances. At port injection timing of 4 ms, the cylinder pressure of the RCCI engine is reduced by 27.48%, 26.33%, and 23.76%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC. At port injection timing of 5 ms, the cylinder pressure of the RCCI is reduced by 25.54%, 22.45%, and 20.82%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC. The findings indicate a maximum reduction in cylinder pressure of 27.48% at 20˚ bTDC with 4 ms port injection duration and a minimum reduction of 20.82% at 26˚ bTDC with 5 ms port injection duration.

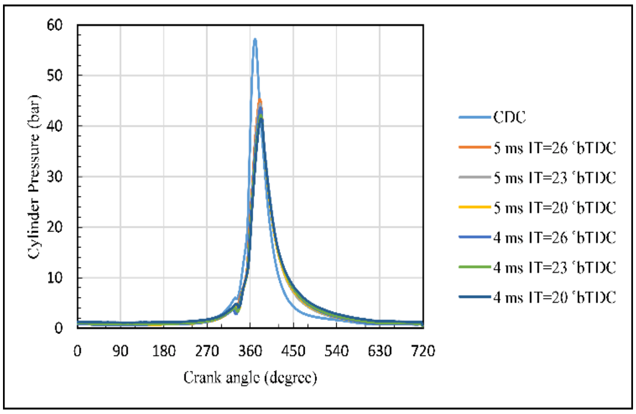

Figure 7. Cylinder pressure vs crank angle at varying injection timing.

3.5. Net heat release rate

The net heat release rate (NHRR) in an RCCI engine is predominantly affected by the disparity in fuel reactivity, combustion timing, and mixture formulation. NHRR escalates with improved premixing and enhanced combustion phasing, resulting in fast energy release during the premixed combustion phase. It diminishes when combustion is postponed or incomplete, frequently as a result of inadequate fuel mixing, low reactivity, or delayed injection timing. Effective regulation of fuel ratio, injection method, and ignition delay is crucial for optimizing NHRR. Figure 8 presents a graph that compares the relationship between NHRR and crank angle for both CDC and RCCI engines, using port fuel injection timings of 4 ms and 5 ms. The findings indicate that the NHRR of the RCCI is increased than that of the CDC throughout all operational circumstances. At port injection timing of 4 ms, the NHRR of the RCCI is increased by 21.01%, 22.38%, and 25.88%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC. At port injection timing of 5 ms, the NHRR of the RCCI is reduced by 17.83%, 19.93%, and 19.39%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC. The findings indicate a maximum increase in NHRR of 25.88% at 26˚ bTDC with 4 ms port injection duration and a minimum increase of 17.83% at 20˚ bTDC with 5 ms port injection duration.

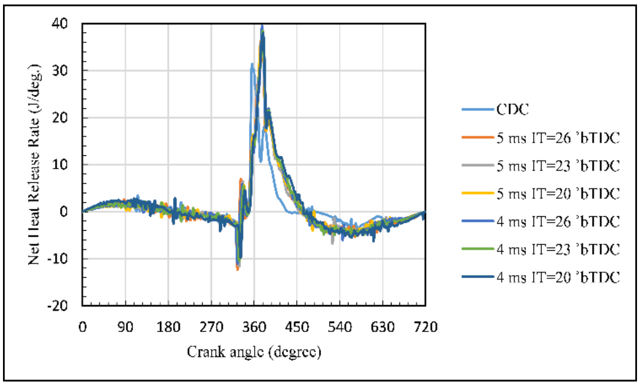

Figure 8. Net heat release rate vs crank angle at varying injection timing.

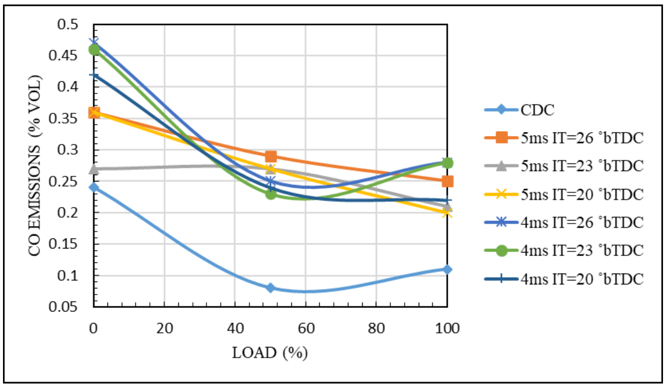

Figure 9. CO emission vs load at varying injection timing.

3.6. Carbon monoxide

Carbon monoxide (CO) emissions in an RCCI engine are primarily influenced by combustion temperature and the quality of air-fuel mixing. They escalate when combustion is incomplete, frequently resulting from low in-cylinder temperatures or excessively rich fuel regions, which impede the oxidation of CO to CO_2. Conversely, carbon monoxide emissions diminish in the presence of adequate oxygen and elevated combustion temperatures that facilitate complete oxidation. Enhancing injection time, fuel ratio, and mixture uniformity mitigates CO production in RCCI engines. Figure 9 illustrates the comparison of CO between the CDC and RCCI engines at port fuel injection timings of 4 ms and 5 ms. The results show that the RCCI engine's CO emissions has increased as compared to the CDC. At 4 ms port injection time, the CO emissions of the RCCI increased by 200.00%, 187.50%, and 212.50%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while the CO emissions of RCCI is increased by 100.00%, 154.55%, and 154.55%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing, the CO emissions of the RCCI is changed by 237.50%, 237.50%, and 262.50%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC under part load while it is changed by 81.82%, 90.91%, and 127.27%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC under maximum load. The findings indicate a maximum increase in CO emissions by 262.50% and 154.55% at 26˚

bTDC, at port injection timings of 5 ms and 4 ms under part load and maximum load respectively.

3.7. Hydrocarbon

Hydrocarbon (HC) emissions in an RCCI engine are mostly determined by the level of charge homogeneity and the completeness of combustion. They escalate when the premixed low-reactivity fuel (such as LPG) does not ignite completely, particularly in low-temperature areas or adjacent to cylinder walls, resulting in unburned hydrocarbons. In contrast, HC emissions diminish with optimal combustion phasing and enhanced mixing, facilitating more thorough fuel oxidation. Parameters such as intake temperature, injection timing, and equivalency ratio significantly influence hydrocarbon emission levels. Figure 10 illustrates the comparison of HC emissions be-tween the CDC and RCCI engines at port fuel injection timings of 4 ms and 5 ms. The results show that the RCCI engine's HC emissions has increased as compared to the CDC. At 4 ms port injection time, the HC of the RCCI increased by 270.00%, 190.00%, and 200.00%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC under part load while the HC emissions of RCCI is increased by 315.63%, 234.38%, and 284.38%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the HC emissions of the RCCI is increased by 215.00%, 210.00%, and 335.00%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while it is changed by 606.25%, 396.88%, and 565.63%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a maximum increase in HC emission by 606.25% and 315.63% at 20˚ bTDC, at port injection timings of 5 ms and 4 ms respectively, under maximum load.

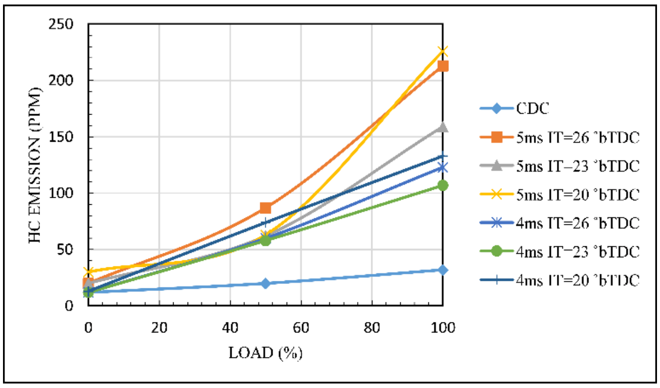

Figure 10. HC emission vs load at varying injection timing.

3.8. Nitrogen Oxide

Emissions in an RCCI engine are primarily determined by in-cylinder temperature and the availability of oxygen during combustion. Elevated combustion temperatures and surplus oxygen facilitate thermal nitrogen oxide (NO) generation, resulting in heightened NO emissions. Conversely, NO emissions diminish when combustion is more uniform and occurs at reduced peak temperatures, characteristic of RCCI mode. Effective regulation of injection timing, fuel stratification, and charge premixing is essential for reducing NO production. Figure 11 illustrates the comparison of NO between the CDC and RCCI engines at port fuel injection timings of 4 ms and 5 ms. The results show that the RCCI engine's NO emissions have reduced as compared to the CDC. At 4 ms port injection time, the NO emissions of the RCCI are decreased by 91.00%, 91.94%, and 89.10%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while it is decreased by 92.10%, 90.76%, and 89.19%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the NO emissions of the RCCI is decreased by 89.34%, 95.66%, and 93.60%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC under part load while it is decreased by 90.07%, 89.10%, and 87.58%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a maximum reduction in NO emissions by 95.66% and 92.10% at 23˚ bTDC and 20˚ bTDC at port injection timings of 5 ms and 4 ms under part load and maximum load conditions.

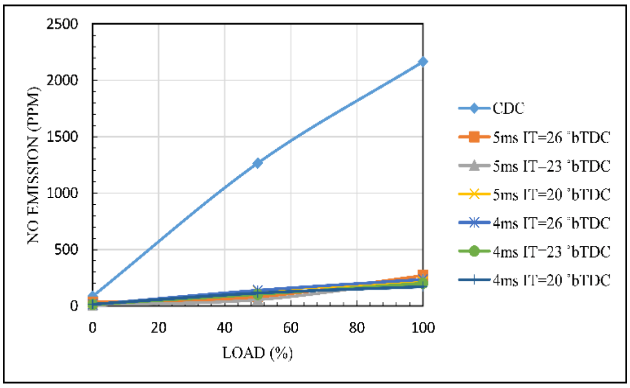

Figure 11. NO emission vs load at varying injection timing.

3.9. Smoke opacity

In the RCCI engine, smoke opacity is mostly determined by the local air-fuel mixture and combustion temperature. It escalates due to inadequate premixing or an excessively rich local mixture, resulting in incomplete combustion and soot generation. Conversely, it diminishes with enhanced premixing of low-reactivity fuels (such as LPG) and improved combustion phasing, which facilitates more complete and cleaner combustion. Injection time, fuel ratio, and in-cylinder temperature regulation are pivotal in managing smoke emissions. Figure 12 illustrates the comparison of smoke opacity between the CDC and RCCI engines at port fuel injection timings of 4 ms and 5 ms. The results show that the RCCI engines smoke opacity has reduced as compared to the CDC. At 4 ms port injection time, the smoke opacity of the RCCI decreased by 18.86%, 28.11%, and 21.35%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while it is changed by 65.84%, 60.37%, and 59.75%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. At 5 ms port injection timing the smoke opacity of the RCCI is decreased by 39.15%, 40.57%, and 46.62%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under part load while it is decreased by 73.17%, 73.04%, and 73.29%, respectively, at 20˚ bTDC, 23˚ bTDC, and 26˚ bTDC, under maximum load. The findings indicate a maximum reduction of smoke opacity by 73.29% and 46.62% at 26˚ bTDC, at port injection timings of 5 ms under maximum load and part load conditions respectively.

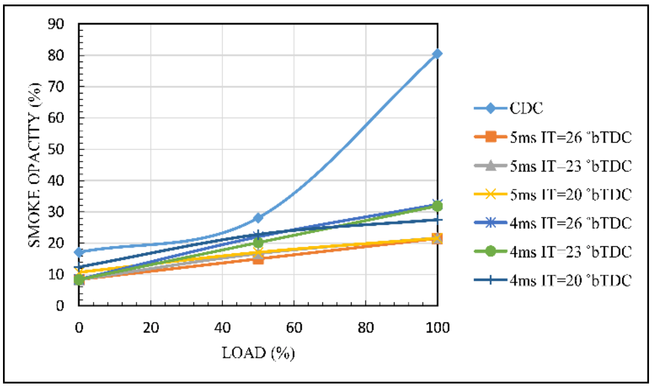

Figure 12. Smoke opacity vs load at varying injection timing.

The findings of this experiment indicate that the RCCI engine improves the IP, BP, and BTE. The cylinder pressure diminished; however, the NHRR increased. CO and HC pollutants have augmented in comparison to the CDC. Exhaust gas treatment technologies, such as thermal converters and catalytic converters, can control or reduce emissions. The decrease in NO and smoke opacity is markedly greater, mitigating the principal drawback of diesel engines. The findings indicate that the RCCI engine demonstrates considerable cyclic variability, complicating the identification of the precise parameters at which RCCI performance excels. Nonetheless, the statistics demonstrate that direct fuel injection at 26˚

bTDC produces more favorable results than 23˚ bTDC and 20˚ bTDC. The LPG port fuel injection timing of 5 milliseconds produces better results than 4 milliseconds. Moreover, additional research is required to determine the causes and factors influencing the cyclic variability in RCCI engine performance.

4. Conclusion

The LPG-diesel dual-fuel RCCI engine demonstrates significant enhancements in combustion efficiency and reductions in NOx and particle emissions. The impact of liquefied petroleum gas on the operation, combustion, and exhaust pollutant attributes of a diesel-powered engine operating in reactivity-controlled compression ignition mode was investigated. The investigations were conducted on CDC and RCCI engines with utmost power output of 3.5 kW. The whole performance of the RCCI engine is deemed good and, in certain respects, outstanding. The subsequent conclusions were derived from the comprehensive study of the results. This study's innovation is in employing LPG as a low-reactivity fuel in an RCCI engine, a topic that has been comparatively underexplored relative to other gaseous fuels. It surpasses prior initiatives by refining dual-fuel solutions to attain reduced emissions and enhanced efficiency across diverse load scenarios.

(1) The IP of RCCI is increased by 50.97% and 46.77% at part load and maximum load at 26˚ bTDC and 4 ms port injection duration in comparison to CDC. The BP of RCCI is increased by 12.99% at 26˚ bTDC for 4ms port injection duration at part load condition in comparison to CDC.

(2) The BTE is elevated by 63.21% and 47.86% at 26˚ bTDC for part load and maximum load at 5ms port injection duration for RCCI in comparison to CDC.

(3) The cylinder pressure of RCCI is decreased by 27.48% at 20˚ bTDC with 4 ms port injection duration and a minimum reduction of 20.82% at 26˚ bTDC with 5 ms port injection duration respectively. The NHRR for RCCI is increased by 25.88% for 26˚ bTDC and 17.83% at 20˚ bTDC for port injection duration of 4 ms and 5 ms respectively.

(4) The CO emission for RCCI is increased by 262.50% and 154.55% at 26˚ bTDC, at port injection duration of 5 ms and 4 ms under part load and maximum load respectively. The HC emission for RCCI is increased by 606.25% and 315.63% at 20˚ bTDC, at port injection duration of 5 ms and 4 ms respectively, under maximum load conditions.

(5) The NO emission for RCCI is decreased by 95.66% and 92.10% at 23˚ bTDC and 20˚ bTDC at port injection duration of 5 ms and 4 ms under part load and maximum load conditions respectively. The smoke opacity is decreased by 73.29% and 46.62% at 26˚ bTDC, at port injection duration of 5 ms under maximum load and part load conditions respectively.

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