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Experimental Investigation of a Reactivity Controlled Compression Ignition Engine Fuelled with Liquified Petroleum Gas

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Abstract The primary source of power in many situations, including backup power, emergencies, isolated locations, construction, etc., is an internal combustion engine. These have higher engine emissions, which is a major drawback. Low temperature combustion engines may prove to be the best option in this case, because they not only produce power with high efficiency but also produce fewer engine emissions. It was investigated how a reactivity-controlled compression ignition engine that runs on liquid petroleum gas performs and produces emissions. The pilot fuel i.e., diesel was directly injected during the compression stroke into the engine cylinder, whereas the main fuel, liquified petroleum gas, was injected during suction stroke into the inlet port via mechanical injection system. At the engine's inlet port, an electronic port injector was mounted. The engine's experimental testing was conducted at a fixed 1500 rotation per minute controlled with the help of governor. The maximum rated power output of the engine was 3.7 kW. The ratio of premixed energy is taken at 95%. Experiments were first carried out on a normal diesel combustion engine before being switched to the reactivity controlled compression ignition (RCCI) engine. The experimental results demonstrate the brake specific fuel consumption (BSFC) and brake power (BP) is reduced up to 83.14% and by 34.65% respectively. The rise in cooling water temperature is reduced by 15.38% and 5.88% at 0% and 100% loading conditions respectively. The exhaust gas temperature is reduced by up to 29.77%. Brake thermal efficiency increased by 19.17%. Smoke opacity is reduced by 81.29% and 69.81% at 0% and 100% loading condition respectively, as compared to the normal diesel combustion engine. According to the findings, a reactivity controlled compression engine may operate efficiently using liquified petroleum gas that contains approximately 95% premixed energy. As a result, there will be less demand for diesel fuel and engine emissions.

Keywords LTCE, RCCI, Liquified Petroleum Gas, Emission, Smoke Opacity

1. Introduction

The transportation, power sectors, agricultural applications, construction sector, and many more fields frequently use internal combustion engines (ICE). The ICEs have various advantages, like well-established technology, low initial and maintenance costs, high power, and easy and fast refueling. Due to all these advantages, ICEs are preferred, and currently there are 1.2 billion vehicles on the road [1]. As we know, ICE primarily uses diesel and gasoline fuel for their power production. The quantity of these hydrocarbon fuels is decreasing every day. Even after that, according to estimates, crude oil can last up to 40–50 years from now [2]. In 2021, 113 exajoule (EJ) energy was used to power transportation globally. Out of that, around 91.15% came from crude oil. Similarly, the future projection for transportation shows that energy consumption has increased from 113 EJ in 2021 to 130 EJ in 2030, 139 EJ in 2040, and 147 EJ in 2050. The crude oil share will be 86.92%, 82.01%, and 78.91% in 2030, 2040, and 2050, respectively, as shown in figure 1 [3]. But ICEs have some major disadvantages like low efficiency and high engine emissions such as carbon monoxide (CO), carbon dioxide (CO_2), hydrocarbons (HC), nitrogen oxides (NOx), and particulate matter (PM). These emissions are causing problems for the environment and human life [2]. Some of the negative impacts on both people and the environment are mentioned below:

(i). Nitrogen oxides: NOx emissions contain NO, a colorless gas with no smell, whereas NO_2 is a gas with a pungent smell and a reddish-brown color. Nitrogen oxides, which are produced by diesel engines, a greater than 50% share of all emissions. These cause a variety of environmental issues, including acidification of aquatic bodies, ozone production, smog production, acid rain, and nutrient enrichment, as well as health issues, such as respiratory infections, human lung diseases, and pollution haze, which reduces visibility [4].

(ii). Particulate matter: The second-highest emission content of all exhaust pollutants comes from diesel engines. PM has an adverse impact on the ecosystem, causing air and water pollution, soiled buildings and monuments, reduced agricultural productivity, reduced visibility, changes in the global environment, etc. Asthma, breathlessness, asphyxia, lung cancer, premature mortality, etc. are just a few of the negative consequences of PM on human health, which also adds to other types of cardiovascular disorders [5, 6].

(iii). Carbon monoxide: It is a gas without color and odor that harms people's health in a number of ways. CO lowers the ability of hemoglobin to transport oxygen when it is breathed along with air. Additionally, it results in asphyxiation, disrupts the various human organs' functions, results in delayed reflexes, impairs focus, and produces confusion and bewilderment [7].

(iv). Hydrocarbons: HC has a number of harmful consequences for the ecosystem, the climatic conditions, living organisms, and health of people. They produce hazardous, cancer-causing ground-level ozone that irritates the respiratory system [4, 7].

Due to these hazardous impacts of emissions on both humans and the environment, a number of regulations are imposed on transport sector, like Euro emission norms, Bharat stage norms etc. One such emission norm is listed in table 1.

Table 1. European Union emission norms for heavy-duty vehicles [8]

Emission norms	HC (kg/ kWh)	CO (kg/ kWh)	PM (kg/ kWh)	NOx (kg/ kWh)
Euro-1	11×10^{-4}	45×10^{-4}	61×10^{-5}	80×10^{-4}
Euro-2	11×10^{-4}	40×10^{-4}	15×10^{-5}	70×10^{-4}
Euro-3	66×10^{-5}	21×10^{-4}	13×10^{-5}	50×10^{-4}
Euro-4	46×10^{-5}	15×10^{-4}	20×10^{-6}	35×10^{-4}
Euro-5	46×10^{-5}	15×10^{-4}	20×10^{-6}	20×10^{-4}
Euro-6	13×10^{-5}	15×10^{-4}	10×10^{-6}	40×10^{-4}

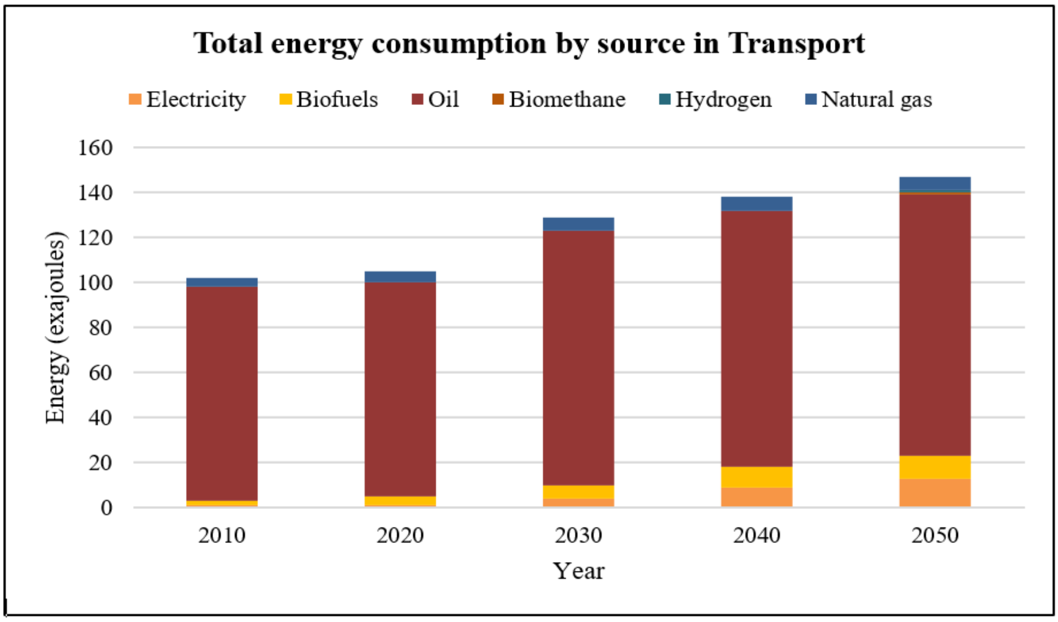

Figure 1. Total energy consumption by source in transport sector in exajoules (EJ) [3]

This leads the researchers to improve the ICEs and find some way to reduce the engine emissions. Low temperature combustion engines (LTCE) are providing such an alternative that not only reduces engine emissions but also improves thermal efficiency [9]. There are different variants of LTCE available, such as homogeneous charge compression ignition (HCCI) engines, premixed charge compression ignition (PCCI) engines, reactivity controlled compression ignition (RCCI) engines, and gasoline direct injection (GDI) engines. Because of its features such as high efficiency, fuel flexibility, minimal cycle-to-cycle fluctuations, etc., the RCCI engine is the most forward-looking technology that can be used in the future [10]. A number of authors performed research and experiments with gasoline as low reactivity fuel (LRF) on a RCCI engine. But research and experiments on RCCI engines with alternative fuels, like biogas, ethanol, CNG, LPG, hydrogen, methanol, etc., are limited.

RCCI combustion tests are carried out by Park and Yoon [11] in a single-cylinder engine using biogas and gasoline as LRF. The HRF fuel injection timing is varied between 5- and 40-degree crank angle (CA) before TDC. According to the findings, the indicated mean effective pressure (IMEP) for biogas decreased by up to 78.9 percent, and the indicated specific fuel consumption (ISFC) increased. The ignition delay is up to 35% longer, the cylinder pressure is higher, and the peak heat release rate is lower. The findings also indicated decreases in NOx and soot emissions, while increases in CO and HC emissions. Qian et al. [12] used biogas as a LRF in RCCI combustion in a four-cylinder engine. The injection was performed between 14 and 10 oCA bTDC. The findings demonstrated that when the biogas ratio rose, the cylinder's temperature and heat release rate (HRR) rose but the pressure inside the cylinder reduced. Poorghasemi et al. [13] employed natural gas as a LRF in a four-cylinder engine. The results demonstrate reduction in HRR, cylinder pressure, and NOx emissions with the increased premix ratio. While CO and HC emissions rose under every operational circumstance. Natural gas was employed as a LRF in a single-cylinder engine by Paykani et al. [14]. The cylinder pressure, HRR, smoke and NOx emissions all decreased as the premixed ratio increased, but CO and UHC emissions rose. In a four-cylinder engine, Kalsi and Subramanian [15] used compressed natural gas as a LRF. The results indicated that a smaller injection of CNG reduced the BSFC and increased thermal efficiency. The period of combustion and the HRR are longer with a lower percentage of CNG. In addition to being lowered, the emissions of NOx and soot were also held steady, while those of CO and HC were not. In a four-cylinder engine, Kalsi and Subramanian [16] used compressed natural gas with hydrogen blend (HCNG) as LRF. The results demonstrate increases in brake thermal efficiency, cylinder temperature, HRR, and combustion duration. Additionally, the results indicated that as the percentage of HCNG was raised, emissions like NOx and smoke decreased, whereas emissions like CO and HC increased. Krishnan et al. [17] used propane gas as a LRF in a single cylinder engine. With advanced start of injection (SOI) and injection pressure, the results demonstrated a decrease in cylinder pressure, and with advanced combustion phasing, a higher HRR. The findings also revealed decrease in emissions like NOx and smoke whereas emissions like CO and HC were increased.

In a heavy-duty engine, Jia Z and Denbratt carried out experimental research of RCCI combustion using diesel as an HRF and natural gas as an LRF. The shares of energy of diesel and CNG were 33% and 67% at 1200 rpm and with a 17:1 compression ratio. The results show that the NOx produced by the CDC engine is lower than those produced by the RCCI engine, at 0.209 g/kWh, but the soot emissions produced by the CDC engine are much higher, at 3.14 g/kWh. The RCCI engine produces more CO and UHC emissions than the CDC engine [18]. Ebrahimi and Jazayeri performed a simulation study on RCCI combustion with diesel as HRF and landfill gas and hydrogen as LRF in a heavy-duty, one-cylinder engine. The result shows increased peak cylinder pressure, whereas the load carrying capacity of engine is diminished by 4%. Additionally, the findings indicate a modest decrease in NOx emissions, whereas a significant decrease in CO emissions [19]. A 17:1 compression ratio was used in the experimental study by Gharehghani et al. [20], which used biodiesel as the fuel with the highest reactivity and CNG as the fuel with the lowest reactivity. In comparison to diesel/CNG, the results demonstrate less variance from cycle to cycle. The result also shows increase in gross thermal efficiency by 2%. The combustion losses are 18.85% and 20.88% for biodiesel/CNG and diesel/CNG respectively. The emissions like UHC and NOx are decreased by 32.5% and 36% respectively on average through all engine loads. In comparison to normal diesel combustion, the data also reveal a decrease in CO emissions.

The information that is currently accessible reveals that various types of gaseous fuels like CNG, HCNG, biogas etc., are used in RCCI. The smoke and NOx emissions are typically observed to be reduced considerably but CO and HC emissions are increased. The use of LPG as low reactivity fuel is not much used in RCCI combustion. So, it is interesting to perform experiments with LPG fuel in RCCI mode and evaluate its performance. The impact of LPG fuel on the operation, emissions and performance of the diesel engine in dual fuel RCCI mode is not adequately covered in the literature. This research aims to analyze how LPG affects the operation, combustion, and emission characteristics of a dual-fuel RCCI engine operating with diesel as pilot fuel.

2. Experimental Setup

A single-cylinder, normally aspirated diesel engine with a set speed of 1500 rpm and a rated power output of 3.7 kW was used for the experimental tests. Table 2 lists the technical details of the experimental engine configuration. Figure 2 displays the conceptual layout of the experimental design.

The experimental engine was changed to operate in RCCI mode. As low- and high-reactivity fuels, LPG gas and diesel were used. A port fuel injector was installed to supply main fuel (LRF), and the conventional direct injector was used to supply pilot fuel (HRF). The engine was connected to an alternator, and to load the engine, an electrical panel was used that is connected to the alternator. To measure smoke emission, an AVL-437 smoke meter was employed. Table 3 lists the physical and chemical characteristics of the primary fuel (LPG) and pilot fuel (Diesel). The burette method was used to measure the fuel flow rate of HRF, while the electronic weight balance method was used to estimate the fuel flow rate of LRF.

The engine efficiency, brake power, fuel consumption, etc., were calculated with the help of an analytical method and are discussed in next section. The quantity of LRF and HRF was kept at 95% by energy ratio, and the fuel injection timing for HRF and LRF was kept at 23 °CA BTDC and 0 °ATDC, respectively. The governor was in charge of keeping the engine speed constant under various loads. Therefore, the quantity of diesel was controlled automatically by governing according to the LPG energy share. The injection pressures of HRF and LRF were kept at 200 bar and 3 bar, respectively.

Table 2. Specifications of Experimental engine setup

Description	**Unit**
Rated speed	1500 rpm
Rated power	3.7 kW
Compression ratio	17.5:1
No. of cylinder	1
Bore x stroke	87.5 x 110 cm
Engine capacity	661 cm^3
Nozzle opening pressure	200 bar
Injector nozzle diameter	0.19 mm
Aspiration mode	Naturally aspirated
Type of cooling	Water

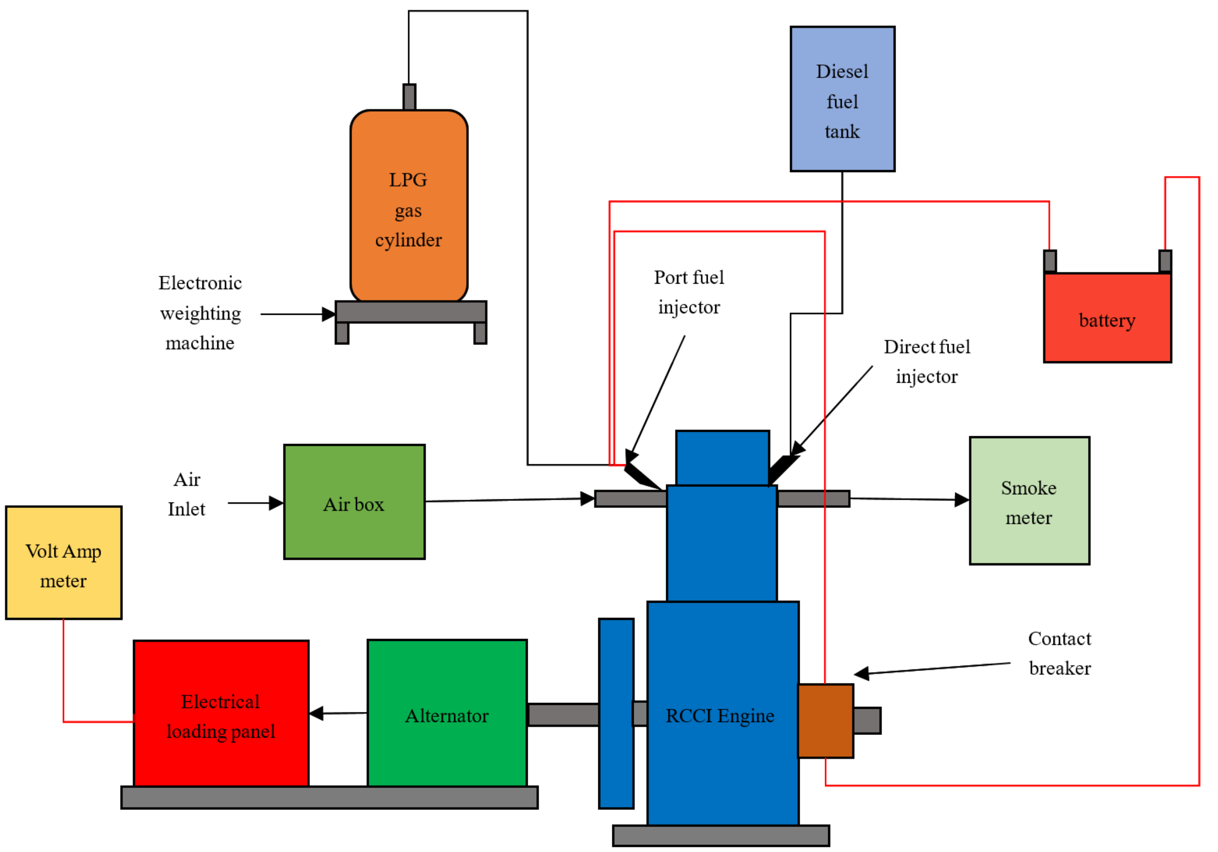

Figure 2. Descriptive view of experimental setup

Table 3. Physical and chemical properties of diesel and LPG [21]

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

3. Experimental Procedure

The experiments conducted are separated into two parts. During the initial stage, the experiments were conducted with diesel fuel only at different loads on the conventional diesel engine. In the second phase, the standard diesel engine was changed to run in RCCI mode. The experiments were carried out using two fuels: LRF i.e., LPG and HRF i.e., diesel. The experimental engine was loaded using the electrical resistive loading panel. The loading of the engine was done in stages. The loading was increased by 500 W at a time during the experiments. The cooling water is allowed to flow into the engine at a constant rate of 250 l/h. The temperature of the exhaust gas, cooling water inlet, and exit are all measured using K-type temperature sensors, which are mounted at various points. The experimental engine was started and allowed to run for some time to warm up. The readings were noted down for each load after waiting for 3–4 minutes so that the engine became stable. The average value of each reading was taken into consideration for evaluation after each test was administered three times. The energy share of LPG-diesel was varied for smooth engine operation and to find the optimum working condition. For all LPG energy shares, the engine's output parameters and emissions can be assessed. The energy share for LPG can be determined using Eq. (1) [22, 23].

$$E_{LPG} = \left(\frac{m_{LPG} \cdot CV_{LPG}}{m_{LPG} \cdot CV_{LPG} + m_{diesel} CV_{diesel}}\right) \times 100\% \quad (1)$$

where, m_{LPG}(kg/s) is the rate of flow of LPG mass, m_{diesel}(kg/s) is the rate of flow of diesel mass and CV_{LPG}, CV_{diesel} are the LPG and diesel fuel calorific values respectively. Equation (2) is used to calculate the diesel engine's brake thermal efficiency [22].

$$\eta_{bth} = \left(\frac{BP}{m_{LPG} \cdot CV_{LPG} + m_{diesel} CV_{diesel}}\right) \times 100\% \quad (2)$$

The BSFC for RCCI and CDC engine can be determined using equation (3) [24].

$$BSFC = \frac{m_{fuel}}{BP} \quad (3)$$

Where, m_{fuel}, is total mass of fuel consumption in kg/hr and BP is engine brake power.

4. Results and Discussion

The effect of an LPG-operated RCCI engine on various parameters is discussed in this section. The factors considered in this study are brake specific fuel consumption (BSFC), brake power (BP), rise in cooling water temperature (RCWT), exhaust gas temperature (EGT), smoke opacity and brake thermal efficiency (η_{bth}).

4.1. Brake Power

Figure 3 displays the brake power (BP) of the CDC and RCCI engines. According to the experimental findings, the CDC and RCCI engines produce a maximum BP of 3.68 kW and 2.41 kW, respectively, at full load. The results show a reduction in BP for the RCCI engine in comparison to the CDC by 0.0%, 22.73%, 29.51%, 31.17%, and 34.65% at 0%, 25%, 50%, 75%, and 100% load, respectively. The slow combustion that resulted from flame front propagation is what causes the RCCI engine's brake power to be less than that of the CDC engine. The temperature and pressure created in the RCCI engine are lower than those of the CDC engine because the fuel is burned more gradually, and it also takes more time to burn the fuel in RCCI combustion. The gradual combustion's impact became apparent when the BP fell.

4.2. Brake Specific Fuel Consumption

Figure 4 displays the BSFC for the CDC and RCCI engines. The experimental results show that the BSFC for the CDC and RCCI engines are 0.37 kg/hr-kW and 0.06 kg/hr-kW, respectively, at full load operating condition. The results show a reduction in BSFC for RCCI engine in comparison to CDC engine of 38.86%, 63.44%, 78.95%, 82.32%, and 83.14% at 0%, 25%, 50%, 75%, and 100% load, respectively. This is due to the fact that in an RCCI engine, the LHV of low-reactivity fuel is greater than that of fuel with higher reactivity. In this investigation, LPG is used as a LRF, and its LHV is higher than that of diesel. Consequently, the total amount of fuel required to produce the same amount of energy was reduced.

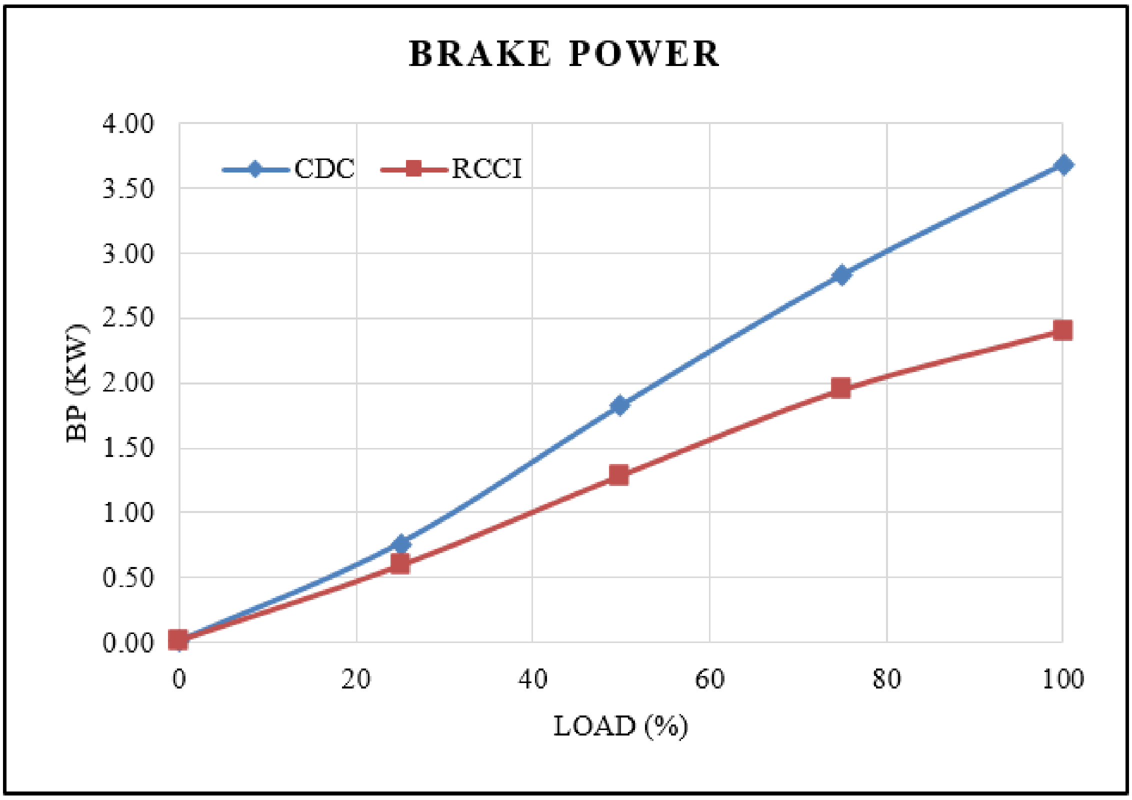

Figure 3. Comparison of brake power vs load for CDC and RCCI

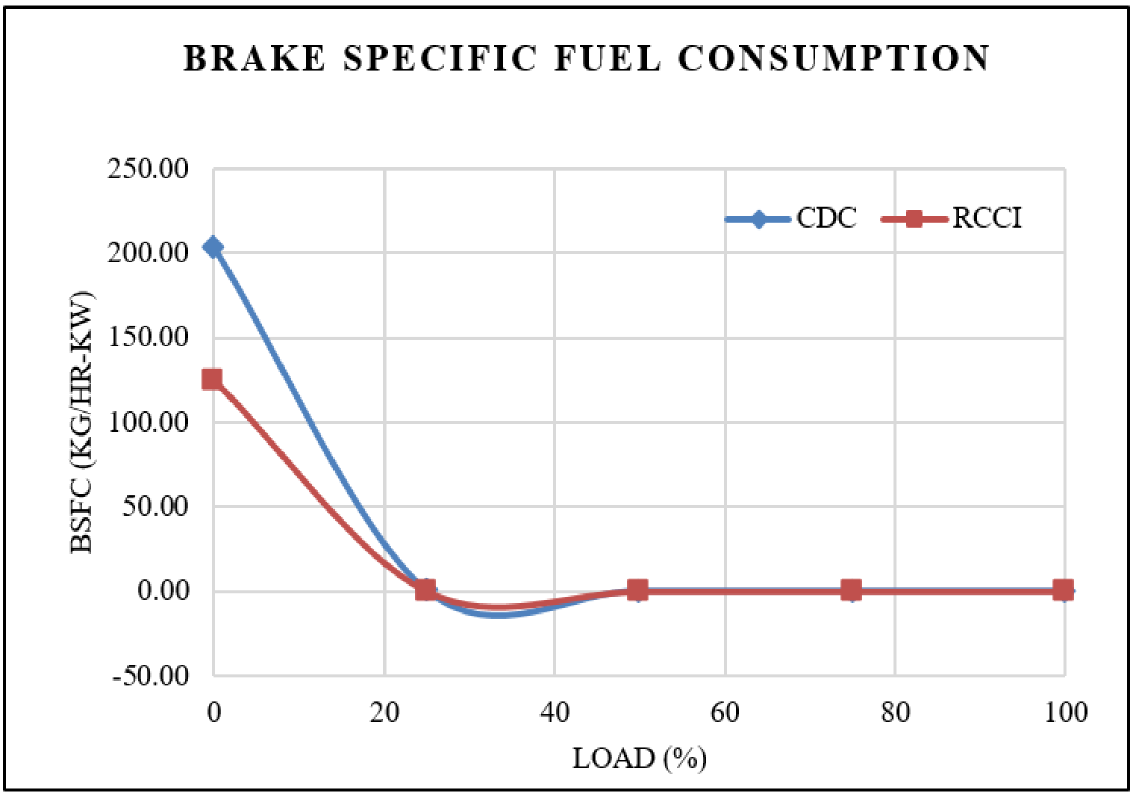

Figure 4. Comparison of brake specific fuel consumption vs load for CDC and RCCI

4.3. Rise in Cooling Water Temperature

Figure 5 depicts the rise in cooling water temperature (RCWT) for CDC and RCCI engines. According to the findings, the maximum RCWT for the CDC and RCCI engines at full load operating conditions are 34 ℃ and 32 ℃, respectively. RCWT for the RCCI engine is lower than for the CDC engine, according to the results by 15.38%, 7.41%, 3.45%, 6.45%, and 5.88% at the loads varying from 0% to 100% respectively. This is because the combustion of fuel in an RCCI engine is more gradual and not instantaneous like in a CDC engine. As a result, the RCCI engine's combustion chamber temperature develops at a lower rate than the CDC engine. This results in the reduction of RCWT.

4.4. Exhaust Gas Temperature

Figure 6 displays the EGT for CDC and RCCI engines. According to the findings of the experimental study, the maximum EGT obtained in a CDC and a RCCI engine at full load is 430 ℃ and 302 ℃, respectively. According to the findings, the RCCI engine's EGT is lower than the CDC engine by 3.73%, 14.35%, 11.54%, 19.82%, and 29.77% at the loads varying from 0% to 100% respectively. This is because the combustion of fuel in an RCCI engine is more gradual and not instantaneous like in a CDC engine. As a result, the RCCI engine's combustion chamber temperature develops at a lower rate than the CDC engine. This results in the reduction of EGT.

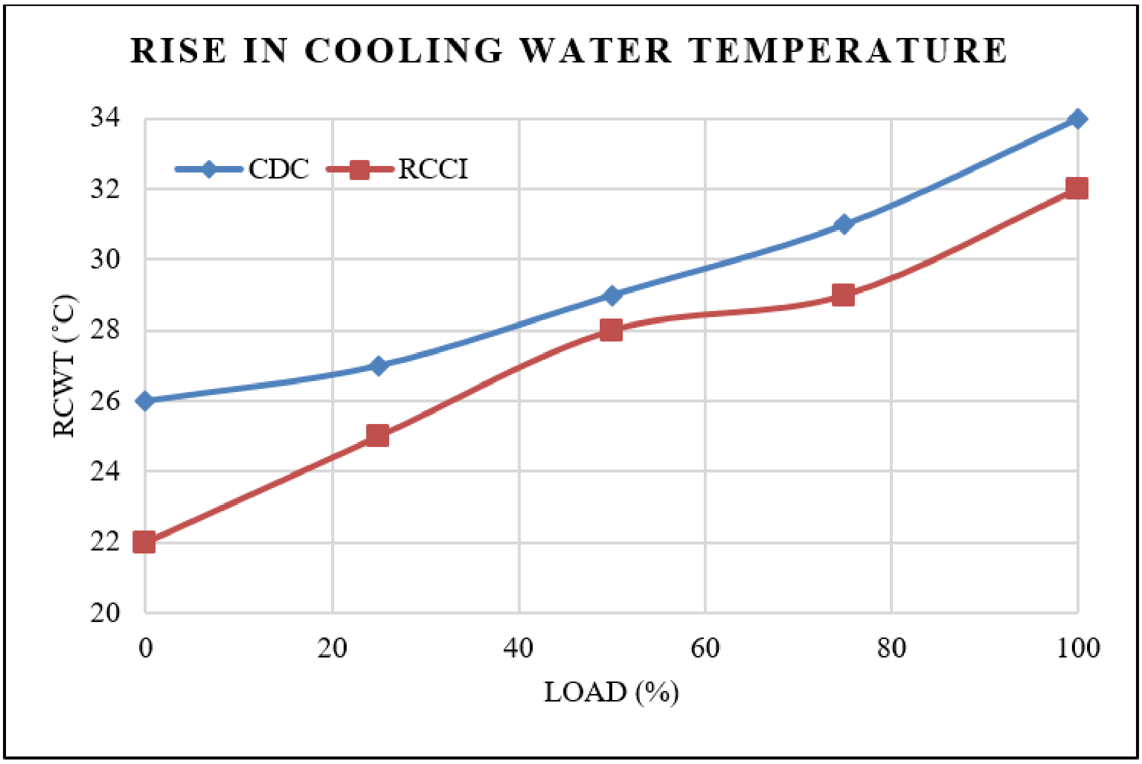

Figure 5. Comparison of rise in cooling water temperature vs load for CDC and RCCI

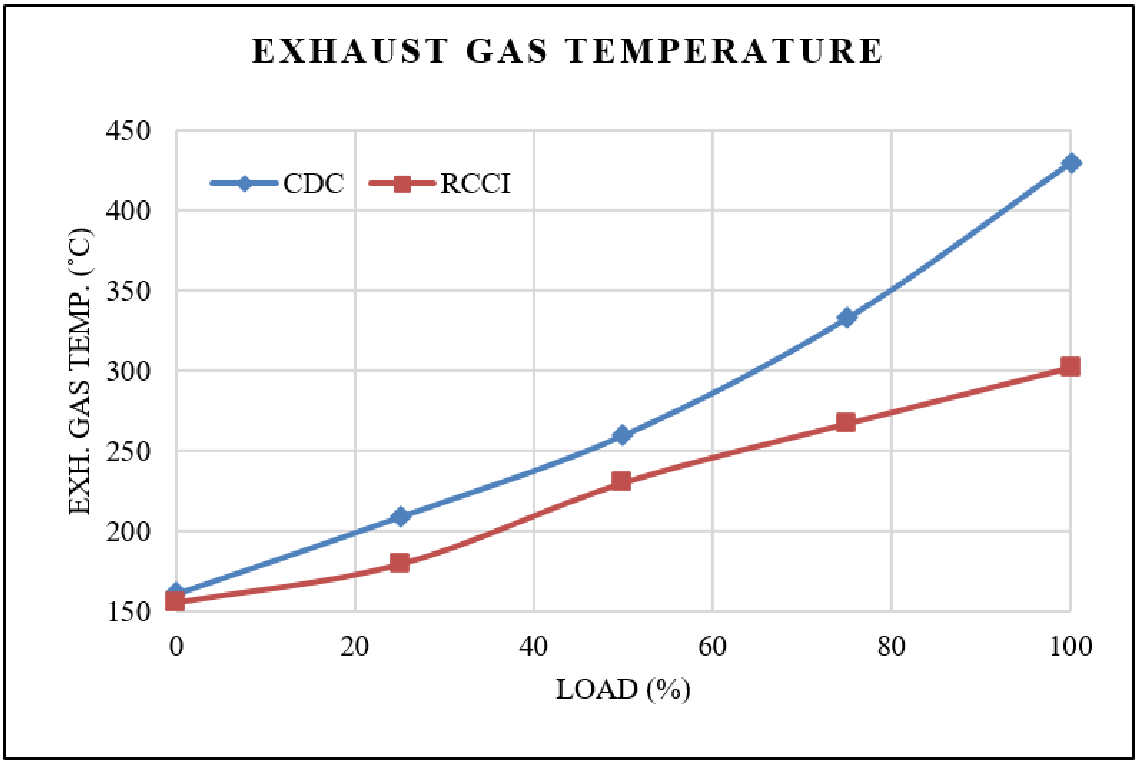

Figure 6. Comparison of exhaust gas temperature vs load for CDC and RCCI

4.5. Brake Thermal Efficiency

The brake thermal efficiency of the CDC and RCCI engines is depicted in figure 7. The maximum η_{bth} produced by the CDC and RCCI engines is 22.97% and 27.37%, respectively, according to the results. The results show that the η_{bth} for the RCCI engine is reduced by 70.91%, 50.37%, and 19.02% at 0%, 25%, and 50% load, respectively, while increasing by 16.23% and 19.17% at 75% and 100% load, respectively.

4.6. Smoke Opacity

Figure 8 depicts the opacity of the smoke in the CDC and RCCI engines. According to the findings of the experimental study, the maximum smoke opacity produced by the CDC and RCCI engines is 80.5% and 24.3%, respectively. The results show a reduction in smoke opacity for the RCCI combustion in comparison to CDC by 81.29%, 74.49%, 58.36%, 68.11%, and 69.81% at 0%, 25%, 50%, 75%, and 100% load, respectively. There are many factors which affect the smoke opacity such as incomplete combustion, incorrect timing, incorrect air-fuel ratio, dirty or worn injectors. The key reason for the smoke that the diesel engine produces is the incomplete combustion of the fuel. High fuel-air ratios (rich mixtures) and insufficient combustion time are two factors that can lead to incomplete fuel combustion. First, incomplete combustion happens when the amount of fuel increases while the amount of oxygen needed decreases. Second, if there is not enough time for combustion, it results in incomplete combustion, and more smoke is produced. There are two factors that contribute to the lowering of smoke in this experimental investigation. First, during the intake stroke, a homogenous mixture is created, making the necessary amount of oxygen accessible for complete combustion. Second, the fuel is completely burned because the progressive combustion allows for sufficient time for the combustion to take place. As a result, the smoke coming from the RCCI engine is reduced.

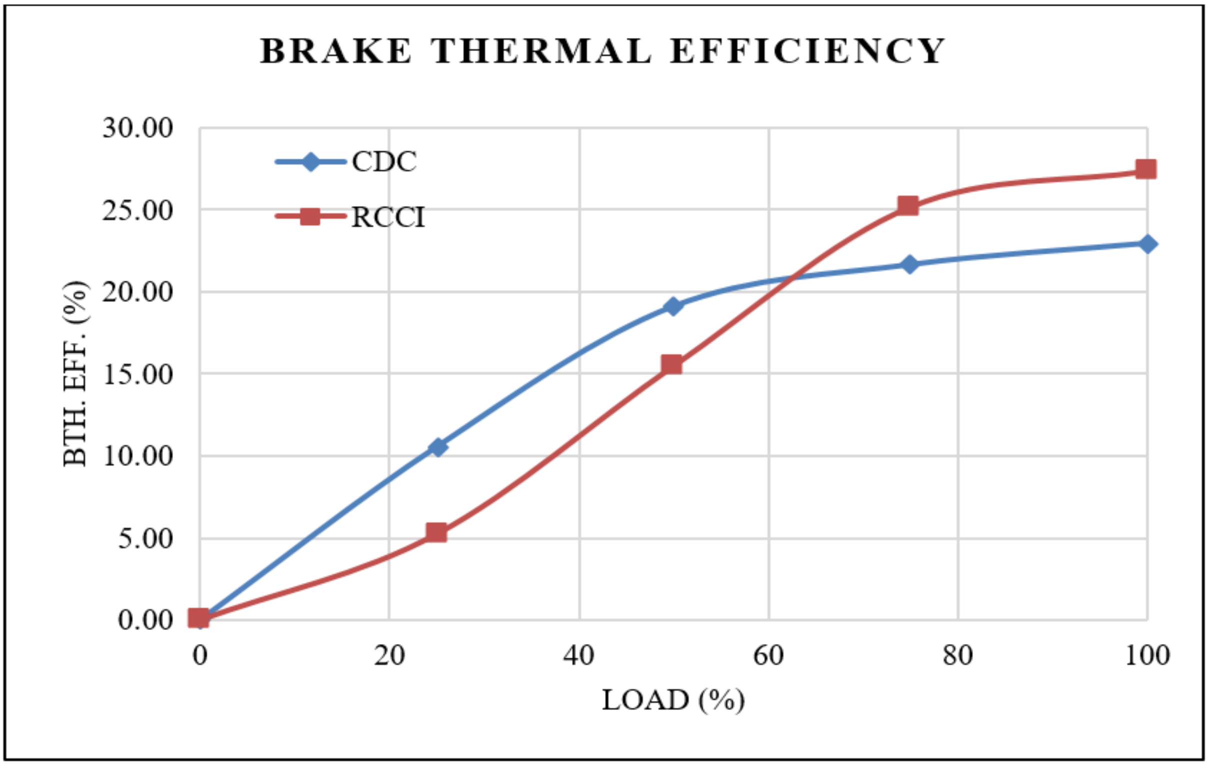

Figure 7. Comparison of brake thermal efficiency vs load for CDC and RCCI engine

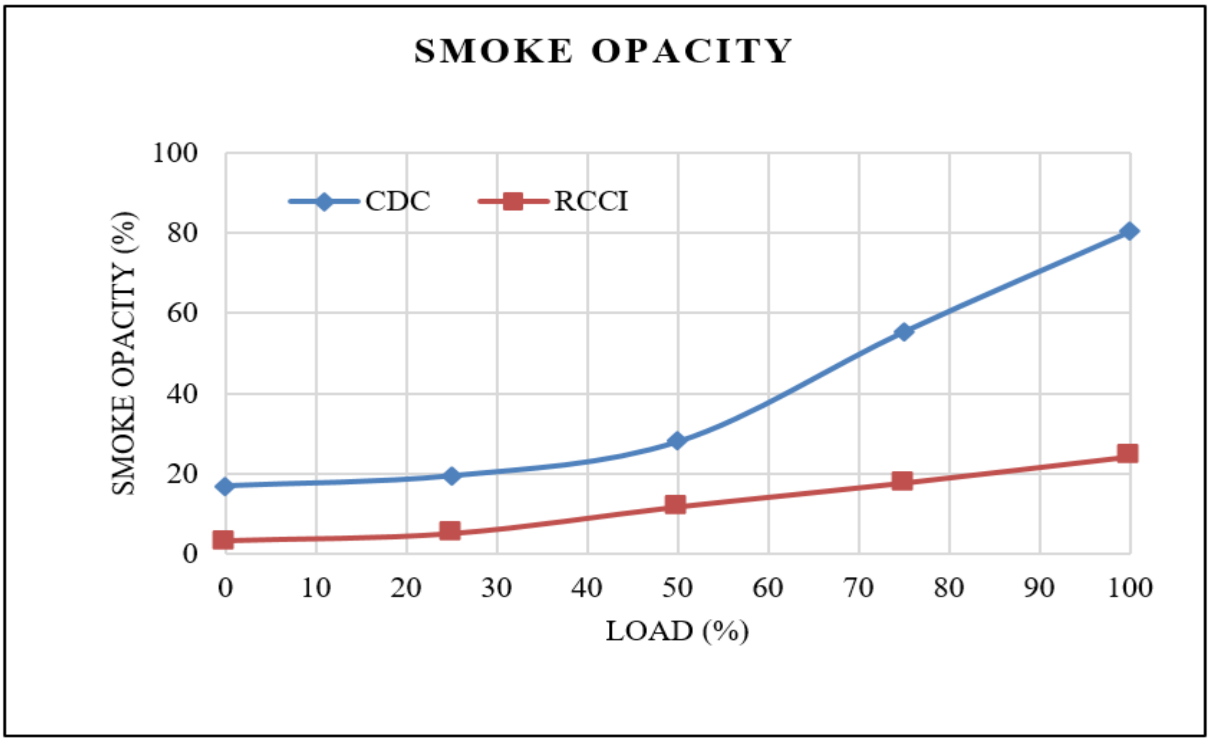

Figure 8. Comparison of smoke opacity vs load for CDC and RCCI

5. Results and Discussion

A compression ignition engine operating in reactivity-controlled compression ignition (RCCI) mode was researched with regard to its performance, combustion, and emissions characteristics. While the main fuel, liquified petroleum gas (LPG), was injected during the suction stroke using a mechanical injection system, the diesel used as the pilot fuel was directly pumped into the engine cylinder during the compression stroke. The engine's experimental testing was done at a fixed 1500 rpm while producing 3.7 kW of electrical power. The major conclusions drawn from this experimental study are given below:

- According to the testing findings, an RCCI engine's BP is lower at full load than a CDC engine by 34.6%.
- The BSFC is reduced up to 38.86% and 83.14% at the loads of 0% and 100% respectively, compared to the CDC engine.
- The rise in cooling water temperature (RCWT) of RCCI engines is reduced by 15.38% and 5.88% at the loads 0% and 100% respectively, compared to CDC engines.
- The exhaust gas temperature (EGT) of RCCI engines is reduced by 3.73% and 29.77% at the loads of 0% and 100% respectively, compared to the CDC engine.
- The η_{bth} of the RCCI combustion is higher by 19.17% at full load in comparison to CDC.
- The smoke opacity of the RCCI engine is reduced by 81.29% and 69.81% at the loads of 0% and 100% respectively, compared to the CDC engine.

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