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# Review on effect of low and high reactive fuels on reactivity controlled compression ignition engine

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**Abstract**

A proposed low-temperature combustion technique for obtaining high thermal efficiency and extremely low emissions in internal combustion engines is reactivity regulated compression ignition (RCCI). In order to control in-cylinder combustion phasing and heat release, the concept depends on the controlled employment of two fuels with different reactivities, usually a low-reactive fuel (like gasoline, ethanol, or natural gas) and a high-reactive fuel (like diesel or biodiesel). The impacts of various low- and high-reactivity fuel mixtures on engine performance, combustion properties, and emission behavior in RCCI operation are rigorously examined in this paper. Analysis is done on how ignition delay, pressure rise rate, and thermal efficiency are affected by fuel characteristics, blending ratios, injection time, and charge preparation. The study also covers contemporary developments in the use of alternative fuels, including biofuels and synthetic fuels, as well as difficulties with fuel selection, mixture preparation, and management systems. The research emphasizes that while preserving or increasing efficiency, the best fuel combination can result in notable reductions in NOx and particulate matter. To improve combustion controllability, increase load range, and guarantee practical application in real-world engines, more research is necessary.

**Keywords:** Reactivity controlled compression ignition (RCCI), dual-fuel combustion, low reactive fuel, high reactive fuel, emissions, engine performance

## Introduction

Diesel engines are widely used in many industrial applications, such as power plants and other forms of transportation, because of their high efficiency and longevity. Furthermore, the development of more affordable and environmentally friendly diesel engines is required due to tight emissions rules and growing concerns about the depletion of petroleum. In particular, it is quite challenging to capture both the soot and nitrogen oxide emissions from diesel engines simultaneously when they are not treated by an after-treatment system. These demands and problems spur engine innovation aimed at increasing engine efficiency and lowering emissions of pollutants at the same time. Furthermore, as bio-fuels and sophisticated solar fuels become more accessible, technological developments need to increase the adaptability of transportation and electricity generation systems to alternative fuels. The development of low temperature combustion (LTC) techniques aims to create a fuel-independent, clean, and efficient engine [1]. The partially-premixed compression ignition (PPCI) type already uses the RCCI (reactivity-controlled compression ignition) technique. Another LTC dual-fuel technique is RCCI combustion. This technique uses in-cylinder fuel mixing to create premixed mixtures from two fuels with different reactivities (low and high reactivity). In order to improve the combustion duration and combustion phasing, this technique combines multiple injection approaches with an appropriate exahust gas recirculation (EGR) rate to adjust in-cylinder charge reactivity. This leads to increased thermal efficiency and ultralow levels of soot and NOx emissions [2]. Combustion luminescence, which began farther out from the injector and varied in position, was used to identify the higher degree of premixing in RCCI operation. A limited brightness was produced by the diluted pilot combustion. The characteristic diesel premixed peak was absent from the heat release profile, which had a Gaussian/bell shape. The captured photos demonstrate that combustion in a typical dual fuel operation follows the diesel injector jet's form. A prominent early peak that resembled the premixed peak found in conventional diesel combustion was visible in the heat release profile. In RCCI mode, there is no heat release peak associated with nitrogen oxide emissions that occurs in normal dual fuel combustion [3]. Because RCCI heavy-duty engines perform better than homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI) engines, the industry, shipping, and heavy vehicles have long explored using them. Its advancement is therefore crucial to achieving increased energy efficiency and lowering environmental contaminants. Researchers are therefore urged to look for renewable sources for this kind of engine given the rising cost of fossil fuels worldwide and the rise in environmental warnings [4]. Human societies are producing more solid waste, which raises issues with the environment and global warming. High amounts of greenhouse gasses are also produced by the widespread usage of dual-fuel and traditional diesel engines, which exacerbates the previously described issues. Thus, lowering the greenhouse gas emissions from current heavy-duty diesel engines that run on a dual fuel of natural gas and diesel is the goal of this study. This was accomplished by simultaneously switching to reactivity-controlled compression-ignition combustion and using landfill gas in a dual-fuel engine rather than natural gas. Furthermore, in order to identify the proper engine operating ranges, a conventional approach was employed to assess the impact of changes in three crucial factors on the engine's performance. According to the modeling results, even when replacing landfill gas alone reduces the 102, 000 cubic meters of natural gas used annually in each cylinder, engine greenhouse gas emissions are still too high in comparison to the applicable emissions criteria. Therefore, landfill gas enrichment with hydrogen was thought to lower engine emissions while maintaining a steady overall energy content of the fuels. According to the modeling results, engine load could be increased by 7% without any exposure to diesel knock if the hydrogen energy share is increased to 37%. The drawback, though, is the 3% drop in gross indicated efficiency. In the meantime, the high levels of unburned methane, a drastic greenhouse gas, and formaldehyde, a related carcinogenic species, can be overcome in addition to reaching the fifth level of the European emission standard for nitrogen oxides and the sixth level of this standard for carbon monoxide [5].

**Literature review**

Although it has two effects, increasing the percentage of anode-off gas increased total power. Because there are more inert gases in the air/fuel mixture, energy is diverted to heat rather than power, increasing stack count for higher power output while decreasing engine power. As the substitution ratio rose from 10% to 60% at utilization factor levels of 85%, 75%, and 65%, respectively, RCCI power dropped by 51%, 28%, and 16%. Increasing the system's overall power initially improved efficiency to a certain point, beyond which the benefits diminished. With a 32.5% substitution ratio for utilization factor 85%, the study's highest observed efficiencies were 52.8% [6].

Compared to both traditional diesel combustion and gasoline-diesel RCCI, methanol-diesel RCCI demonstrated promise in reducing the cytotoxic effects of particulate matter, especially under medium load settings. Furthermore, when compared to traditional diesel combustion, RCCI combustion demonstrated a reduction in the metal-related cancer risk potential at lower engine loads. Under medium load conditions, the RCCI combustion showed a reduction in emissions equal to the potentials for eutrophication, acidification, and global warming when compared to conventional diesel combustion.

Notwithstanding their benefits, RCCI combustion engines' non-regulated emissions showed higher levels of ozone-forming and cancer-risk potential than those of traditional diesel combustion engines. However, in RCCI engines, these potentials diminish as engine load increases. Both the ozone-forming potential and the cancer risk potential linked to non-regulated emissions from RCCI engines increase as the fuel premixing ratio is increased under constant load conditions [7].

While the rich premixed flame spread into the fuel-rich n-heptane/ammonia/air mixture because of low-temperature ignition, the lean premixed flame spread into the ambient ammonia/air mixture, greatly impacting combustion efficiency and the generation of nitrogen oxides. While the rich ammonia oxidation layer promotes ammonia oxidation and forms intermediate species like hydrogen, amino radicals, and nitrene radicals, the dual fuel oxidized nitrogen oxides and combustion intermediates. These species ultimately take part in the reactions in the dual fuel and rich premixed flame. Jet-induced vortices and turbulence, heat and radical transfer from the dual fuel, and n-heptane mixing all affected the back-supported propagation of the lean premixed flame. This impact was amplified by increasing the n-heptane jet speed, which improved the combustion efficiency of ammonia. While nitrogen oxides were produced in the lean premixed flame and rich premixed flame and consumed in the rich ammonia oxidation layer, they were mostly formed in the lean premixed flame and consumed in the dual fuel. Because of improved mixing and ammonia entrainment, higher n-heptane jet velocity increases the generation of oxides of nitrogen while accelerating the consumption of oxides of nitrogen. Gaining knowledge of these processes helped optimize RCCI combustion in ammonia-fueled marine engines for lower emissions and increased efficiency [8].

For the combination of 64.86% load and 50 ppm Copper oxide + 50 ppm Zinc oxide mixed fuel, the Deep Neural Nework and Gannet Optimization Algorithm produced the best braking thermal efficiency of 27.19% with acceptable levels of nitrogen oxides 437 ppm and hydrocarbon emissions 13.82 ppm. The work emphasizes how AI-driven optimization and nano-additives may improve the biodiesel performance of RCCI engines [9].

Elevated ignition delay values often indicated a decline in combustion stability as the amount of biogas added increased. Greater engine loads and greater diesel pilot ratios, however, lessened this adverse effect. Cylinder pressures, pressure rise rates, ignition delays, and ringing intensities all rose as the amount of biogas grew. On the other hand, peak heat release rates, combustion durations, and combustion stabilities all dropped. By encouraging the use of alternative fuels in internal combustion engines, this research advances sustainable engine technology practices and supports initiatives to improve energy efficiency and sustainability in the transportation industry [10].

The highest in-cylinder pressures for hydrogen and hydrogen blended CNG (HCNG) were 72 and 76 bar, respectively, at injector open time 8 ms of low reactive fuel.ES75% and 22% of low reactive fuels, hydrogen and HCNG, respectively, produced heat release rates of 86 and 87.9 J/deg. An ideal 75-80% and 30-40% energy substitution of HCNG and hydrogen low reactive fuel, respectively, was seen for clean combustion and emissions approach using the RCCI mode of a modified diesel engine. Hydrocarbon emissions for hydrogen and HCNG low reactive fuels were 42 and 138 parts per million, respectively, at peak performance. Lower carbon monoxide emissions of 0.04 and 0.02% were achieved with increased energy substitution of low reactive fuel [11].

In comparison to the basic condition, the gross indicated efficiency could be raised from 38.1% to over 50% by adding argon, hydrogen, and syngas. While a 20% hydrogen or syngas mass fraction was necessary to maintain low hydrocarbon emissions, a 10% argon mass fraction may be introduced to achieve maximum gross indicated efficiency and low hydrocarbon emissions. Consequently, under low load circumstances, argon has the most potential to enhance the performance of natural gas/diesel RCCI engines [12].

Maximum use of hydrogen fuel in a medium-duty twin-cylinder engine at low and moderate loads. For both traditional dual fuel and RCCI combustion modes, the effects of raising the hydrogen energy share on combustion, performance, and emissions were evaluated. Diesel fuel was used just for ignition; hydrogen was used as the main fuel. With only fuel injection restrictions, both modes were able to provide 90% hydrogen fuel at low load without experiencing combustion instabilities. Significant pollution reductions were seen without sacrificing performance, and RCCI outperformed dual fuel in terms of emissions. Knock limited mid-load operation to 40% hydrogen, resulting in lower emissions other than nitrogen oxides. Both modes showed no nitrogen oxides and very little smoke at low load with 90% hydrogen. As the percentage of hydrogen increased, brake thermal efficiency increased as well, with RCCI outperforming dual fuel at mid-load. Although the study demonstrated that high hydrogen fuel consumption, particularly at low load, has the potential to significantly reduce emissions, issues with knock at higher loads and fuel injector technology need to be resolved before broad adoption can occur [13].

Using polyoxymethylene dimethyl ether as the ignition trigger fuel decreased unburned ammonia by 13% as compared to diesel. An earlier injection start and a greater ammonia energy ratio were made possible by raising the intake temperature, which also increased the engine's claimed thermal efficiency. Polyoxymethylene dimethyl split injection or advanced injection start could increase spatial reactivity and reduce unburned ammonia by up to 71%. In order to get appropriate engine performance, the trade-off relationship between indicated thermal efficiency, unburned ammonia, and greenhouse gas could be overcome by using the three enhancement measures. Lastly, unburned ammonia fell by 79.3%, greenhouse gas emissions decreased by 25.4%, and indicated thermal efficiency increased by 7.56% all at the same time [14].

The local rapid heat release during the initial combustion stage was primarily responsible for the onset of pressure oscillations under conditions of inhomogeneous heat release; however, in conditions of relatively homogeneous and high-reactivity mixture, the end-gas autoignition was the primary source of pressure oscillations. With a first and second injection start of −60 and −15°CA aTDC, the double injection method may successfully reduce pressure oscillations and shorten the time it takes for pressure waves to propagate, in contrast to the single injection technique. However, because premixed hydrogen's higher fuel reactivity made end-gas autoignition worse, pressure oscillations sharply increased as the second injection start was moved to −30°CA aTDC. According to the frequency study of in-cylinder pressures, the first resonance frequency decreased as in-homogeneous combustion increased. Since the increased in-cylinder temperature brought on by the piston compression during the compression stroke facilitated the creation and propagation of pressure waves, the initial resonance frequency rose as the intake pressure was increased from 1.2 to 1.6 bar. The autoignition of end gas with enhanced fuel reactivity was directly associated with the high-order resonance. Although the linearity of hydrogen/polyoxymethylene dimethyl ethers RCCI was superior to that of hydrogen/diesel RCCI, overall, the combustion noise demonstrated a good linear association with peak heat release rate for all the locations examined in the work. Additionally, the engine noise related to piston compression increased as the in-cylinder pressure rose [15].

According to heat release rate analysis, phytol increased combustion efficiency in RCCI mode by up to 20%. Brake thermal efficiency was decreased, nevertheless, when the phytol blend was increased over 30%. Additionally, RCCI mode demonstrated higher emissions of hydrocarbons and carbon monoxide but lower emissions of smoke and nitrogen oxides. Compared to conventional techniques, improved homogeneity and ignition delay lead to increased peak pressure and heat release rate. The effects of copper oxide nanoparticles on a 20% combination of phytol and Cymbopogon martinii Methyl Ester B25 in RCCI mode were the subject of additional research. 50-100 ppm of copper oxide nanoparticles improved combustion parameters like cylinder pressure, ignition delay, and heat release rate, decreased pollutants, and increased brake thermal efficiency. According to the study's findings, a blend of Cymbopogon martinii Methyl Ester B25 and 20% phytol with 100 ppm copper oxide in RCCI mode was a viable substitute for conventional diesel fuel, providing an effective and sustainable energy source [16].

When gasoline was injected instead of a traditional diesel engine, the nitric oxide emissions were significantly reduced. When gasoline is added, a homogeneous mixture is formed, which greatly reduces the amount of soot generation. High emissions of carbon monoxide and hydrocarbons were a penalty; these emissions can be eliminated by regulating the timing of gasoline injection and guaranteeing uniform combustion. In comparison to a typical diesel engine, the injection of gasoline resulted in a drop in thermal efficiency and an increase in specific fuel consumption [17].

The RCCI mode of operation resulted in relatively poor brake thermal efficiency at all loads due to the important control needed to burn both low and high reactive fuels together and the reduced heat release rate (during the effective uncontrolled combustion stage). When compared to the homogeneous charge compression ignition mode of operation, the complete combustion characteristics with added butanol resulted in lower carbon monoxide emissions, while the rich mixing (mixture) characteristics of the RCCI mode produced higher hydrocarbon emissions under low loads that were unable to provide the necessary temperature. The cylinder temperature and pressure increased as a result of the ineffective heat generated during RCCI combustion. Under such operating mode, that suggests an undesirable scenario of increased emissions of smoke and nitrogen oxides [18].

As the high reactive fuel injection angle increased to 70° bTDC, the combustion stability improved. Under the same circumstances, the hydrogen premix ratio may be increased to 60% without impairing engine performance. A 50% premixed ratio and an upgrade in 70° bTDC high reactive fuel injection increased the brake thermal efficiency by 15%. Regarding base-diesel conventional diesel combustion operation, a 73% decrease in nitric oxide emissions, an 85% decrease in soot emissions, a 61% decrease in carbon monoxide emissions, and a 42% decrease in hydrocarbon emissions were also noted at the same operating point. As a result, the engine's low reactive fuel range was increased by high reactive fuel injection development [19].

When compared to alternative fuel combinations and combustion regimes, the reactivity regulated compression ignition engine fuelled by diesel and hydrogen had a greater braking thermal efficiency. For reactivity regulated compression ignition mode of combustion, the highest thermal efficiency of around 30.32% was achieved at 40% gaseous fuel energy sharing using diesel-compressed natural gas. As the energy share of gaseous fuels increases, so do the emissions of carbon monoxide and hydrocarbons. In comparison to homogenous charge compression ignition engines, reactivity regulated compression ignition engines that fueled hydrogen and hydrogen mixed compressed natural gas had lower emissions of hydrocarbons and carbon monoxide. As the proportion of energy from gaseous fuels grew, nitrogen oxides and smoke emissions declined. In comparison to homogenous charge compression ignition engines, reactivity regulated compression ignition engines driven by hydrogen and hydrogen blended compressed natural gas emitted fewer oxides of nitrogen. However, when compared to homogenous charge compression ignition engines, the smoke emissions from reactivity regulated compression ignition engines powered by hydrogen and hydrogen blended compressed natural gas were higher. When compared to homogenous charge compression ignition engines, the in-cylinder pressure rise and heat release rate of reactivity-controlled compression ignition engines driven by hydrogen and hydrogen mixed compressed natural gas were higher [20].

As the hydrogen addition increased from 0% to 40%, the ignition moments and the combustion end time advanced correspondingly. According to the emission analysis, adding hydrogen reduced emissions of carbon monoxide, total hydrocarbons, and soot while increasing emissions of nitrogen oxides. It's interesting to note that when the hydrogen content reaches 60%, the cylinder's heat release rate exhibits three peaks and its pressure increase rate surpasses the normal combustion value. This indicates that a detonation takes place in the cylinder, and the main combustion period is considerably altered in comparison to normal combustion. Because of the high temperature during combustion, a detonation with a greater hydrogen proportion significantly increases the emissions of nitrogen oxides [21].

Brake thermal efficiency dropped as the proportion of pentanol in injected fuels rose. When compared to other fuels, diesel and pentanol have the highest braking thermal efficiency among the evaluated fuel combinations. At 10% pentanol in injected fuels, a greater brake thermal efficiency of 22.15% was achieved for diesel and pentanol, which was around 9.1% and 27.3% higher than other fuel combinations. Emissions of carbon monoxide and hydrocarbons rose as the proportion of pentanol in injected fuels rose, whereas emissions of smoke and nitrogen oxides dropped. Compared to other fuel combinations, tested diesel and pentanol produce higher emissions of nitrogen oxides and lower emissions of smoke, carbon monoxide, and hydrocarbons. When compared to other fuel combinations, diesel and pentanol fuel combinations had the highest in-cylinder pressure and heat release rate at 10% pentanol in injected fuels [22].

By adding hydrogen to the ammonia-diesel reactivity controlled compression ignition combustion blend, emissions of carbon monoxide, hydrocarbons, and unburned ammonia are reduced while combustion efficiency and indicated mean effective pressure are significantly increased. In particular, a minimum intake valve closing temperature of 440 K was required while using an 80% ammonia energy component without hydrogen in order to prevent incomplete combustion, which would have increased nitrogen oxide emissions by about 20 g/kWh. However, the intake valve closing temperature requirement decreases to 380 K when 20% hydrogen energy portion is added to the fuel mixture, lowering nitrogen oxide emissions to about 13 g/kWh [23].

In comparison to 50% load for all fuel combinations, 75% load produced reduced emissions of hydrocarbons, carbon monoxide, and smoke, as well as improved brake thermal efficiency and oxides of nitrogen. Effective combustion results from the correct in-cylinder blending of the fuel and air mixture at 75% load. When compared to other fuel combinations, the diesel-compressed natural gas fuel combination produced reduced emissions of hydrocarbons, carbon monoxide, and smoke, as well as higher brake thermal efficiency and oxides of nitrogen at all loads. Diesel, the high reactive fuel, helped ignite compressed natural gas, the low reactive fuel, which was dispersed uniformly throughout the combustion chamber. In comparison to other fuel combinations under investigation, the diesel-compressed natural gas combination yields a greater brake thermal efficiency of approximately 29.74% at 75% load. Decreased reactivity of the dual fuel charge due to the charge dilution effect taking precedence when the charge's specific heat capacity increases [24].

Diesel and methyl ester-based dual fuel modes of operation for the same producer gas supply increased brake thermal efficiency by 4.01% and 6.4%, respectively, compared to the operation with homogenous charge compression ignition mode at 80% load. When compared to the operation with the same fuel combination on homogenous charge compression ignition mode at 80% load, the diesel-producer gas and dairy scum oil methyl ester-producer gas based dual fuel mode of operation provided somewhat lower hydrocarbon by 7.4% and 9.6%, respectively. Likewise, when operating in dual fuel mode for identical fuel combinations, carbon monoxide was slightly lower by 8.5% and 13.4%, respectively, than when using the homogenous charge compression ignition mode at 80% load. Smoke levels were consistently reduced while using the homogenous charge compression ignition mode of operation. When compared to the operation with homogenous charge compression ignition mode at 80% load, the dual fuel mode based on diesel and dairy scum oil methyl ester produced higher amounts of nitrogen oxides by

26.4% and 23.6%, respectively. With slightly increased emissions of carbon monoxide and hydrocarbons, the homogenous charge compression ignition engine produced reduced levels of smoke and nitrogen oxides. Lower thermal efficiency, however, came at the expense of that. Because of the decreased volumetric efficiency, homogenous charge compression ignition operation utilizing producer gas or gas with a lower energy content was not appropriate for applications needing high torque [25].

When compared to other injection pressures, 230 bar produced fewer emissions of hydrocarbons, carbon monoxide, and smoke, as well as improved brake thermal efficiency and oxides of nitrogen. When compared to alternative injection timings, the 26° bTDC injection timing produced fewer emissions of hydrocarbons, carbon monoxide, and smoke, as well as improved brake thermal efficiency and oxides of nitrogen. When compared to other nozzles, the five-hole nozzle produced fewer emissions of smoke, carbon monoxide, and hydrocarbons, as well as improved brake thermal efficiency and oxides of nitrogen. When evaluated against other fuel blends, the B20 mix outperformed diesel in terms of brake thermal efficiency, nitric oxide emissions, and hydrocarbon, carbon monoxide, and smoke emissions [26].

The computational viability and practical utility of simulations are guaranteed by a well considered choice of parameters, meshing schemes, and numerical techniques. With a 10% efficiency boost over the diesel engine, the methane-diesel reactivity controlled compression ignition engine had the greatest indicated thermal efficiency of 51%. The highest power obtained with an acceptable range of emissions and without being used after treatment was 7.03 kW, which was within 0.89 of the maximum allowable load of a conventional diesel engine. Reactivity controlled compression ignition engines were able to achieve high power, but at the expense of high nitrogen oxide emissions that were below standards. Every possible result fell between 2.22 and 2.37 for the access air ratio, 9.45 to 9.87 mg for the methane mass, 1.71 to 2.44 mg for the diesel mass, 1.99 to 3.36 crank angle for the duration, −34 to 35° aTDC for the injection start, and 2020 to 3010 rpm for the rotational speed. The corresponding minimum results for carbon monoxide, hydrocarbon emissions, and indicated specific fuel consumption were 0.00543, 0.0187, and 146 g/kWh, respectively. These results were 99%, 72%, and 28.7% lower than those of conventional diesel engines. When the cylinder temperature was at its lowest and remained there for a brief amount of time, the least amount of nitric oxide emissions were produced. This was 97.5% less than a traditional diesel engine and fell below the EURO IV limits of 0.4 g/kWh. When combustion progressed and the in-cylinder temperature in the high range was higher than in other chosen ideal circumstances, the least amount of soot emissions were produced. Using methane diesel in internal combustion engines is a viable and affordable strategy, as evidenced by the improvement in indicated specific fuel consumption and decreased emissions [27].

With a brake thermal efficiency of approximately 29.32% at 50° aTDC injection timing, the diesel-compressed natural gas mixture outperformed the diesel-compressed biogas, B20-compressed natural gas, B20-compressed biogas, and B100-compressed natural gas combinations by a margin of 1.77, 3.58, 5.56, 7.51, and 8.54%, respectively, among the various injection timings examined. At 6 ms injection duration, the diesel-compressed natural gas fuel combination had the highest brake thermal efficiency, approximately 30.25%. This was 1.69, 3.48, 5.32%, 7.24, and 9.16% higher than the diesel-compressed biogas, B20-compressed natural gas, B20-compressed biogas, B100-compressed natural gas, and B100-compressed biogas fuel combinations. In comparison to other fuel mixes, the diesel-compressed natural gas mixture exhibited lower emissions of smoke, carbon monoxide, and hydrocarbons and higher emissions of nitric oxide for all injection timings and durations. In comparison to other fuel combinations, diesel-compressed natural gas produced higher in-cylinder pressure and heat release rate for all injection timings and durations [28].

Peak cylinder pressure increased when the premixed ratio based on the same total lower heating value was increased. This was because the amount of fuel supplied into the crevice volume with extending the start of energizing timing was decreased. Furthermore, the low cylinder temperature caused by the ethanol's evaporation latent heat resulted in a longer ignition delay and decreased compression loss, which raised the stated mean effective pressure value. Because of the narrow distribution of the rich equivalency ratio in the cylinder, the cylinder temperature decreased, which in turn decreased the amount of nitric oxide generation. Due to poor ethanol flame propagation from diesel combustion and the slow oxidation of the carbon monoxide produced by combustion due to the low temperature of the cylinder caused by the evaporation latent heat of ethanol, the carbon monoxide value increased the premixed ethanol ratio based on the same total lower heating value in-cylinder. According to these findings, the ideal operating parameters for lowering exhaust emissions and enhancing combustion performance were determined to be 23° bTDC for the start of energizing timing and 40% for the premixed ethanol ratio based on the same total lower heating value in-cylinder [29].

As pilot fuel injection time improved, the dual fuel engine's brake thermal efficiency when running on biogas increased as well. Because pilot fuel had enough time to create a uniform flammable mixture, biogas burned more effectively. Because B100 blends have a larger calorific value and a lower viscosity than B20 blends, more smoke emissions were seen from the pilot fuels. Compared to B20 and B100 with raw biogas, the hydrocarbon emissions from B20 and B100 with purified biogas were lower. Carbon monoxide emissions were reduced by 11.64% on average when injection timing was advanced from 19 to 31° bTDC. Purified biogas exhibited higher emissions of nitrogen oxides than raw biogas. This was because raw biogas had a lower calorific value and a higher percentage of carbon dioxide than purified biogas, which lowered the combustion temperature. Because more fuel was delivered into the engine cylinder as the injection timing was advanced, the ignition delay period lengthened. Because biogas has a higher calorific content than B100, it exhibited a higher peak pressure than B20 [30].

Up until a certain point, advancing the injection start boosted mixture homogeneity; after that, no more benefits were seen, according to fuel distribution visualizations using local equivalency ratios. However, the voided region was equally discernible in the event of greater homogeneity. It has been found that greater in-cylinder temperatures linked to late injection timings result in increased hydrocarbon consumption and, as anticipated, higher nitric oxide production. A higher percentage of fuel mass was found near the liner for an early injection start, according to an analysis of the spatial distribution of high reactive fuel with injection timing. The fuel distribution, however, is progressively closer to the cylinder axis for injections that occur later than this number. High localized in-cylinder temperature zones that approach >2000 K are caused by later injection timing. As a result, nitrogen oxide emissions exceed the baseline value by more than 100 times. In late injection techniques, greater in-cylinder temperatures result in increased hydrocarbon consumption. The recommended mean effective pressure is higher in late injection situations. Therefore, for an injection timing of 48 CAD bTDC, the maximum indicated mean effective pressure was achieved. A fuel distribution visualization using the local equivalency ratio shows that a later injection start results in less spatial inhomogeneity. However, with injection timings of 80 CAD bTDC, that impact levels off [31].

For a given hydrogen flow rate, diesel engines fueled by B20 blends of biodiesel had a higher brake thermal efficiency than those powered by B100 blends. When compared to their respective B100 biodiesels, the brake thermal efficiency of dual fuel engines powered by B20 blends of nigella sativa and jack fruit biodiesel B20 rose by 5.77% and 7.91%, respectively. During induction, the smoke emissions somewhat increased at lower gaseous fuel flow rates. However, compared to B20 fuels, B100 blends showed higher smoke emissions for the pilot fuels. Compared to jack fruit biodiesel B20, nigella sativa biodiesel B20 has a higher smoke opacity. When compared to their respective B100 biodiesels, the smoke emissions of B20 blends of nigella sativa and jack fruit biodiesel powered by dual fuel engines dropped by 9% and 10.17%, respectively. As the pilot injected fuels were found to be lower, hydrocarbon emissions decrease with increased hydrogen gas flow rates. Dual fuel engines fueled by nigella sativa biodiesel B20 have higher hydrocarbon emissions than jack fruit biodiesel B20, despite hydrogen induction being widespread. Higher hydrocarbon levels occur in dual fuel mode when hydrogen flow rates exceed 0.2 k/h. When compared to their respective B100 biodiesels, the hydrocarbon emissions of B20 blends of nigella sativa and jack fruit biodiesel, which are used in dual fuel engines, dropped by 11.11% and 15%, respectively. Compared to jack fruit biodiesel B20 with hydrogen induction, nigella sativa biodiesel B20 exhibits fewer carbon monoxide emissions. The carbon monoxide increased when the hydrogen flow rate exceeded 0.2 kg/h. When compared to their respective B100 biodiesels, the carbon monoxide emissions of the B20 blends of nigella sativa and jack fruit biodiesel powered by dual fuel engines dropped by 27.27% and 23%, respectively. When compared to diesel engines that use hydrogen induction, biodiesel-powered dual fuel engines exhibit reduced emissions of nitrogen oxides. However, when compared to B100 fuel operation, B20 blends exhibit a relatively higher quantity of nitrogen oxide emissions. In dual fuel engines, the nigella sativa biodiesel B20 exhibits higher nitrogen oxide emissions than the jack fruit biodiesel B20 with hydrogen induction. When compared to their respective B100 biodiesels, the oxides of nitrogen emissions of the B20 blends of nigella sativa and jack fruit biodiesel powered by dual fuel engines rose by 5% and 3.69%, respectively. For all fuel combinations, the dual fuel engine's peak pressure rose as the hydrogen flow rate rose. When compared to pure biodiesels, the hydrogen in ordinary B20 blends exhibits higher peak pressures. In dual fuel engines, the nigella sativa biodiesel B20 exhibits greater peak pressures than the jack fruit biodiesel B20 with hydrogen induction. For all fuel combinations, the engine exhibits knocking behavior and lower peak pressures at high hydrogen flow rates exceeding 0.2 kg/h. When compared to their respective B100 biodiesels, the peak pressure of the B20 blends of nigella sativa and jack fruit biodiesel for dual fuel engines rose by 4.2% and 3.55%, respectively [32].

During the suction stroke, compressed natural gas and compressed biogas were injected with air; during the compression stroke, diesel, B20, and B100 blends of thevetia peruviana methyl ester were injected. For all fuel combinations, increased brake thermal efficiency, peak pressure rise, heat release rate, and nitrogen oxide emissions were noted at 40% gaseous fuel energy share, whereas reduced hydrocarbon and carbon monoxide emissions were noted. Out of all the fuel combinations that were evaluated, compressed natural gas produced lower emissions of carbon monoxide and hydrocarbons and higher values of braking thermal efficiency, peak pressure rise, heat release rate, and nitrogen oxides. When compared to diesel, the reactivity-controlled compression ignition engine running on thevetia peruviana methyl ester fuel performed worse. However, the demand for diesel fuel was reduced by the use of thevetia peruviana methyl ester, which allowed for limited diesel fuel replacement [33].

Improving the timing of pilot-injected fuel injections lowers emissions of carbon monoxide, hydrocarbons, and smoke while increasing the brake thermal efficiency of dual fuel engines with biogas induction. In the dual fuel engine running on biodiesel and biogas, respectively, improving the injection timing decreased the ignition delay and increased the peak pressure. When compared to algae biodiesel B100, biogas-inducted dual fuel engines running on the B20 blend demonstrated improved thermal efficiency and nitrogen oxides by 4.37% and 2.9%, respectively, and decreased smoke, hydrocarbons, and carbon monoxide by 5.4%, 13.63%, and 2%. Peak pressure rose by 3% and ignition delay dropped by 9.5%, respectively. The engine's performance was significantly enhanced by the addition of graphene oxide nanoparticles to the algal biodiesel blend. When compared to diesel fuel for its commercial applications, the synthesis, characterisation, and stability of the nano-biodiesel blends were crucial factors [34].

Engines run smoothly thanks to the B20 mixes of Nigella sativa and Ceiba pentandra oils. In comparison to diesel base line operation, the performance of diesel engines using two B20 mixes was lower. The Ceiba Pentandra Oil Methyl Ester B20 Blend exhibits reduced engine emission characteristics and comparatively better engine performance when compared to the other test fuels. With both B20 blends, the engine performance is enhanced by the use of swirl techniques. When cylinder cavities and domes were increased from 2C-1D to 3C-2D, fewer emissions and higher brake thermal efficiency were noted. The engine's performance did not much improve with additional dome and cavity expansion. It was determined that an engine powered by Ceiba Pentandra oil methyl ester B20 with a cylinder having 3C-2D cavities and domes might yield satisfactory results [35].

Utilizing exhaust gas recirculation had a major impact on engine performance and reduced nitrogen oxide levels. In comparison to dual fuel combustion powered by dairy scum oil methyl ester, the diesel-producer gas combustion demonstrated a 12.5% increase in brake thermal efficiency. In contrast to the dairy scum oil methyl ester-producer gas operation, the dual fuel combustion powered by a 5% exhaust gas recirculation induction and 10 lpm hydrogen showed a 6.1% recovery in brake thermal efficiency. Additionally, diesel and dairy scum oil methyl ester-fueled dual fuel combustion provided power derating by 2.4% and 16.1%, respectively, at 80% load with the same 10 lpm of hydrogen and 10% exhaust gas recirculation rate as the operation with 5% exhaust gas recirculation rate. Additionally, compared to diesel-producer gas operation with the same 10% exhaust gas recirculation rate, the thermal efficiency of dairy scum oil methyl ester-producer gas combustion with hydrogen at a flow rate of 10 lpm and 5% exhaust gas recirculation was somewhat lower by 2.1%. In comparison to the same fuel combination with a 10% exhaust gas recirculation rate, the combustion of dairy scum oil methyl ester producer gas with fixed hydrogen addition and a 5% exhaust gas recirculation rate showed 10.2% lower smoke, 8.01% lower hydrocarbon, 14.5% lower carbon monoxide, and 32.2% higher oxides of nitrogen emission levels. Smoother engine operation was achieved by running the engine under ideal operating conditions for six hours. The optimization of producing gas-fueled dual-fuel engines by the installation of a turbocharger with an increased compression ratio to improve performance and emissions is the focus of potential advances [36].

When compared to diesel operation, common rail direct injection engines operating at an advanced injection timing of 17° bTDC demonstrated improved brake thermal efficiency, decreased emissions of smoke, hydrocarbons, and carbon monoxide, respectively, and increased emissions of nitrogen oxides. The addition of the antioxidant pyrogallol to B20 blends of fish oil biodiesel enhanced engine performance by lowering emissions of carbon monoxide, hydrocarbons, and smoke, respectively. When compared to B20 blends of fish oil biodiesel, the common rail direct injection engine operating with fish oil biodiesel B20-PG3 under the optimized conditions of 17° bTDC, injection pressure 600 bar, 6-hole injector, and torroidal re-entrant combustion chamber increased brake thermal efficiency by 3.8%, 8.54%, and reduced smoke by 15.22%, 43.07%, hydrocarbon by 21.36%, 49.23%, and carbon monoxide by 12.5%, 34.8% [37].

Compared to prior attempts, the 9 mm size venturi increased the brake thermal efficiency for various fuel combinations. A nearly stoichiometric air-gas ratio may result from the uniform mixing of air and introduced gases. Because B100 blends have a larger calorific value and a lower viscosity than B20 blends, more smoke emissions were seen from the pilot fuels. For both compressed natural gas and biogas dual fuel engine operation, the jamune oil methyl ester B20 demonstrated reduced hydrocarbon emissions in comparison to the jamune oil methyl ester B100. Comparing the 9 mm venturi to other efforts examined for all the fuel combinations in dual fuel mode, lower carbon monoxide emissions were noted. Because compressed natural gas has a lower burning temperature than biogas, it emitted more nitrogen oxides. As the number of holes on the venture increased, the ignition delay period shortened. Because the former fuel has a larger calorific value than biogas, the jamune oil methyl ester, which is the same compressed natural gas, exhibits a higher peak pressure [38].

A possible backup strategy that would also aid in meeting the world's energy needs was adding water to diesel, which decreased emissions of nitric oxide and particulate matter while improving combustion efficiency. Additionally, the full combustion of the fuel mixtures was guaranteed by the injection of alcohol. In order to examine the impact of ethanol on the performance of a modified diesel engine, ethanol was injected into the intake manifold using two distinct emulsions with varying water-to-diesel ratios at an injection timing of 5° aTDC and an injection duration of 3 ms. According to the test results, during 80% load conditions, a higher brake thermal efficiency of 31.25% was achieved with 20% water in the diesel emulsion and manifold-injected ethanol. Engine operation on water-in diesel emulsion with ethanol manifold injection resulted in a 10.3% increase in brake thermal efficiency over baseline diesel operation. When the engine was powered by a 20% water-in-diesel emulsion with ethanol pumped into the manifold at 80% load, fewer smoke emissions were produced. When ethanol injection was used instead of baseline diesel operation, the smoke emissions decreased by 27.9%. When the engine was fuelled with a 20% water emulsion in diesel with manifold injected ethanol, the emissions of nitrogen, carbon monoxide, and hydrocarbon oxides decreased by 10.8%, 24.3%, and 21% at 80% load, respectively, in comparison to baseline diesel operation. Better engine performance and emissions were obtained in compression ignition engines by using a 20% water-in-diesel emulsion with numerous injections of ethanol, provided at the ideal injection timing and injection length [39].

The performance of the engine will be enhanced by making it completely adiabatic by mixing biodiesel with diesel and coating the piston, cylinder wall, and inlet and outlet valves with thermal barrier coatings (TBC). Accordingly, Karanja oil methyl ester (KOME) and its mixes were used to evaluate TBC diesel engine components coated with partially stabilized zirconia (PSZ) and aluminum oxide (Al2O3) ceramics in a diesel engine. Brake thermal efficiency (BTE) emissions from diesel and KOME and its blends were measured both before and after coating, and the results were consistent. With a PSZ coated engine and B20 fueled biodiesel, engine performance is almost equal with acceptable emission standards. The highest BTE among the tested biodiesels was found in diesel, which was 2.79%, 5.89%, 7.75%, 9.64%, and 14.03% higher than fuel blends B10, B20, B30, B40, and B100. Among the evaluated fuels, diesel had the lowest smoke emissions, which were 6.06%, 12.9%, 20.68%, 29.62%, and 39.88% lower than those of the B10, B20, B30, B40, and B100 fuel blends [40].

When compared to the same fuel combination without nanoparticles, the 40 ppm dairy scum oil methyl ester-multi walled carbon nanotubes with producer gas induction increased thermal efficiency by 9.1% while lowering emissions. Nevertheless, additional nanoparticle growth in a methyl ester of dairy scum oil has no beneficial effects. When compared to a diesel-producer gas combination, a dual fuel engine running at 80% load on dairy scum oil methyl ester with producer gas induction produced 5.7% lower brake thermal efficiency, 17.9% higher smoke opacity, 14.1% higher hydrocarbon, 12.4% higher carbon monoxide, and 8.6% lower oxides of nitrogen emission levels. In comparison with dairy scum oil methyl ester - 40 ppm - producer gas, the combustion of dairy scum oil methyl ester - producer gas and dairy scum oil methyl ester - producer gas with 20 and 60 ppm of multi walled carbon nanotubes nanoparticles resulted in decreased brake thermal efficiency by 4.08% and 7.1%, increased smoke opacity by 15.2% and 9.5%, increased hydrocarbon emissions by 7.8% and 12.5%, increased carbon monoxide pollutants by 11.6% and 17.8%, and amplified nitrogen pollutants by 12.1% and 5.6%, respectively, at 80% applied load. When compared to dairy scum oil methyl ester-producer gas, both with and without the addition of nanoparticles, the diesel-producer gas combustion demonstrated higher cylinder pressure and heat release rate. Pilot fuel savings ranged from 56 to 66% with dual fuel operation. At 80% load, the greatest diesel replacement achieved was 66% [41].

When it came to brake thermal efficiency, the B20 blend outperformed the B100 biodiesel. As biogas gas flow rates increased, so did smoke emissions. B100 produced more smoke than B20 fuels at the same gaseous fuel flow rates. Compared to biodiesel B20 and B100 with biogas combinations, diesel and biogas combined demonstrated lower hydrocarbon emissions. In comparison to B100, B20 biodiesel demonstrated fewer hydrocarbon emissions when combined with biogas. It was discovered that engines running on B20 biodiesels and filtered biogas had lower carbon monoxide emissions than those running on B100 and raw biogas. It was discovered that engines running on B20 biodiesels and purified biogas had higher nitrogen oxide emissions than those running on B100 and raw biogas. As the biogas flow rates for dual fuel engines increased, the peak pressure dropped and the ignition delay rose. In comparison to B100 and raw biogas, respectively, the B20 and purified biogas operation demonstrated higher peak pressure and smaller ignition delay [42].

The purpose of the study was to examine how a reactivity-controlled compression ignition engine was affected by low reactive fuel injection timing and duration. Low reactive fuels included ethanol, plastic pyrolysis oil, diesel, and combinations of karanja oil and methyl ester. The injection timings for the low reactive fuels under investigation are 40° aTDC, 45° aTDC, and 50° aTDC. The study also examined how the reactivity regulated compression ignition mode of combustion was affected by various low reactive fuel injection durations of 3, 6, and 9 ms. 75% of the rated power was used throughout the entire experiment. The results showed that the diesel-gasoline mixture had the highest braking thermal efficiency of all the injection timings and injection durations tested at 45° aTDC and 6 ms. While smoke, carbon monoxide, and hydrocarbon emissions were lower for the diesel-gasoline mixture than for other fuel blends, nitric oxide emissions were higher for all injection timings and durations. At all injection timings and injection durations, the diesel-gasoline fuel mixture increased peak pressure more than other test samples [43].

The performance of a dual fuel engine powered by hydrogen was examined in relation to nano-biodiesel mixes and injection time. The flow rate of hydrogen was kept constant at 0.15 kg/h. B20 and nano-blends of nigella sativa oil methyl ester and Jack fruit seed oil methyl ester fuel combinations were utilized for that. Using the probe sonication approach, graphene amine nanoparticles in different proportions ranging from 60 to 100 ppm were used to create nano-blends of the corresponding B20 biodiesels. The amount of nanoparticles, the amount of sodium dodecyl sulfate utilized as a surfactant, and the duration of sonication were used to guarantee the stability of the nano-blends. Due to better combustion than B20 biodiesel blends, adding nanoparticles to B20 biodiesel blends up to 80 ppm significantly enhanced the performance of diesel engines running on a single fuel. Non-homogeneous mixes of nano-biodiesel blends caused the performance to decline above 80 ppm. The redesigned dual fuel engine's injection time was adjusted in 4° bTDC increments from 19 to 31° bTDC. When compared to B20 operation, dual fuel engine performance with B20 + GA80 blends of both biodiesels and hydrogen induction was enhanced by further increasing the injection time from 19 to 27° bTDC. Using computational fluid dynamics research, further air-hydrogen mixing in the dual fuel engine's input manifold is investigated for a range of hydrogen flow rates, from 0.10 to 0.2 kg/h in steps of 0.05 kg/h [44].

The brake thermal efficiency and nitrogen oxide emissions dropped as the exhaust gas recirculation percentage rose from 0% to 10%, but carbon supported and smoke exhaust did not. Performance and emission parameters improved as the load climbed from 6 to 12 kg. Diesel outperforms the other pilot fuels being tested in terms of performance and emission characteristics. Hydrogen was the most efficient fuel to replace fossil fuels in diesel engines. Under homogenous charge compression ignition operation, hydrogen and B20 Mahua oil methyl ester were the most appropriate fuel combination for compression ignition engines [45].

Brake thermal efficiency declined as the proportion of n-butanol in injected fuels rose. Diesel and n-butanol outperformed other fuel combinations in terms of brake thermal efficiency. The maximum braking thermal efficiency was achieved at 10% n-butanol injection. Carbon monoxide and hydrocarbons rose as the proportion of n-butanol in injected fuels increased. As the amount of n-butanol in injected fuels grew, the emissions of smoke and nitrogen oxides decreased [46].

In comparison to B20 and B100 producer gas fuel combinations, the braking thermal efficiency of a reactivity-controlled compression ignition engine running on diesel was higher. At an energy contribution of 40% gaseous fuels, the highest braking thermal efficiency was achieved. As the energy share of gaseous fuels increased, so did the emissions of smoke, carbon monoxide, and hydrocarbons. Compared to B20 and B100 producer gas, the hydrocarbon and carbon monoxide emissions from diesel produced by reactivity-controlled compression ignition engines were lower. As the energy proportion of gaseous fuels rose, the emissions of smoke and nitrogen oxides fell. Reactivity-controlled compression ignition engines powered by diesel produced greater nitric oxide emissions than those powered by B20 and B100. Diesel-powered compression ignition engines have higher in-cylinder pressure rises and heat release rates than B20- and B100-powered engines. The maximum thermal efficiency and lowest emissions were achieved at 40% of the energy share of gaseous fuels with diesel-producer gas [47].

With biogas and diesel as fuels, the reactivity regulated compression ignition mode of operation of an internal combustion engine has the potential to provide high efficiencies and low emissions of soot and nitrogen oxides. The wall wetting of the diesel that had to be injected early in the compression stroke was one of the disadvantages of converting conventional diesel engines to operate in that mode. This results in substantial emissions of total hydrocarbons and dilution of lubricating oil. In that experiment, a twin-cylinder turbocharged common rail diesel engine was adapted to accept both the current wide angle injector and a recently designed narrow angle injector enabling early diesel injection without wall wetting. Additionally, the engine was operated in the reactivity regulated compression ignition mode and biogas was pumped into the intake manifold. This innovative twin injector dual fuel reactivity controlled compression ignition idea has been evaluated experimentally at various biogas energy shares with a constant engine speed of 1500 rpm and a brake mean effective pressure of 3 bar. The conventional dual fuel mode, reactivity controlled compression ignition with the wide angle injector alone, reactivity controlled compression ignition with the narrow angle injector alone, and reactivity controlled compression ignition with the combined narrow angle injector and wide angle injector were the various biogas-diesel modes that were taken into consideration. Due to diesel cylinder wall wetness, using only the wide angle injector in the biogas diesel reactivity regulated compression ignition mode produced low efficiency, high total hydrocarbon emissions, and smoke. Although cylinder wall wetting could be avoided by using simply a narrow angle injector in this mode, poor performance was caused by the diesel striking the piston bowl directly. In contrast to the traditional dual fuel mode, the combination of narrow angle and wide angle injectors produced good brake thermal efficiency, very low levels of nitrogen oxide emissions, and decreased total hydrocarbon. When the energy share of biogas was altered, the injection timing range for the narrow angle injector was 90° to 45° bTDC, and for the wide angle injector, it was 42° to 4° bTDC. In order to operate at significantly variable injection timings as needed in biogas diesel reactivity regulated compression ignition operation, two injectors with wide and narrow injection spray angles for diesel were used [48].

Without changing the injection parameters, chemically stabilized waste plastic oil can be effectively used in the reactivity controlled compression ignition combustion concept. With the right control parameters, ultra-low emissions of nitrogen oxides and particulate matter can be achieved with the fuels used. Oxides of nitrogen and particulate matter emissions were reduced for diesel-methane mix by 82.0% and 93.2%, respectively, when compared to conventional compression ignition operation. In contrast, oxides of nitrogen and particulate matter emissions were reduced for waste plastics oil-methane mix by 88.7% and 97.6%, respectively. This can be attributed to the waste plastics oil's favorable chemical characteristics for the combustion concept. The mean effective pressure covariance was maintained at 2.5% in the least favorable operating position among those examined, which was significantly below 5%, which is regarded as the limit for stable engine running [49].

While exhaust gas temperature dropped, brake thermal efficiency, brake-specific fuel consumption, and brake-specific energy consumption all increased. The average decrease in carbon dioxide and nitric oxides was 20.7% and 61.36%, respectively. On the other hand, compared to petrodiesel, carbon oxide emissions increased by 22.9% and hydrocarbon emissions by 33.1% on average, with the exception of the blend of biodiesel with a high diethyl ether concentration, which demonstrated a notable decrease in hydrocarbon emissions of 83.3%. Alternative fuel modes' combustion parameters are fairly comparable to those of diesel fuel combustion [50].

The goal of this work is to determine the properties of an RCCI engine that uses methane-enriched hydrogen and spirulina microalgae biodiesel as low and high-reactive fuels, respectively. Eighty percent of the hydrogen refueling fuel is made by combining diesel, dimethyl ether, and microalgae spirulina biodiesel in the following ratios. Diesel, biodiesel, and dimethyl ether account for 48% of the total. The effects of employing methane-enriched hydrogen as 20% of the low-reactive fuel will be investigated by varying the engine's settings. The low reactive fuel blend's methane to hydrogen ratios can vary from 1% to 5%, with corresponding concentrations of 1%, 2%, 3%, 4%, and 5%. Additionally, the ratio may fall anywhere in the middle. The characteristics analysis is conducted by varying the engine's fuel injection pressure and brake application speed while keeping the compression ratio at 17.5:1. This categorization comprised the features of brake thermal efficiency, brake-specific fuel consumption, exhaust gas temperature, and emissions of oxides of nitrogen, carbon dioxide, smoke, and hydrocarbons. The results of the experiments show that using fuel mixtures has the potential to lower engine emissions while also improving engine performance [51].

A modified dual-fuel compression ignition engine was used to conduct an experimental and simulation research of the reactivity regulated compression ignition combustion method of coal-based fuels. It was discovered how methanol and coal-based fuels, which differ significantly in reactivity, enhance engine combustion and thermal efficiency. The findings demonstrate that when the methanol load increased, the reactivity gradient inside the cylinder rose, the ignition region's area quickly expanded, and the ignition region's reactivity increased. Soot and nitrogen oxides decreased by a maximum of 25% and 15%, respectively, while hydrocarbon and carbon monoxide emissions increased by 1350% and 6866%. The ignition region's reactivity rose by 8% to 26%. Reactivity and diffusion combustion have a strong linear relationship, with the premixed-to-diffusion combustion ratio being reduced by up to thirty times. There was a maximum rise of 18% in the concentration of the exothermic center toward the top dead center, a maximum increase of 90% in the coefficient of variability, a maximum increase of 22% in indicative thermal efficiency, and a maximum increase of 17% in brake thermal efficiency [52].

Compared to conventional biodiesel combustion, the 30% compressed natural gas share increased thermal efficiency by 4.35% and reduced nitric oxide and smoke by 25 and 31%, respectively. In order to predict the dependent variables separately from the independent variables, two machine learning models were created: the least absolute shrinkage and selection operator regression and the gradient boosting regressor [53].

The combustion phase of the reactivity controlled compression ignition mode was progressively delayed, the ignition delay and combustion duration were extended, the combustion temperature decreased, the mixture distribution became more uniform, and the combustion process became smoother as the exhaust gas recirculation ratio increased.

Brake thermal efficiency dropped from 45.45 to 43.19%, oxides of nitrogen generation declined by 47.3%, and ringing intensity and maximum pressure rise rate decreased by 1.31 MW/m2 and 0.37 MPa/°CA, respectively, as the exhaust gas recirculation ratio rose from 0 to 30%. The combustion start point was progressively delayed, the peak heat release rate and combustion temperature continuously decreased, and the indicated mean effective pressure and brake thermal efficiency both showed a downward trend as the compression ratio was reduced. The change in compression ratio improves combustion stability and breaks the trade-off connection between soot and nitrogen oxides, but it is also constrained by the decrease in braking thermal efficiency. Brake thermal efficiency dropped from 45.45 to 44.52% and oxides of nitrogen and soot output decreased by 28.58 and 20.1%, respectively, when the compression ratio dropped from 17.5 to 15.5 [54].

**Conclusion**

For current engines, Reactivity Controlled Compression Ignition (RCCI) is a highly adaptable and effective combustion approach that can achieve strict emission and fuel economy requirements. The selection and combination of low- and high-reactivity fuels are crucial in defining combustion characteristics, performance, and emission consequences, according to a review of the literature. While high-reactivity fuels like diesel or biodiesel start regulated autoignition and allow accurate combustion phasing, low-reactivity fuels like gasoline, ethanol, or natural gas efficiently prolong the ignition delay and encourage homogeneous charge generation. Achieving low $NO_x$ and particulate matter emissions without sacrificing thermal efficiency requires an ideal balance between the reactivity difference, fuel ratio, and injection approach. Sustainability and carbon reduction potential are further improved by the use of renewable and alternative fuels within RCCI frameworks. Fuel reactivity matching, transient combustion control, and system integration for real-world applications are still difficult tasks, nonetheless. To increase the working range and stability of RCCI engines, future research should concentrate on sophisticated fuel modeling, real-time combustion control, and hybrid fuel compositions. RCCI has the potential to be a crucial step toward clean, effective, and fuel-flexible internal combustion engines with further developments in fuel technology and engine control.

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