**Performance, Emission and Combustion Characteristics of Various Combustion Modes – A Review Approach**

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**Abstract**

Concern about diesel engine emission and growing global need for energy in case of transportation sector have made alternative fuels for internal combustion engines more significant. Numerous researchers have looked into alternative fuels in the dual fuel mode in order to optimize emission profiles and energy consumption for transportation and distribution activities. This review article comprehensively compares the engine performance, combustion, and emission characteristics of alternative fuels in conventional, dual fuel, and reactivity regulated compression ignition (RCCI) mode combustion. The RCCI engine had favourable emission characteristics due to differences in fuel qualities, spray droplet size distribution, and subsequent mixing with ambient air. The RCCI technique is capable of controlling the combustion phase, peak pressure rise and heat release rate through regulating the reactivity stratification to attain the resolution of combustion process optimization.

**Keywords:** Conventional, Dual fuel, RCCI, Performance, Combustion, Emission.

**1. Introduction**

Population expansion is increasing energy demand in the transportation sector, while economic policies strive to increase efficiency and reduce dangerous pollutant emissions such as nitrogen oxides, unburned hydrocarbons, and particulate matter. Clean, high efficiency engines are required to meet the strict emissions rules and deliver power effectively. The effectiveness of modern engines has been increased by the investigation of numerous approaches. These include innovations like variable valve timing, which aims to lower pumping losses related to the gas exchange process, and variable geometry turbochargers, which aims to increase the power density of engines by utilizing exhaust energy. Additionally, more sophisticated fuel injection systems have been put into use to inject gasoline at higher pressures and thereby encourage fuel and air mixing. Increased mixing will boost combustion performance while lowering particulate emissions. The development of dual fuel combustion techniques can also make use of more sophisticated fuel injection systems. Both compression ignition and spark ignition engines have shown to benefit from dual fuel combustion techniques [1-17]. Dual fuel injection systems have lately been employed to promote the use of less reactive fuels and to enable more advanced combustion strategies. Some dual fuel combustion modes have showed substantial promise in terms of efficiency and pollution emission. This is frequently accomplished throughout a large operating range by using two fuels with different concentrations at the same time to induce premixing of the fuel or stratification of the reactivity of the in-cylinder mixture. Dual fuel injection techniques have historically been employed on compression ignition engines to convert old diesel engines to run on less expensive fuel. The implementation allowed for lower particulate matter emissions in addition to the use of a different power source. Although dual fuel engines have the potential to be extremely efficient and environmentally friendly, their use may also be constrained by infrastructure issues and customer acceptance. Users will need to fill up two fuel tanks, and they must have access to the necessary fuels over a sufficiently large area [18-30]. The RCCI technique is capable of controlling the combustion phase, peak pressure rise and heat release rate through regulating the reactivity stratification to attain the resolution of combustion process optimization, stimulating thermal efficiency and decreasing engine-out emissions. RCCI mode of combustion is emerged from dual fuel combustion in which two fuels of variant reactivity are used to increase the process of combustion and diminish the engine-out emissions. RCCI combustion characteristics are controlled by changing the fuel quantity of the charge. Low reactive fuel is most important key factor that affects the performance and combustion characteristics of RCCI engine. It is not only affects the mixing of fuel and air inside the combustion chamber, but also affects the processes of heat transfer and heat release. At initial stage of RCCI combustion study, gasoline is used as the low reactive fuel and it could achieve lower emissions of nitric oxide and soot along with higher indicated mean effective pressure. In latest years, alcoholic fuels are used as the low reactive fuels for RCCI combustion mode and gained huge attention from various researchers due to their outstanding physico-chemical properties [31-39].

**2. Exhaustive review on conventional, dual fuel and RCCI combustion modes**

**2.1 Conventional mode of combustion**

* **Brake thermal efficiency (BTE)**

At 80% load, the maximum BTE values for diesel and methyl ester of rice bran oil were determined to be 31.09% and 28.27%, respectively [3]. Injection timing of 26°BTDC was shown to be more thermally efficient for 3-hole nozzles than injection timings of 23°BTDC and 20°BTDC [4,11,15]. The BTE increased with increasing load for all of the fuels examined. At all power outputs, the BTE of biodiesel blends was found to be lower than that of diesel. At 80% load, all evaluated fuels had higher BTE than at 100% load [5-8, 16]. Among the many nozzles examined, the 5-hole nozzle provided the highest BTE [9]. The karanja B20 blend’s BTE was 25.52%, which was higher than that of other biodiesel mixes [10]. For diesel, simarouba oil methyl ester, and hippe oil methyl ester biodiesels, the BTE amplified with load. At full load, hippe oil methyl ester B20 outperformed simarouba oil methyl ester B20 in terms of BTE [12]. Among the injection pressures examined, 230 bar provided the highest BTE, followed by 210 and 250 bar. Thermal efficiency decreased at 250 bar when compared to 230 bar [13]. Among the biodiesels tested, diesel had the highest efficiency, with 2.79%, 5.89%, 7.75%, 9.64%, and 14.03% higher than B10, B20, B30, B40, and B100 karanja biodiesel blends [17]. Ceiba pentandra oil methyl ester B20 demonstrated higher BTE than nigella sativa oil methyl ester B20 due to its higher calorific value and lower viscosity [25]. Because of their lower calorific value and higher viscosity, higher biodiesel blends have lower BTE than diesel [40-52].

* **Brake specific fuel consumption**

At all loads, the brake specific fuel consumption of rice bran oil methyl ester was higher than that of diesel. The brake specific fuel consumption at 100% load for diesel and rice bran oil methyl ester was 0.2752 kg/kW-hr and 0.3383 kg/kW-hr, respectively [3]. When using a 3-hole nozzle, the injection timing of 26°BTDC resulted in lower brake specific fuel consumption when compared to other injection timings such as 23 and 20°BTDC [4,11]. The minimum brake specific fuel consumption for diesel fuel at 3.52 kW of braking power was found to be 0.325 kg/kW-hr, 0.352 kg/kW-hr, and 0.367 kg/kW-hr at 26, 23, and 20°BTDC injection timings, respectively. The minimum brake specific fuel consumption for B20 gasoline at 3.52 kW of braking power was found to be 0.316 kg/kW-hr, 0.328 kg/kW-hr, and 0.355 kg/kW-hr for 26, 23, and 20°BTDC injection timings, respectively [4]. Brake specific gasoline usage fell as the load increased. When compared to diesel, biodiesel blends had higher brake specific fuel consumption [5-7]. Among the nozzles examined, the 5-hole nozzle had the lowest brake specific fuel usage. The brake specific gasoline consumption rose with a 4-hole nozzle [9]. The karanja B20 blend has a reduced brake specific fuel consumption of 0.341 kg/kW-hr when compared to other biodiesel blends [10]. At maximum load, the brake specific fuel consumption of hippe oil methyl ester B20 was smaller than that of simarouba oil methyl ester B20 and the same as that of diesel [12].

* **Exhaust gas temperature**

In all circumstances, the temperature of the exhaust gas increased as the brake power increased. At 80% load, the maximum exhaust gas temperature was 442°C with the B20 blend and 446°C with the jatropha oil methyl ester. With diesel, the maximum exhaust gas temperature is 390°C [7].

* **Volumetric efficiency**

At all load situations, hippe oil methyl ester B20 had a higher volumetric efficiency than simarouba oil methyl ester B20 [12].

* **In-cylinder pressure and peak pressure rise**

Diesel fuel had a higher peak pressure rise than B20 fuel at all loads [1, 15]. At 26°BTDC injection timing, the largest peak pressure rise for a 3-hole nozzle was 62.83 bar for diesel fuel and 60.64 bar for B20 fuel. At 26°BTDC injection time, the highest peak pressure rise for 4 hole nozzle was found to be 57.28 bar for diesel fuel and 57.11 bar for B20 fuel. At 26°BTDC injection time, the largest peak pressure rise for a 5-hole nozzle was determined to be 59.30 bar for diesel fuel and 57.44 bar for B20 fuel [2]. The peak pressure for diesel operation was higher than for rice bran oil methyl ester, with values at 100% load of 77 bar and 70 bar, respectively [3]. Among the various injection pressures studied, 230 bar injection pressure resulted in a larger peak pressure rise than 210 bar and 250 bar injection pressures. From the various injection timings investigated, 26°BTDC produced a larger peak pressure rise than 20 and 23°BTDC injection timings. The 5-hole injection nozzle outperformed the 3-hole and 4-hole injection nozzles in terms of peak pressure rise [14].

* **Indicated mean effective pressure**

At 26°BTDC injection time, the highest indicated mean effective pressure for diesel fuel was 16.31 bar and 16.06 bar for B20 fuel with a 3-hole nozzle. At 26°BTDC injection time, the highest indicated mean effective pressure for diesel fuel was 15.69 bar and 15.42 bar for B20 fuel for a 4-hole nozzle. At 26°BTDC injection time, the highest indicated mean effective pressure for a 5-hole nozzle was 15.16 bar for diesel fuel and 15.03 bar for B20 fuel [2].

* **Ignition delay period**

The reduced ignition delay may result in somewhat higher nitrogen oxide emissions when using a biodiesel blend [12]. The ignition delay and combustion duration of biodiesel fuels were found to be greater than diesel fuels at all injection timings. Increasing biodiesel injection timing reduces both ignition delay and combustion duration [15].

* **Combustion duration**

At 17°BTDC, the combustion duration and ignition delay of the engine fuelled with fish biodiesel and their blends show a decreasing trend at a fixed fuel injection pressure of 600 bar [15].

* **Smoke emissions**

Rice bran oil methyl ester produced more smoke than diesel. At 100% load, the smoke level was 72 HSU for diesel and 80 HSU for rice bran oil methyl ester [3]. For diesel, jatropha oil methyl ester, and their mixtures, the smoke opacity rose as the brake power increased. The smoke opacity rose as the proportion of jatropha oil methyl ester in the diesel blend increased. At 80%, the smoke level was 67 HSU (Hartridge Smoke Units) for the methyl ester, 63 HSU for the B20 blend, and 84 HSU for the B80 mix. At 80% load, the smoke level with diesel was 63 HSU [7]. Because the B0 blend mixes better with the air, it emits less smoke than the other blends tested. Among the several injection timings evaluated, 26°BTDC produces the least amount of smoke [11]. Among the various injection pressures evaluated, 230 bar produces less smoke than 210 and 250 bar [13,15]. The viscosity of the blends grew as the percentage of biodiesel in the blends increased, resulting in greater smoke. Among the biodiesels tested, diesel had the lowest smoke emissions, which were 6.06%, 12.9%, 20.68%, 29.62%, and 39.88% lower than B10, B20, B30, B40, and B100 fuel blends [17]. Due to better fuel characteristics, ceiba pentandra oil methyl ester B20 emits lower exhaust emissions than nigella sativa oil methyl ester B20 [25].

* **Hydrocarbon and carbon monoxide emissions**

Rice bran oil methyl ester produced more hydrocarbons and carbon monoxide than diesel. Diesel and rice bran oil methyl ester hydrocarbon emissions were 60 ppm and 71 ppm at full load, respectively. Carbon monoxide levels at full load for diesel and rice bran oil methyl ester were 0.2% and 0.51%, respectively [3]. The hydrocarbon emissions from B20, jatropha oil methyl ester, and diesel were found to be 66 ppm, 70 ppm, and 40.5 ppm, respectively. Carbon monoxide emissions from B20, jatropha oil methyl ester, and diesel were 0.1245%, 0.132%, and 0.1125%, respectively [7]. When compared to biodiesel blends, plain diesel emits less hydrocarbons and carbon monoxide [8]. Among the numerous nozzles evaluated, the 5-hole nozzle emits the least amount of hydrocarbons and carbon monoxide. The hydrocarbon and carbon monoxide emissions increased with the 4-hole nozzle [9]. When compared to other biodiesel blends, the karanja B20 mix emits 63 ppm less hydrocarbons. When compared to biodiesel blends, the karanja B20 blend emits 0.26% less carbon monoxide [10]. Among the different injection timings examined, 26°BTDC has the lowest hydrocarbon and carbon monoxide emissions because there was more time available for fuel and air mixing as the injection timing advanced from 20 to 26°BTDC [11]. Because nanoparticles supply oxygen for the oxidation of hydrocarbons and carbon monoxide during combustion, the addition of aluminum oxide nanoparticles reduced hydrocarbon emissions [12]. Among the various blends evaluated, the B100 blend emits more hydrocarbons than the other gasoline mixes [13,16]. Among the various injection pressures studied, 230 bar emits less hydrocarbon than 210 bar and 250 bar [13]. As engine load increased, so did hydrocarbon and carbon monoxide emissions. Hydrocarbon and carbon monoxide emissions from incomplete combustion are more visible in karanja biodiesel and its mixes B30 and B40 than in diesel. In comparison to the other B30 and B40 blends tested, B20 yielded acceptable levels of hydrocarbons and carbon monoxide [17]. Due to differences in fuel characteristics, ceiba pentandra oil methyl ester B20 emits less hydrocarbon and carbon monoxide than nigella sativa oil methyl ester B20 [25].

* **Nitric oxide emissions**

The nitric oxide emission values for rice bran oil methyl ester were 1147 ppm, compared to 1120 ppm for diesel operation at 100% load [3]. For all fuel combinations, nitric oxide emissions increased as load increased. Nitric oxide emissions were 1193 ppm, 1096 ppm, 921 ppm, 903 ppm, and 1100 ppm for B20, B40, B60, B80, and B100, respectively, compared to 900 ppm for diesel running at full load [7]. When compared to biodiesel mixes, plain diesel emits a higher level of nitric oxide [8,16]. Among the several nozzles studied, the 5-hole nozzle emits the most nitrogen oxide. Because the spray pattern of the 4-hole nozzle was uneven, the greatest amount of fuel impinged on the cylinder wall, nitrogen oxide emissions were reduced [9]. When compared to biodiesel blends, the karanja B20 blend emits more nitric oxide (1205 ppm) [10]. Among the several injection timings investigated, 26°BTDC emits the most nitric oxide [11]. Among the various injection pressures studied, 230 bar emits more nitric oxide than 210 and 250 bar [12]. B20 produced slightly higher nitric oxide emissions than the other B30 and B40 blends tested [17]. Due to differences in fuel characteristics, Ceiba pentandra oil methyl ester B20 produced more NOx than Nigella sativa oil methyl ester B20 [25].

* **Carbon di-oxide emissions**

At all loading conditions, all biodiesel blends emit less carbon dioxide than diesel. Because the oxygen concentration was lower at lower loads than when the engine was driven at higher loads, the carbon di-oxide emissions for all of the tested fuels increased as the brake power increased [17].

**2.2 Dual fuel combustion**

* **Brake thermal efficiency**

The rate of exhaust gas recirculation increased brake thermal efficiency [18]. The venture carburettor outperformed the simple carburettor in terms of brake thermal efficiency. At 80% load, the brake thermal efficiency values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with a 3 mm hole geometry carburettor were 26.16% and 22.88%, respectively [19]. The brake thermal efficiency increased for 80% and 100% loads as the injection timing was moved from 19 to 27°BTDC [20,21]. At 80% load, the brake thermal efficiency values for dual fuel operation with diesel-compressed biogas and rice bran oil methyl ester-compressed biogas were 23.48% and 21.29%, respectively [20]. The injection timing advancement of pilot fuel enhanced the brake thermal efficiency of a dual fuel engine when operated with biogas. In comparison to the biodiesel-biogas mode, the diesel-biogas mode demonstrated higher brake thermal efficiency [22]. More hydrogen gas flow rates reduce brake thermal efficiency. The brake thermal efficiency of a dual fuel engine powered by nigella sativa oil methyl ester B20 was greater than that of a jack fruit oil methyl ester B20 [23]. Increasing the injection timing for a dual fuel engine boosted engine performance, and thus brake thermal efficiency increased until 27°BTDC, after which it dropped. Due to differences in fuel characteristics, biogas, a typical diesel fuel powered dual fuel engine, demonstrated increased brake thermal efficiency, followed by B20 and B100 algal biodiesel blends [24]. When compared to dairy scum oil methyl ester-based dual fuel combustion, induction of producer gas and diesel-supported dual fuel combustion improves brake thermal efficiency by 12.5%. In the presence of hydrogen and 5% exhaust gas recirculation, the dairy scum oil methyl ester-based dual fuel combustion improved brake thermal efficiency by 6.1% over the dairy scum oil methyl ester-producer gas operation [26]. The diesel-compressed natural gas gaseous combination outperformed jamune biodiesel and its B20 blend in terms of brake thermal efficiency. When compared to previous ventures tried, the 9 mm size venture enhanced thermal efficiency for all fuel combinations [27]. The thermal efficiency of the brakes was decreased at medium loads and increased marginally at higher loads. At 80% load, a higher thermal efficiency of 31.25% was reached using 20% water in diesel emulsion and manifold injected ethanol [28]. Increasing the injection timing from 23° to 27°BTDC increased thermal efficiency for all fuel combinations tested. As the injection timing was advanced beyond 27 to 31°BTDC, the BTE dropped as more fuel burned in the diffusion combustion phase [29]. Because of the increased viscosity of mixed fuel and the reduced atomization of fuel droplets with nanoparticles, the dairy scum oil methyl ester + multi walled carbon nanotubes combination exhibits poor thermal efficiency for the same producer gas induction, and catalytic activity may be retarded [30]. At 0.25 kg/hr flow rate, more tested fuels are injected, improving the usage of gaseous fuel and thereby improving thermal efficiency. When compared to other evaluated fuel combinations, the diesel-biogas combination had the highest thermal efficiency [39].

* **Brake specific fuel consumption**

When carburettor-2 was employed, the brake specific fuel consumption value was lower during engine running. When the injection timing was increased from 19 to 27°BTDC, the brake specific fuel consumption value decreased [19-21].

* **Peak pressure rise**

Pressure rise was found to be greater in carburettor-2 for both diesel- and rice bran oil methyl ester-compressed biogas. At 80% load, the pressure rise values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas were 58.9 and 59.3 bar, respectively [19]. With advancing injection timing, the dual fuel engine’s improved thermal efficiency resulted in higher peak pressure. Dual fuel biodiesel-biogas engines have a longer delay period than single fuel engines [22]. Peak pressure in a dual fuel engine grew as hydrogen flow rate increased for all fuel combinations. B20 mixes with hydrogen have higher peak pressures than pure biodiesels. For all fuel combinations, high hydrogen flow rates exceeding 0.2 kg/h resulted in knocking behaviour and lower peak pressures [23]. Because of the enhanced injection timing, more pilot fuel is injected into the engine cylinder, increasing braking power. That pattern persisted until 27°BTDC, when peak pressure began to fall. Peak pressure is lower in biodiesel and biodiesel blends than in diesel operation [24]. Peak pressure grew as the number of holes on venture increased. The higher the thermal efficiency of the dual fuel engine, the higher the peak pressure with increasing holes on the gas mixing venture [27]. Dual fuel engine operating with B20 blends of jack fruit oil methyl ester B20 and nigella sativa oil methyl ester B20 with hydrogen induction produces higher peak pressure than pure biodiesel [29].

* **Ignition delay**

The ignition delay period grew as the injection timing was advanced because more fuel was injected into the engine cylinder. Biodiesel-biogas dual fuel engines have a longer delay period than single fuel engines. Biogas with the same Ceiba pentandra oil methyl ester B20 has a shorter ignition delay than Ceiba pentandra oil methyl ester B100 [22]. Longer delay periods were obtained for biogas induction and biodiesel blends injection when compared to diesel fuel operation. Increasing the injection time of the dual fuel engine to 27°BTDC resulted in shorter ignition delay periods due to improved fuel combustion. Because of its higher calorific value, algal oil methyl ester B20 demonstrated a shorter ignition delay with biogas induction than algae oil methyl ester B100 [24]. The igniting delay period reduced as the number of holes on the venture grew. Due to the higher methane concentration of the latter gas with higher in-cylinder pressures, biodiesel-biogas dual fuel engines have a longer delay period than biodiesel-compressed natural gas fuel operation. When compared to jamune oil methyl ester B100, gaseous fuels have a shorter igniting delay with jamune oil methyl ester B20 [27]. Dual fuel engine operating with B20 mixes of jack fruit oil methyl ester B20 and nigella sativa oil methyl ester B20 with hydrogen induction exhibited a longer delay duration than pure biodiesel. The increased cetane number and lower viscosity of B20 blends assures improved combustion and, as a result, a shorter delay period [29].

* **Hydrocarbon and carbon monoxide emissions**

With increasing load and exhaust gas recirculation rate, hydrocarbon and carbon monoxide emissions rose. Lower available oxygen for combustion resulted in a rich mixture, which resulted in incomplete combustion and greater hydrocarbon and carbon monoxide emissions [18]. The 3 mm carburettor produced the least amount of hydrocarbons and carbon monoxide. At 80% load, hydrocarbon levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with a 3 mm hole geometry carburettor were 63 and 64 ppm, respectively. Carbon monoxide levels were reported to be 0.15% and 0.165% at 80% load for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with 3 mm hole geometry carburettor, respectively, and 0.17% and 0.19% at 100% load [19]. The hydrocarbon emission for both loads reduced significantly as the injection timing increased [20,21]. At 80% load, hydrocarbon emission values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas dual fuel operation are 76, 68, and 63 ppm and 84, 74, and 71 ppm, respectively [20]. Carbon monoxide emission dropped significantly as injection timing was pushed from 19 to 27° BTDC [20,21]. Carbon monoxide emissions were 0.21, 0.14, and 0.12% for diesel-compressed biogas and 0.24, 0.21, and 0.18% for rice bran oil methyl ester-compressed biogas dual fuel operation at 19, 23, and 27°BTDC injection timing, respectively, at 80% load [20]. The lower velocity of the biogas flame in dual fuel mode contributed more to the formation of hydrocarbon emissions. The higher viscosity and reduced calorific value of raw biogas compared to purified biogas may be the cause of this behaviour. The reduction in hydrocarbon emissions was determined to be 15.24% when the injection timing was advanced from 19 to 31°BTDC [22]. Hydrocarbon and carbon monoxide emissions reduced when hydrogen gas flow rates increased because pilot injected fuels were discovered to be lower. Despite the prevalence of hydrogen fuel, dual-fuel engines powered by biodiesels produced higher hydrocarbon and carbon monoxide emissions than diesel engines. When compared to pure biodiesels, B20 blends of the respective biodiesels produced less hydrocarbon and carbon monoxide [23]. Injection timing advancement boosted the thermal efficiency of the dual fuel engine, resulting in lower hydrocarbon and carbon monoxide emissions. After 27°BTDC, engine performance degrades [24]. When compared to biodiesel-based dual fuel combustion, diesel-fueled dual fuel combustion reduced hydrocarbon and carbon monoxide emissions by 37.2% and 34.1%, respectively [26]. The use of 9 mm venture reduced hydrocarbon and carbon monoxide emissions. B100 and B20 blends with compressed natural gas had lower amounts of hydrocarbons and carbon monoxide than B100 and B20 blends with biogas [27]. When compared to baseline diesel operation, engine fuelled with an emulsion of 20% water in diesel with manifold injected ethanol reduced hydrocarbon and carbon monoxide emissions by 24.3% and 21%, respectively, at 80% load [28]. Dual fuel engine operating with B20 mixes of jack fruit and nigella sativa with hydrogen induction emits less hydrocarbon and carbon monoxide than pure biodiesels [29]. Although the producer gas induction was the same, the biodiesel-based dual fuel operation produced 32.6% and 29.8% more hydrocarbons and carbon monoxide at 80% load than the diesel operation without nanoparticles. Carbon monoxide levels were reduced by 20.6%, 14.8%, and 8.2%, respectively, at 80% load, compared to biodiesel-based dual fuel operation without nanoparticles [30].

* **Nitric oxide emissions**

Nitric oxide emissions reduced as the rate of exhaust gas recirculation was increased [18]. At 80% load, nitric oxide levels for diesel-compressed biogas and biodiesel-compressed biogas operation with a 3 mm hole geometry carburettor were 943 and 815 ppm, respectively [19]. Nitric oxide emission rose significantly when injection timing increased [20,21]. Nitric oxide emissions are observed to be reduced in dual fuel mode as compared to diesel fuel. On average, advancing injection timing from 19 to 31°BTDC resulted in a 40.02% increase in nitric oxide emissions [22]. When compared to B100 fuel operation, B20 blends produce significantly greater nitric oxide emissions. As the hydrogen flow rate increased, so did the combustion activity, resulting in higher in-cylinder pressures and temperatures within the combustion chamber of dual fuel engine operation, and hence higher nitric oxide emissions [23]. Adding nanoparticles to algae biodiesel increases dual fuel engine performance significantly by increasing in-cylinder pressures and temperatures, resulting in higher nitric oxide outputs when compared to algae biodiesel and its blend [24]. Diesel-based dual fuel combustion mode raised nitric oxide levels by 26.4% when compared to dairy scum oil methyl ester-based dual fuel operation for same producer gas induction. Dairy scum oil methyl ester-producer gas with hydrogen and 5% exhaust gas recirculation had 12.4% higher nitric oxide levels than without exhaust gas recirculation. When compared to combustion with a 5% exhaust gas recirculation rate, higher exhaust gas recirculation rates reduce nitric oxide emissions by 32.2% [26]. For all dual fuel operations, the 9 mm venturi produced more nitric oxide than the 2 mm and 6 mm venturies. Gaseous fuels, such as jamune oil methyl ester B20, have increased nitric oxide levels as a result of blending with diesel, which enhances fuel properties while also increasing engine temperature and heat release rate [27]. When compared to baseline diesel operation, engine fuelled with an emulsion of 20% water in diesel with manifold injected ethanol reduced nitric oxide emissions by 10.8% at 80% load [28]. The nitric oxide emissions from biodiesel operation were found to be lower than those from diesel mode. As more fuel was delivered into the engine cylinder with a longer ignition delay, increasing the injection time from 23 to 27°BTDC resulted in increased nitric oxide. Nitric oxide emissions dropped as the injection time was advanced beyond 27°BTDC, i.e., at 31°BTDC, as less fuel participated in the regulated combustion phase with lower thermal efficiency [29]. When comparing dairy scum oil methyl ester-based dual fuel operation to fossil fuel-based dual fuel operation at 80% load, trial results revealed that the dairy scum oil methyl ester-based dual fuel operation lowered nitric oxide levels by 12.8% [30].

* **Smoke emissions**

As the rate of exhaust gas recirculation rose, so did the smoke [18]. The smoke opacity was reduced by using a mixing chamber venture with 3 mm hole geometry. At 80% load, smoke values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operations with a 3 mm hole geometry carburettor were 61 and 68 HSU, respectively [19]. The opacity of the smoke reduced as the injection timing was increased. Smoke emissions increased marginally as engine load increased due to a decrease in air volumetric efficiency in dual fuel mode [20,21]. At 80% load, smoke levels were determined to be 66, 61, and 64 HSU for diesel-compressed biogas and 74, 68, and 71 HSU for rice bran oil methyl ester-compressed biogas dual fuel operation at 19, 23, and 27°BTDC injection timing [20]. Ceiba pentandra oil methyl ester B100 and B20 with purified biogas in dual fuel mode operation produced less smoke than Ceiba pentandra oil methyl ester B100 and B20 with raw biogas operation. The B20 blend produced less smoke than the B100 blend due to the blending of biodiesel with diesel, which affected and enhanced the fuel property of the B20 blend [22]. With greater hydrogen gas flow rates, there is less smoke emissions. When compared to B20 fuels, B100 mixes produced more smoke for pilot fuels. Because of fuel attribute variation, Nigella sativa oil methyl ester B20 had higher smoke opacity than B20 blend [23]. As the performance of the dual fuel engine improved, smoke emissions decreased until they reached 27°BTDC, after which they increased again. Because biodiesel and its blends are somewhat viscous, they emit more smoke. In dual fuel mode, algae oil methyl ester B20 emits less smoke than B100 blend with biogas induction [24]. Diesel-based dual fuel operation with producer gas emits 27.4% less smoke than dairy scum oil methyl ester-based dual fuel combustion. Dairy scum oil methyl ester-producer gas with hydrogen and 5% exhaust gas recirculation reduced smoke levels by 10.2% when compared to the same fuel combination without exhaust gas recirculation [26]. The 9 mm venture had lower smoke opacity than the other ventures evaluated. Smoke levels were found to be higher in jamune oil methyl ester and biogas injected operation than in compressed natural gas powered dual fuel mode operation [27]. Lower smoke emissions of 41 HSU were recorded when the engine was fuelled with a 20% water in diesel emulsion with manifold injected ethanol at 80% load. When ethanol injection was used, smoke emissions were reduced by 27.9% compared to baseline diesel operation [28]. Because nigella sativa oil methyl ester B20 has a lower viscosity and a higher calorific value than jack fruit oil methyl ester B20, it has lower smoke opacity [29]. The increased dosage of nanoparticles in the fuel was limited, resulting in lower smoke levels. As a result, the obtained smoke levels were proportional to the nanoparticle concentration [30].

**2.3 RCCI combustion**

* **Brake thermal efficiency**

With an increase in the amount of gasoline fuel, thermal efficiency decreased. Because of the decrease in viscosity, the increased calorific value of diesel resulted in correct mixing of the air-fuel combination, and thus there was more thermal efficiency for diesel as compared to biodiesel [31]. The engine's braking thermal efficiency improved for RCCI combustion mode against homogeneous charge compression ignition (HCCI) mode up to 40% of gaseous fuels energy share, but decreased with further increase in gaseous fuels energy share [32]. With increasing pentanol content, brake thermal efficiency declined. At 10% pentanol in injected fuels, diesel and pentanol fuel combination had a better thermal efficiency of 22.15%, which was roughly 9.1% and 27.3% higher than other fuel combinations [33]. The 75% load variation has higher thermal efficiency than the 50% load variation. At 75% load, a higher thermal efficiency of approximately 29.74% was attained for the diesel-compressed natural gas fuel combination [34]. The thermal efficiency amplified as the injection timing increased from 45 to 50°ATDC. The thermal efficiency of the brake reduced as the injection timing increased from 50 to 55°ATDC [35]. The engine’s brake thermal efficiency increased as the gaseous fuel energy contribution grew up to 40%, but dropped over 40% [36,38]. When compressed natural gas was used as a low reactive fuel, the engine's braking thermal efficiency improved [36]. The thermal efficiency of the brakes dropped as the proportion of n-butanol increased. The highest thermal efficiency was obtained for a diesel and n-butanol fuel mixture containing 10% n-butanol [37].

* **Specific fuel consumption**

Specific fuel consumption increased as the percentage of gasoline fuel increased. When compared to diesel, biodiesel operation consumed more gasoline. Due to the prevalence of gasoline, the qualities of diesel, biodiesel, and their blends were accountable for the trends. Biodiesel's lower calorific value and increased viscosity resulted in poor air-fuel mixing inside the engine cylinder. As a result, biodiesel operations consumed more gasoline than diesel operations [31].

* **Nitric oxide emissions**

The concentration of nitric oxide decreased as the fraction of gasoline increased. Less nitric oxide was discovered in biodiesel due to decreased in-cylinder pressure and temperature, as well as prolonged residual time; higher viscosity resulted in incomplete combustion, which lowers in-cylinder pressure and temperature [31]. Nitric oxide emissions decreased as the energy percentage of gaseous fuels increased. When compared to other fuel combination types, diesel and hydrogen produce more nitric oxide [32]. Nitric oxide emissions reduced as pentanol fraction increased. At 10% pentanol, the diesel and pentanol fuel combination produced the highest nitric oxide emissions [33]. The 75% load variation produces more nitric oxide than the 50% load variation. The diesel-compressed natural gas fuel mix produced higher nitric oxide emissions [34]. Among the various injection timings studied, 50°ATDC emits more nitric oxide than 45 and 55°ATDC [35]. As the amount of energy from gaseous fuels increased, so did nitric oxide emissions. As the energy share increases, the fuel and mixture become lean, resulting in low-temperature combustion. The low temperature combustion technique reduced nitric oxide emissions [36]. Nitric oxide emissions dropped as the proportion of n-butanol increased. The highest nitric oxide emissions were obtained for a diesel and n-butanol fuel mixture containing 10% n-butanol [37]. When compared to other fuel combinations, RCCI engines operated by diesel and producing gas produced higher nitric oxide emissions [38].

* **Smoke**

As the energy percentage of gaseous fuels increased, so did smoke emissions. When compared to other fuel combinations, the diesel and hydrogen fuel combination produces less smoke [32]. Smoke emissions reduced as the percentage of pentanol increased. Diesel and n-pentanol had the lowest smoke emissions when 10% pentanol was used in injected fuels [33]. When compared to the 50% load variation, the 75% load variation emits less smoke. Among the various fuel combinations, diesel and compressed natural gas emit less smoke than other fuel combinations [34]. Smoke emissions are quite minimal while using the RCCI combustion mode. Injection timing of 50°ATDC produced less smoke than injection timing of 45 and 55°ATDC. Thevetia peruviana methyl ester-powered RCCI combustion produced more smoke than diesel. Poor combustion characteristics of biodiesel-injected RCCI combustion produced a substantial amount of smoke [35]. Smoke emissions dropped as the proportion of n-butanol increased. At 10% n-butanol in injected fuels, the lowest smoke emissions were recorded for diesel and n-butanol. Smoke emissions were extremely low due to substantially premixed n-butanol and a longer mixing period for diesel [37]. When compared to other fuel combinations, the RCCI engine driven by diesel and producing gas produced less smoke. Biodiesel activity produced more smoke [38].

* **Hydrocarbon and carbon monoxide**

The increase in gasoline content increased hydrocarbon and carbon monoxide emissions. Biodiesel operation resulted in higher hydrocarbon and carbon monoxide emissions. One of the reasons for greater hydrocarbon and carbon monoxide emissions could be biodiesel's higher specific fuel consumption compared to diesel [31]. As the energy share increased, so did the emissions of hydrocarbons and carbon monoxide. When compared to other fuel combination types, diesel and hydrogen produce less hydrocarbon and carbon monoxide emissions [32]. The percentage of pentanol used increased hydrocarbon and carbon monoxide emissions. At 10% pentanol in injected fuels, diesel and pentanol produced the lowest hydrocarbon and carbon monoxide emissions [33]. When compared to the 50% load variation, the 75% load variation emits less hydrocarbon and carbon monoxide. Diesel and compressed natural gas produced the least amount of hydrocarbon and carbon monoxide emissions of the fuel combinations examined [34]. Among the numerous injection timings investigated, 50°ATDC emits the fewest hydrocarbons and carbon monoxide. Biodiesel-powered RCCI combustion emits a greater number of hydrocarbons and carbon monoxide [35]. The amount of energy from gaseous fuels increased the amount of hydrocarbon and carbon monoxide engine-out emissions. Compressed biogas had higher hydrocarbon and carbon monoxide engine-out emissions than compressed natural gas as a low reactive fuel [36]. The proportion of n-butanol enhanced the hydrocarbon and carbon monoxide emissions. At 10% n-butanol in injected fuels, the lowest hydrocarbon and carbon monoxide emissions were recorded for diesel and n-butanol [37]. When compared to other fuel combinations, diesel and producer gas powered RCCI combustion modes produced lower hydrocarbon and carbon monoxide emissions [38].

* **Carbon di-oxide emissions**

The increase in gasoline content reduced carbon dioxide emissions. Diesel gasoline had higher emissions because it burned more efficiently than B20 and B100 mixes [31].

* **In-cylinder pressure and peak pressure rise**

When compared to other types of combustion modes, the RCCI mode produced higher in-cylinder pressure. In both combustion modes, the pressure rise for diesel and hydrogen fuel combinations was greater than for other fuel combinations [32]. When compared to other evaluated fuels, the maximum pressure was reported for the diesel and pentanol fuel combination mode [33]. Diesel and compressed natural gas fuel combinations produced increased in-cylinder pressure at all injection timings. When compared to diesel, the engine running on biodiesel demonstrated a slower rate of in-cylinder pressure rise [35]. When compared to other fuel combinations, the diesel and compressed natural gas fuel combination had a higher peak pressure rise. Peak pressure rise was lower in biodiesel-powered RCCI engines than in diesel-powered engines [36]. When compared to other evaluated fuels, the diesel and production gas fuel combination mode produced the highest pressure [38].

* **Heat release rate**

When compared to other combustion modes, the RCCI mode produced a higher heat release rate. In both combustion modes, the heat release rate for diesel and hydrogen fuel combinations was higher than for other fuel combinations [32]. When compared to other evaluated fuels, the diesel and pentanol fuel combination mode produced the highest heat release rate [33]. Diesel with compressed natural gas had a higher heat release rate as compared to other fuel combinations. When compared to compressed natural gas, the engine powered by compressed biogas produced a lower heat release rate [35]. When compared to other fuel combinations, the diesel and compressed natural gas fuel combination produced the highest heat release rate. The heat release rate of a biodiesel-powered RCCI engine was lower than that of a diesel-powered engine [36].

**3. Conclusion**

According to the extensive literature review, RCCI combustion is an effective ignition control that maintains high efficiency and low emissions. RCCI combustion is more promising than other low temperature combustion modes. The ignition timing in RCCI combustion can be adjusted by altering the ratio of high cetane fuel to high octane fuel. With the addition of high reactive fuel to RCCI combustion, nitric oxide emissions were reduced. The hydrocarbon and carbon monoxide emissions from RCCI combustion were somewhat increased. Biodiesel can be used to partially replace fossil diesel, reducing the need for petroleum fuel and providing an environmentally friendly energy source.

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