**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 investigated alternative fuels in the dual fuel mode to improve the emission profiles and energy consumption for transportation and distribution activities. In this review, the engine performance, combustion, and emission characteristics of alternative fuels in conventional, dual fuel and reactivity controlled compression ignition (RCCI) mode combustion are thoroughly analyzed. Due to different fuel properties, the size distribution of spray droplets, and consequent mixing with ambient air, generally the RCCI engine was favourable in terms of emission characteristics. The RCCI technique is capable of controlling the combustion phase, peak pressure rise and heat release rate (HRR) through regulating the reactivity stratification to attain the resolution of combustion process optimization.

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

**1. Introduction**

Population growth is driving up energy demands in the transportation sector and at the same time, economic policies are seeking to boost efficiency and reduce on dangerous pollutant emissions like 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]. More recently, dual fuel injection methods have been used to promote the utilization of less reactive fuels and facilitate more advanced combustion strategies. Some dual fuel combustion modes have shown significant promise and operate with high efficiency and low pollutant output. This is often achieved over a wide operating range by simultaneously utilizing two fuels with differing concentrations to promote premixing of the fuel or create 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 mode of combustion 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 mode of combustion**

**2.1 Conventional mode of combustion**

* **Brake thermal efficiency**

The brake thermal efficiency for diesel was higher than rice bran oil methyl ester, because of high viscosity and density of vegetable oil that leads to low volatility which resulted in poor mixture formation. The maximum values were found to be 31.09% and 28.27% for diesel and rice bran oil methyl ester respectively at 80% load [3]. Brake thermal efficiency increased as the brake power increased. Brake thermal efficiency of B20 fuel was better than diesel fuel due to more oxygen content in the B20 blend. For 3-hole nozzle 26°BTDC injection timing gave higher thermal efficiency as compared to other injection timings such as 23°BTDC and 20°BTDC injection timing. The increase in thermal efficiency at advanced injection timing may be attributed to improved combustion due to sufficient time availability for evaporation and better mixing of fuel and air, which resulted in higher premixed combustion. Retarded injection timing resulted in lower thermal efficiency due to less time for evaporation and better mixing of fuel and air resulting late combustion in operating cycle [4]. For all the fuels tested the brake thermal efficiency increased with increase in load. The brake thermal efficiency of biodiesel blends was found to be lower as compared to diesel at all power output. The lower calorific value, higher viscosity, higher density which leads to poor atomization of biodiesel than diesel which resulted into increase of brake thermal efficiency for diesel than biodiesel blends. At 80% load condition all tested fuels gave higher brake thermal efficiency than at 100% load condition. The power produced from the engine was less than the amount of fuel consumed to develop that power at 100% load condition so that brake thermal efficiency decreased at 100% load condition as compared to 80% load condition [5-8, 16]. Among all the blends tested B0 blend gave higher brake thermal efficiency because of higher calorific value of the B0 blend. Among different nozzles tested, 5-hole nozzle gave the higher brake thermal efficiency because diameter of nozzle holes decreased so that area of nozzle holes also decreased which resulted into lower fuel consumption and higher thermal efficiency. For 4-hole nozzle, the brake thermal efficiency decreased because the spray pattern of 4-hole nozzle was irregular so that maximum amount of fuel was impinged on the cylinder wall and hence brake thermal efficiency decreased [9]. The brake thermal efficiency of karanja B20 blend was 25.52%, which was higher as compared to other biodiesel blends [10]. Among all the blends tested B0 blend gave higher brake thermal efficiency because of higher calorific value of the B0 blend. Among different injection timings tested 26°BTDC provided higher brake thermal efficiency because with the advancing in injection timing from 20°BTDC to 26°BTDC there was much time availability for the mixing of fuel and air and hence brake thermal efficiency increased [11]. The brake thermal efficiency increased with the load for diesel, simarouba oil methyl ester and hippe oil methyl ester biodiesel. The brake thermal efficiency of hippe oil methyl ester B20 was better than that of simarouba oil methyl ester B20 at full load [12]. Among various injection pressures tested 230 bar gave higher thermal efficiency as compared with 210 and 250 bar. As the injection pressure increased from 210 to 230 bar, fine atomization of the fuel particles takes place and these fuel particles well mixed with air inside the cylinder and hence thermal efficiency increased at 230 bar. At 250 bar, there was decrease of thermal efficiency as compared with 230 bar. As the injection pressure increased from 230 to 250 bar there was much more fine atomization of fuel particles takes place but the velocity with which these fuel particles are accumulated inside the combustion chamber is more and hence for 250 bar thermal efficiency decreased [13]. As the injection timing was advanced or retarded the thermal efficiency of the engine showed varied behaviour when powered with different fuel combinations. Biodiesel and their blends with diesel showed lower efficiency as compared to diesel due to their higher viscosity and lower energy content. Diesel showed higher efficiency when injected at 10°BTDC while the biodiesels and their blends showed improved performance when injected with advancing the injection timing of 17°BTDC [15]. The rise in brake power was accompanied by a steady increase in thermal efficiency. Lower blend ratios of karanja biodiesel, such as B10 and B20 mix operation, revealed tendencies similar to diesel operation in terms of brake thermal efficiency. When the quantity of biodiesel in the blend was increased, the efficiency decreased. Among the biodiesels tested, the maximum efficiency was recorded with diesel which was 2.79%, 5.89%, 7.75%, 9.64% and 14.03% greater than B10, B20, B30, B40 and B100 fuel blends [17]. More brake thermal efficiency was observed for ceiba pentandra oil methyl ester B20 than nigella sativa oil methyl ester B20 because of its higher calorific value and lower viscosity. For a fixed number of cavities on a piston as number of cavities and domes were increased from 2C-1D to 3C-2D, higher thermal efficiency was found. Increase in swirl and improved homogeneity of mixture formation. Further increase in domes and cavities from 3C-2D to 4C-3D, showed drop in thermal efficiency due to knocking tendency [25]. Brake thermal efficiency of higher biodiesel blends was lower than diesel due to lower calorific value and higher viscosity [40-52].

* **Brake specific fuel consumption**

The brake specific fuel consumption for rice bran oil methyl ester was more comparing to diesel at all loads. Low in energy value of rice bran oil methyl ester which required more fuel to produce same power at same load compared to diesel fuel used. The values of brake specific fuel consumption at 100% load were 0.2752 kg/kW-hr and 0.3383 kg/kW-hr for diesel and rice bran oil methyl ester respectively [3]. Brake specific fuel consumption decreased as the brake power increased. Brake specific fuel consumption for B20 blend was slightly lower than diesel because of extra oxygen present in the blend was taking part in combustion process. For 3-hole nozzle 26°BTDC injection timing gave lower brake specific fuel consumption as compared to other injection timings such as 23 and 20°BTDC. For diesel fuel at 3.52 kW of brake power the minimum brake specific fuel consumption was found to be 0.325 kg/kW-hr, 0.352 kg/kW-hr, 0.367 kg/kW-hr at 26, 23 and 20°BTDC injection timings respectively. For B20 fuel at 3.52 kW of brake power, the minimum brake specific fuel consumption was found to be 0.316 kg/kW-hr, 0.328 kg/kW-hr, 0.355 kg/kW-hr at 26, 23 and 20°BTDC injection timings respectively [4]. As the load increased brake specific fuel consumption decreased. Brake specific fuel consumption for blends of biodiesel blends was higher when compared with diesel. For effective burning of the fuel the calorific value of the fuel should be higher so that the evaporation of the fuel was also high. The calorific values of blends of biodiesel were lower when compared with diesel and hence the fuel evaporation was slower. Slower evaporation rates leads to higher brake specific fuel consumption [5-7]. Among all the blends tested B0 blend gave lower brake specific fuel consumption because of higher calorific value of the B0 blend. Among different nozzles tested 5-hole nozzle gave lower brake specific fuel consumption because diameter of nozzle holes decreased so that area of nozzle holes also decreased. That resulted into lower fuel consumption and higher thermal efficiency. For 4-hole nozzle the brake specific fuel consumption increased because the spray pattern of 4-hole nozzle was irregular so that maximum amount of fuel was impinged on the cylinder wall and hence brake specific fuel consumption increased [9]. Brake specific fuel consumption for karanja B20 blend was 0.341 kg/kW-hr which was lower as compared to other biodiesel blends [10]. Among all the blends tested, B0 blend gave lower brake specific fuel consumption because of lower viscosity of the B0 blend. Among different injection timings tested 26°BTDC gave lower brake specific fuel consumption because with the advancing in injection timing from 20°BTDC to 26°BTDC there was much time availability for the mixing of fuel and air and hence brake specific fuel consumption decreased [11]. The brake specific fuel consumption for hippe oil methyl ester B20 was less than simarouba oil methyl ester B20 and same as that of diesel at full load condition [12].

* **Total fuel consumption**

As the load increased, total fuel consumption increased for all fuels tested. Total fuel consumption for diesel was less as compared to biodiesel blends. Higher viscosity, higher density which leads to higher fuel consumption of biodiesel than diesel [5-7]. Among all the blends tested B0 blend gave lower total fuel consumption because of higher calorific value of the B0 blend. Among different nozzles tested 5-hole nozzle gave lower total fuel consumption because diameter of nozzle holes decreased so that area of nozzle holes also decreased. That resulted into lower fuel consumption and higher thermal efficiency. For 4-hole nozzle the total fuel consumption increased because the spray pattern of 4-hole nozzle was irregular so that maximum amount of fuel was impinged on the cylinder wall and hence total fuel consumption increased [9]. Total fuel consumption for karanja B20 blend was 1.86 kg/hr which was lower as compared to other biodiesel blends [10]. Among all the blends tested B0 blend gave lower total fuel consumption because of lower viscosity of the B0 blend. Among different injection timings tested 26°BTDC gave lower total fuel consumption because with the advancing in injection timing from 20 to 26°BTDC there was much time availability for the mixing of fuel and air and hence total fuel consumption decreased [11].

* **Exhaust gas temperature**

The exhaust gas temperature increased with increase in brake power in all the cases. The decrease in exhaust gas temperature was only up to B20 biodiesel blend and further increasing the blend ratio, temperatures were found to increase slightly. The maximum exhaust gas temperature at 80% load was 442°C with B20 blend and 446°C with the jatropha oil methyl ester. The peak exhaust gas temperature is 390°C with diesel [7].

* **Volumetric efficiency**

The volumetric efficiency for hippe oil methyl ester B20 was more than simarouba oil methyl ester B20 at all load conditions [12].

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

At all loads, the peak pressure rise of diesel fuel was more than B20 fuel [1, 15]. For 3-hole nozzle, highest peak pressure rise was found to be 62.83 bar for diesel fuel, 60.64 bar for B20 fuel at 26°BTDC injection timing. For 4 hole nozzle, highest peak pressure rise was found to be 57.28 bar for diesel fuel, 57.11 bar for B20 fuel at 26°BTDC injection timing. For 5-hole nozzle, highest peak pressure rise was found to be 59.30 bar for diesel fuel, 57.44 bar for B20 fuel at 26°BTDC injection timing [2]. The peak pressure for diesel operation was more compared to rice bran oil methyl ester and the values were 77 bar and 70 bar at 100% load respectively [3]. Among various injection pressures studied 230 bar injection pressure gave higher peak pressure rise as compared with 210 bar and 250 bar injection pressures. From various injection timings studied 26°BTDC gave higher peak pressure rise as compared with 20 and 23°BTDC injection timings. Among various injection nozzles studied 5-hole nozzle gave higher peak pressure rise as compared with 3-hole and 4-hole injection nozzles [14].

* **Indicated mean effective pressure**

For 3-hole nozzle, highest indicated mean effective pressure was found to be 15.63 bar for diesel fuel at 3.52 kW of brake power and 230 bar injection pressure, 15.73 bar for B20 fuel at 3.52 kW of brake power and 250 bar injection pressure. For 4-hole nozzle, highest indicated mean effective pressure was found to be 15.54 bar for diesel fuel at 3.52 kW of brake power and 250 bar injection pressure, 15.42 bar for B20 fuel at 3.52 kW of brake power and 230 bar injection pressure. For 5-hole nozzle, highest indicated mean effective pressure was found to be 15.46 bar for diesel fuel at 3.52 kW of brake power and 250 bar injection pressure, 15.79 bar for B20 fuel at 3.52 kW of brake power and 250 bar injection pressure [1]. For 3-hole nozzle, highest indicated mean effective pressure was found to be 16.31 bar for diesel fuel, 16.06 bar for B20 fuel at 26°BTDC injection timing. For 4-hole nozzle, highest indicated mean effective pressure was found to be 15.69 bar for diesel fuel, 15.42 bar for B20 fuel at 26°BTDC injection timing. For 5-hole nozzle, highest indicated mean effective pressure was found to be 15.16 bar for diesel fuel, 15.03 bar for B20 fuel at 26°BTDC injection timing [2].

* **Ignition delay period**

The shorter ignition delay may give to marginally increased oxides of nitrogen emissions with biodiesel blend [12]. At all injection timings, the ignition delay and combustion duration of the biodiesel fuels were found to be higher than the diesel. Advancing the injection timing for biodiesel fuels lower both ignition delay and combustion duration [15].

* **Combustion duration**

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

* **Smoke emissions**

The smoke emission for rice bran oil methyl ester was higher than diesel. Smoke level at 100% load was 72 HSU for diesel and 80 HSU for rice bran oil methyl ester [3]. The smoke opacity increased with increase in brake power for diesel, jatropha oil methyl ester and its blends. As the percentage of jatropha oil methyl ester increased in the blend with diesel the smoke opacity increased. Smoke level at 80% was 67 HSU (Hartridge Smoke Units) for the methyl ester and 63 HSU for the B20 blend and 84 for B80 blend. The smoke level with diesel was 63 HSU at 80% load [7]. Among all the blends tested B0 blend provide lower smoke emissions because of better mixing of the B0 blend with the air. Among different injection timings tested 26°BTDC exhibit the lower smoke emissions [11]. Among various injection pressure tested 230 bar provide lower smoke emissions as compared with 210 and 250 bar [13]. Diesel showed lower smoke when injected at 10°BTDC while the biodiesels fuelled engine showed lower smoke with advancing the injection timing of 17°BTDC. Pyrogallol biodiesel enhanced the oxidation stability of biodiesel and hence smoke emissions were improved with the addition of antioxidant in the fuel [15]. As the percentage of biodiesel in the blends increased, the viscosity of the blends also increased, resulted in more smoke. Among the biodiesels tested, the minimum smoke emissions were recorded with diesel which were 6.06%, 12.9%, 20.68%, 29.62% and 39.88% lower than B10, B20, B30, B40 and B100 fuel blends [17]. Lower exhaust emissions are observed for ceiba pentandra oil methyl ester B20 than nigella sativa oil methyl ester B20 due to improved fuel properties. For a fixed number of cavities on piston as the number of cavities and domes were increased from 2C-1D to 3C-2D both B20 fuels show decrease in smoke emissions caused by improved swirling rate and uniform air–fuel mixture and improved oxidation rate [25].

* **Hydrocarbon and carbon monoxide emissions**

The hydrocarbon and carbon monoxide emissions for rice bran oil methyl ester was higher than the diesel. At full load the values of hydrocarbon emissions for diesel and rice bran oil methyl ester were 60 ppm and 71 ppm respectively. At full load the values of carbon monoxide for diesel and rice bran oil methyl ester were 0.2% and 0.51% respectively at 100% load [3]. The hydrocarbon emissions were found to be 66 ppm, 70 ppm and 40.5 ppm for B20, jatropha oil methyl ester and diesel respectively. Carbon monoxide emissions were 0.1245%, 0.132% and 0.1125% for B20, jatropha oil methyl ester and diesel respectively [7]. The neat diesel exhibit lower amount of hydrocarbon and carbon monoxide emissions as compared to biodiesel blends [8]. Among different nozzles tested 5-hole nozzle exhibit the lower hydrocarbon and carbon monoxide emissions. For 4-hole nozzle the hydrocarbon and carbon monoxide emissions increase because the spray pattern of 4 hole nozzle was irregular so that maximum amount of fuel was impinged on the cylinder wall [9]. The karanja B20 blend exhibit lower amount of hydrocarbon emissions 63 ppm as compared to other biodiesel blends. The karanja B20 blend exhibit lower amount of carbon monoxide emissions 0.26% as compared to biodiesel blends [10]. Among different injection timings tested 26°BTDC exhibit the lower hydrocarbon and carbon monoxide emissions because with the advancing in injection timing from 20 to 26°BTDC there was much time availability for the mixing of fuel and air [11]. Extra adding of aluminium oxide nanoparticles reduced the hydrocarbon emissions, since nanoparticles supplies the oxygen for the oxidation of hydrocarbon and carbon monoxide during combustion [12]. Among various blends tested B100 blend exhibit higher hydrocarbon emissions as compared with other fuel blends [13, 16]. Among various injection pressure tested 230 bar exhibit lower hydrocarbon emissions as compared with 210 bar and 250 bar [13]. As engine load increased, hydrocarbon and carbon monoxide emissions were also increased. Incomplete combustion causes hydrocarbon and carbon monoxide emissions, which are more noticeable in karanja biodiesel and its blends B30 and B40 than in diesel. When compared to other B30 and B40 blends evaluated, B20 produced acceptable hydrocarbon and carbon monoxide levels [17]. For a fixed number of cavities on the piston as number of cavities and domes of cylinder head increased from 2C-1D to 3C-2D there was a decrease in hydrocarbon and carbon monoxide emission levels found. Positive trends were observed when number of cavities and domes were increased from 2C-1D to 3C-2D beyond which no improvement was found. Lower hydrocarbon and carbon monoxide emissions were found for ceiba pentandra oil methyl ester B20 than nigella sativa oil methyl ester B20 due to differences in their fuel properties [25].

* **Nitric oxide emissions**

The nitric oxide emission values were 1147 ppm for rice bran oil methyl ester as compared to 1120 ppm with diesel operation at 100% load [3]. The nitric oxide emissions increased with increase in load for all the fuel combinations. The nitric oxide emission values for B20, B40, B60, B80 and B100 were 1193 ppm, 1096 ppm, 921 ppm, 903 ppm and 1100 ppm respectively as compared to 900 ppm for diesel operation at full load operation [7]. The neat diesel exhibit higher amount of nitric oxide emissions as compared to biodiesel blends [8,16]. Among different nozzles tested 5-hole nozzle exhibit the higher nitrogen oxide emissions. For 4-hole nozzle the nitrogen oxide emission decreased because the spray pattern of 4-hole nozzle was irregular so that maximum amount of fuel was impinged on the cylinder wall [9]. The karanja B20 blend exhibit higher amount of nitric oxide emissions 1205 ppm as compared to biodiesel blends [10]. Among different injection timings tested 26°BTDC exhibit the higher nitric oxide emissions [11]. Among various injection pressures tested 230 bar exhibit higher nitric oxide emissions as compared with 210 and 250 bar [12]. When compared to the other B30 and B40 blends tested, B20 produced somewhat greater nitric oxide emissions [17]. Ceiba pentandra oil methyl ester B20 showed higher NOx emissions than nigella sativa oil methyl ester B20 due to differences in the fuel properties. For a fixed number of cavities of piston as the number of cavities and domes were increased from 2C-1D to 3C-2D high turbulence resulted into uniform air-fuel mixture which increased the engine temperature there by nitric oxide levels were increased for both tested fuels [25].

* **Carbon di-oxide emissions**

All biodiesel blends emit less carbon di-oxide than diesel at all loading situations. The carbon di-oxide emissions for all of the tested fuels increase when the brake power increased, because the oxygen concentration was lower at lower loads than when the engine was operated at greater loads [17].

**2.2 Dual fuel combustion**

* **Brake thermal efficiency**

Brake thermal efficiency increased with increase in exhaust gas recirculation rate. With exhaust gas recirculation was due to re-burning of hydrocarbon that entered combustion chamber with the recirculation of exhaust gases and also exhaust gas recirculation increased intake charge temperature which increased the rate of combustion [18]. Brake thermal efficiency was found to be higher for venture carburettor compared to simple carburettor. Brake thermal efficiency values for diesel-compressed biogas and rice bran oil methyl ester- compressed biogas operation with 3 mm hole geometry carburettor were found to be 26.16% and 22.88% respectively at 80% load [19]. As the injection timing was advanced from 19 to 27°BTDC, the brake thermal efficiency increased for 80% and 100% loads. More time would be available for CBG fuel burning and resulted in better performance with improved brake thermal efficiency [20,21]. Brake thermal efficiency values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas dual fuel operation at 27°BTDC injection timing were found to be 23.48% and 21.29 % respectively at 80% load [20]. Brake thermal efficiency of dual fuel engine when operated with biogas improved with the injection timing advancement of pilot fuel. When compared to biodiesel-biogas mode the diesel-biogas mode showed more brake thermal efficiency [22]. Brake thermal efficiency reduced with more hydrogen gas flow rates. Incomplete combustion of injected pilot fuels of diesel, biodiesel and their B20 blends associated with reduction in air entrapment and decreased ignition source. At lower hydrogen gas flow rates more pilot fuel injection occurs hence improving the gaseous fuel utilization. Brake thermal efficiency of dual fuel engine fuelled with nigella sativa oil methyl ester B20 was higher compared with jack fruit oil methyl ester B20 because of variations in properties fuels and improved catalytic combustion of the former biodiesel blends [23]. Advancing the injection timing for dual fuel engine improved the engine performance and accordingly the brake thermal efficiency increased till 27°BTDC beyond which it decreased. At advanced injection timing delay period increased and more pilot fuel was injected inside the engine cylinder. Biogas being common diesel fuel operated dual fuel engine showed improved brake thermal efficiency followed by B20 and B100 algae biodiesel blends due to variations in their fuel properties [24]. For the induction of producer gas and diesel supported dual fuel combustion exhibit 12.5% improved brake thermal efficiency as compared to dairy scum oil methyl ester based dual fuel combustion. For the hydrogen and 5% exhaust gas recirculation introduction the dairy scum oil methyl ester based dual fuel combustion presented recovered brake thermal efficiency by 6.1% compared to the dairy scum oil methyl ester-producer gas operation [26]. Brake thermal efficiency was higher for diesel-compressed natural gas gaseous combination compared to jamune biodiesel and its B20 blend due to high flame velocity and high energy content in diesel. Thermal efficiency for all fuel combinations was improved with 9 mm size venture compared to other ventures tried [27]. Brake thermal efficiency was lower at medium loads and increase slightly with higher loads. Higher thermal efficiency of 31.25% was obtained with 20% water in diesel emulsion along with manifold injected ethanol at 80% load conditions [28]. Advancing the injection timing from 23° to 27°BTDC resulted in increased thermal efficiency for all the fuel combinations considered. Further advancing the injection timing beyond 27 to 31°BTDC, the BTE decreased as more fuel burnt in the diffusion combustion phase [29]. For the same producer gas induction, the dairy scum oil methyl ester + multi walled carbon nanotubes combination exhibit poor thermal efficiency because of the increased viscosity of blended fuel and the reduced atomization of fuel droplets with nanoparticles, catalytic activity may be slowed [30]. As flow rate increased incomplete combustion occurred for all the tested fuels in compliance with reduction of air which leads to decreased to ignition source causes the depilation of thermal efficiency. At 0.25 kg/hr of flow rate leads to injection of more tested fuels which improved the utilization of gaseous fuel which results in to improvement of thermal efficiency. The combination of diesel-biogas showed maximum thermal efficiency as compared to other tested fuel combinations, diesel of its high calorific value and purified biogas with more methane percentage could be the reason [39].

* **Brake specific fuel consumption**

Brake specific fuel consumption value was less for the engine operation when carburettor-2 was used. Increase in injection timing from 19 to 27°BTDC, it was found decrement in the brake specific fuel consumption value [19-21].

* **Peak pressure rise**

Pressure rise found to be more for carburettor-2 for both diesel-compressed biogas and rice bran oil methyl ester-compressed biogas. The values of pressure rise for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas at 80% load found as 58.9 and 59.3 bar respectively [19]. As the injection timing was advanced the peak pressure increased more fuel injected into the engine cylinder. Higher thermal efficiency of the dual fuel engine resulted into higher peak pressure with advancing injection timing. Biodiesel-biogas dual fuel engines show higher delay period as related to single fuel operation [22]. Peak pressure of dual fuel engine increased when hydrogen flow rate increased for all combinations of fuels. Hydrogen being common B20 blends show higher peak pressures compared to pure biodiesels. At high hydrogen flow rates beyond 0.2 kg/h the engine experienced knocking behaviour with reduced peak pressures for all the fuel combinations [23]. For advanced injection timing more pilot fuel gets fed into the engine cylinder, which increased the brake power. That trend continued till 27°BTDC, after which the peak pressure begins to decline. When compared to diesel operation, biodiesel and biodiesel blends have reduced peak pressure [24]. As the number of holes on venture increased the peak pressure increased. Higher thermal efficiency of the dual fuel engine results into higher peak pressure with increasing holes on gas mixing venture. Biodiesel-biogas dual fuel engine showed higher delay period as related to compressed natural gas dual fuel operation due to higher methane content and calorific value of latter gas [27]. Dual fuel engine operation with B20 blends of jack fruit oil methyl ester B20, nigella sativa oil methyl ester B20 with hydrogen induction in dual fuel mode operation shows higher peak pressure than pure biodiesel [29].

* **Ignition delay**

As the injection timing was advanced the ignition delay period increased due to more fuel was injected inside the engine cylinder. Biodiesel-biogas dual fuel engines show higher delay period as related to single fuel operation. Biogas being same ceiba pentandra oil methyl ester B20 show lower ignition delay as compared to ceiba pentandra oil methyl ester B100 [22]. For biogas induction, and biodiesel blends injection longer delay times were obtained as compared to diesel fuel operation. Increasing the injection timing of the dual fuel engine till 27°BTDC resulted in lower ignition delay periods owing to better combustion of the fuels used. Due to higher calorific value of algae oil methyl ester B20 with biogas induction showed lower ignition delay than algae oil methyl ester B100 [24]. As the number of holes on venture increased the ignition delay period decreased. Biodiesel-biogas dual fuel engines show higher delay period as related to biodiesel-compressed natural gas fuel operation due to higher methane content of latter gas with higher in-cylinder pressures. Gaseous fuels show lower ignition delay with jamune oil methyl ester B20 compared to jamune oil methyl ester B100 [27]. Dual fuel engine operation with B20 blends of jack fruit oil methyl ester B20, nigella sativa oil methyl ester B20 with hydrogen induction in dual fuel mode operation showed higher delay period than pure biodiesel. The higher cetane number and lower viscosity of B20 blends ensures improved combustion and hence the delay period decreased [29].

* **Hydrocarbon and carbon monoxide emissions**

Hydrocarbon and carbon monoxide emissions were increased with increase in load and exhaust gas recirculation rate. Lower oxygen content available for combustion, resulted rich mixture which resulted incomplete combustion and resulted higher hydrocarbon and carbon monoxide emissions [18]. The hydrocarbon and carbon monoxide emissions were least for 3 mm carburettor. Hydrocarbon levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with 3 mm hole geometry carburettor were found to be 63 and 64 ppm respectively at 80% load. Carbon monoxide levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with 3 mm hole geometry carburettor were found to be 0.15% and 0.165% respectively at 80% load and 0.17% and 0.19% for 100% load [19]. As the injection timing increased, the hydrocarbon emission decreased considerably for both loads [20,21]. Hydrocarbon emission levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas dual fuel operation at 19, 23 and 27°BTDC injection timing, at 80% load are found to be 76, 68 and 63 and 84, 74 and 71 ppm respectively [20]. As the injection timing was advanced from 19 to 27° BTDC the carbon monoxide emission decreased considerably [20,21]. Carbon monoxide emission levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas dual fuel operation at 19, 23 and 27°BTDC injection timing, at 80% load was found to be 0.21, 0.14 and 0.12% and 0.24, 0.21 and 0.18% respectively [20]. Under dual fuel mode, the lesser velocity of biogas flame contributed more to form the hydrocarbon emissions. Hydrocarbon emissions are more for the raw biogas compared purified one the more viscosity and low calorific value may be the cause for that behaviour. Advancing the injection timing from 19 to 31°BTDC there was 15.24% reductions in hydrocarbon emissions were found [22]. Hydrocarbon and carbon monoxide emissions decreased for increased hydrogen gas flow rates as the pilot injected fuels are found to be lower. Hydrogen fuel being common, dual fuel engine fuelled with biodiesels showed higher hydrocarbon and carbon monoxide emissions as compared to diesel. B20 blends of the respective biodiesels showed lower hydrocarbon and carbon monoxide compared to pure biodiesels [23]. Advancement in the injection timing increased thermal efficiency of the dual fuel engine, resulted into reduced hydrocarbon and carbon monoxide emissions. Engine performance deteriorates after 27°BTDC [24]. Diesel fuelled dual fuel combustion provided lesser emissions of hydrocarbon and carbon monoxide by 37.2% and 34.1% as related with biodiesel based dual fuel combustion [26]. Hydrocarbon and carbon monoxide emissions were found to be lower using 9 mm venture. Between B100 and B20 blends with compressed natural gas showed lesser hydrocarbon and carbon monoxide levels as compared to B100 and B20 blends with biogas [27]. Declined emissions of hydrocarbon and carbon monoxide by 24.3% and 21% at 80% load were figured out when engine fuelled with an emulsion of 20% water in diesel with manifold injected ethanol compared to baseline diesel operation [28]. Dual fuel engine operation with B20 blends of jack fruit and nigella sativa with hydrogen induction in dual fuel mode operation shows lower hydrocarbon and carbon monoxide emissions than pure biodiesels [29]. Producer gas induction was the same, but biodiesel based dual fuel operation exhibited 32.6% and 29.8% higher hydrocarbon and carbon monoxide levels at an 80% load than diesel operation without nanoparticles. At an 80% load, carbon monoxide levels were lowered by 20.6%, 14.8%, and 8.2%, respectively, compared to biodiesel based dual fuel operation without nanoparticles [30].

* **Nitric oxide emissions**

Nitric oxide emissions decreased with increase in exhaust gas recirculation rate [18]. Nitric oxide levels for diesel-compressed biogas and biodiesel-compressed biogas operation with 3 mm hole geometry carburettor were found to be 943 and 815 ppm respectively at 80% load [19]. As the injection timing increased the emission of nitric oxide increased considerably [20,21]. The nitric oxide emissions under dual fuel mode are found to be lower as related to diesel fuel. On an average, by advancing injection timing from 19 to 31°BTDC there was 40.02% increment of nitric oxide emissions were found [22]. The B20 blends show comparatively more quantity of nitric oxide emissions as compared to B100 fuel operation. As the hydrogen flow rate increased the combustion activity increased which resulted into higher in-cylinder pressures and temperatures within the combustion chamber of dual fuel engine operation and hence nitric oxide emissions increased [23]. Adding nanoparticles in the algae biodiesel improves the dual fuel engine performance considerably with higher in-cylinder pressures and temperatures and hence higher nitric oxide missions were obtained compared to algae biodiesel and its blend [24]. For same producer gas induction, diesel based dual fuel combustion mode showed increased nitric oxide levels by 26.4% compared to dairy scum oil methyl ester based dual fuel operation. Dairy scum oil methyl ester-producer gas with hydrogen and 5% exhaust gas recirculation showed marginally greater nitric oxide levels by 12.4% as related to without exhaust gas recirculation. Greater exhaust gas recirculation rate lower the nitric oxide emissions by 32.2% as related to the combustion with 5% exhaust gas recirculation rate [26]. For all the dual fuel operation using 9 mm venturi showed higher nitric oxide emission levels compared to 2 mm and 6 mm venturies. Gaseous fuels being common jamune oil methyl ester B20 showed higher nitric oxide level due to blending with diesel which improves fuel property and increase engine temperature and heat release rate [27]. Declined emissions of nitric oxide by 10.8% at 80% load were figured out when engine fuelled with an emulsion of 20% water in diesel with manifold injected ethanol compared to baseline diesel operation [28]. The nitric oxide emissions with biodiesel operation was found to be lower in comparison to diesel mode. Advancing the injection timing from 23 to 27°BTDC resulted in higher nitric oxide as more fuel was injected into the engine cylinder with longer ignition delay. Further advancing the injection timing beyond 27°BTDC i.e., at 31°BTDC, the nitric oxide emissions decreased as lesser fuel participate in the controlled combustion phase with reduced thermal efficiency [29]. When comparing dairy scum oil methyl ester based dual fuel operation to fossil fuel-based dual fuel operation at an 80% load, experimental results showed that dairy scum oil methyl ester based dual fuel operating reduced nitric oxide levels by 12.8% [30].

* **Smoke opacity**

The smoke increased slightly as the exhaust gas recirculation rates increased [18]. The smoke opacity was lesser with mixing chamber venture having 3 mm hole geometry. Smoke values for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas operation with 3 mm hole geometry carburettor were found to be 61 and 68 HSU respectively at 80% load [19]. The smoke opacity decreased with increase in injection timing. As engine load increased, the smoke emissions increase slightly due to the decrease of air volumetric efficiency in dual fuel mode [20,21]. Smoke levels for diesel-compressed biogas and rice bran oil methyl ester-compressed biogas dual fuel operation at 19, 23 and 27°BTDC injection timing were found to be 66, 61 and 64 and 74, 68 and 71 HSU respectively at 80% load [20]. In comparison between ceiba pentandra oil methyl ester B100 and B20 with purified biogas in dual fuel mode operation showed less smoke emissions than B100 and B20 with raw biogas operation. Between B20 and B100 blends, the B20 showed less smoke due to blending of biodiesel with diesel which effects and improved the fuel property of B20 blend [22]. Less amount of smoke emissions are found with increased gas flow rates of hydrogen. More smoke emission for the pilot fuels was found with B100 blends when compared to B20 fuels. Nigella sativa oil methyl ester B20 showed higher smoke opacity than B20 blend due to fuel property variation [23]. As the performance of the dual fuel engine improved, smoke emissions drop until it reached to 27°BTDC, beyond which emissions rise again. Biodiesel and their mixtures being somewhat viscous, produce more smoke. Algae oil methyl ester B20 emits less smoke than B100 blend with biogas induction in dual fuel mode [24]. Diesel based dual fuel operation with producer gas exhibit reduced smoke levels by 27.4% than dairy scum oil methyl ester based dual fuel combustion. Dairy scum oil methyl ester-producer gas combination with hydrogen and 5% exhaust gas recirculation exhibited reduced levels of smoke by 10.2% as compared to same combination of fuel without exhaust gas recirculation [26]. The 9 mm venture resulted in lesser smoke opacity than other ventures tested. Higher smoke levels have been found for jamune oil methyl ester and biogas inducted operation compared to compressed natural gas operated dual fuel mode operation [27]. Lower smoke emissions of 41 HSU was obtained when engine was fuelled with emulsion of 20% water in diesel with manifold injected ethanol at 80% load condition. The smoke emissions were deteriorated by 27.9% with ethanol injection comparatively to that obtained with baseline diesel operation [28]. Nigella sativa oil methyl ester B20 showed lower smoke opacity compared to jack fruit oil methyl ester B20, as the former has comparatively lower viscosity and higher calorific value [29]. Increased dosage of nanoparticles in the fuel was limited and hence it resulted in reduced smoke levels. Therefore, the smoke levels obtained were proportional to the nanoparticle concentration [30].

**2.3 RCCI combustion**

* **Brake thermal efficiency**

There was decrease in thermal efficiency with increase in percentage of gasoline fuel. Higher thermal efficiency was found for diesel because of decrease in viscosity, higher calorific value of diesel results into proper mixing of air-fuel mixture and hence there was more thermal efficiency for diesel as compared to biodiesel [31]. The brake thermal efficiency of the engine improved for RCCI combustion mode as compared with homogeneous charge compression ignition (HCCI) mode up to 40% of gaseous fuels energy share while it diminished with further increment in gaseous fuels energy share [32]. Brake thermal efficiency decreased with the increased percentage of pentanol. Higher thermal efficiency of 22.15% was found for diesel and pentanol fuel combination at 10% of pentanol in injected fuels, which was about 9.1% and 27.3% higher than other fuel combinations [33]. The 75% load variation exhibit higher thermal efficiency as compared with 50% load variation. Higher thermal efficiency about 29.74% was obtained for diesel-compressed natural gas fuel combination at 75% load [34]. As the injection timing increased from 45 to 50°ATDC the brake thermal efficiency also increased. As the injection timing increased from 50 to 55°ATDC the brake thermal efficiency decreased [35]. The brake thermal efficiency of the engine increased as the gaseous fuels energy share increased up to 40% whereas it decreased beyond 40% [36,38]. The brake thermal efficiency of engine improved for compressed natural gas as compared with compressed biogas as low reactive fuel. [36]. The brake thermal efficiency decreased with the increase in n-butanol percentage. Highest thermal efficiency obtained for diesel and n-butanol fuel combination at 10% of n-butanol [37].

* **Specific fuel consumption**

There was increase in specific fuel consumption with increase in percentage of gasoline fuel. Higher fuel consumption was found for biodiesel operation compared to diesel. Gasoline being common the properties of diesel, biodiesel and its blends were responsible for the trends. Properties like lower calorific value and higher viscosity of biodiesel resulted into improper mixing of air-fuel inside the engine cylinder. Hence there was more fuel consumption for biodiesel operation as compared to diesel [31].

* **Nitric oxide emissions**

There was decrease in concentration of nitric oxide with increase in gasoline percentage. For biodiesel lesser nitric oxide was found due to lower in-cylinder pressure and temperature and increased residual time, also higher viscosity resulted into incomplete combustion which lowers in-cylinder pressure and temperature [31]. Nitric oxide emissions were diminished as the gaseous fuels energy share increased. Diesel and hydrogen fuel combination exhibit higher amount of nitric oxide emissions as compared with other fuel combination types [32]. Nitric oxide emissions were decreased with the increase in pentanol percentage. Highest nitric oxide emissions were obtained for diesel and pentanol fuel combination at 10% of pentanol [33]. The 75% load variation exhibit higher nitric oxide emissions as compared with 50% load variation. Higher nitric oxide emissions were obtained for diesel-compressed natural gas fuel combination [34]. Among various, injection timings studied 50°ATDC exhibit higher nitric oxide emissions as compared with 45 and 55°ATDC [35]. Nitric oxide emissions decreased as the amount of energy from gaseous fuels increased. As the energy share increased the fuel and mixture turns into lean mixture which results into low temperature combustion. That low temperature combustion strategy decreased the emissions of nitric oxide [36]. The nitric oxide emissions decreased with the increase in n-butanol percentage. Highest nitric oxide emissions obtained for diesel and n-butanol fuel combination at 10% of n-butanol [37]. RCCI engine powered with diesel and producer gas discharged higher nitric oxide emissions as compared with other fuel combinations [38].

* **Smoke**

Smoke emissions were reduced as the gaseous fuels energy share increased. The diesel and hydrogen fuel combination exhibit lower amount of smoke emissions as compared with other fuel combination types [32]. As the percentage of pentanol increased, smoke emissions were decreased. Lowest smoke emissions were found for diesel and n-pentanol at 10% of pentanol in injected fuels [33]. The 75% load variation exhibit lower amount of smoke emissions related with 50% load variation. Among various fuel combinations diesel and compressed natural gas exhibit lower quantity of smoke emissions related with other fuel combinations [34]. For RCCI combustion mode, very low levels of smoke emissions are obtained. Injection timing of 50°ATDC exhibited a fewer amount of smoke as related to 45 and 55°ATDC. Thevetia peruviana methyl ester operated RCCI combustion showed more quantity of smoke than diesel. Poor combustion characteristics of biodiesel injected RCCI combustion exhibited a large quantity of smoke emissions [35]. As the percentage of n-butanol increased smoke emissions decreased. Lowest smoke emissions obtained for diesel and n-butanol at 10% of n-butanol in injected fuels. Because of highly premixed n-butanol and more time of mixing for diesel, smoke emissions were very low [37]. RCCI engine powered with diesel and producer gas transmitted lower smoke emissions as compared with other fuel combinations. Higher smoke was found for biodiesel activity [38].

* **Hydrocarbon and carbon monoxide**

The hydrocarbon and carbon monoxide emissions were increased with the increase in gasoline content. Higher hydrocarbon and carbon monoxide emissions were found for biodiesel operation. Higher specific fuel consumption for biodiesel compared to diesel could be one of the reasons for higher hydrocarbon and carbon monoxide emissions [31]. The hydrocarbon and carbon monoxide emissions increased as the energy share increased. Diesel and hydrogen fuel combination exhibit lower amount of hydrocarbon and carbon monoxide emissions as compared with other fuel combination types [32]. The hydrocarbon and carbon monoxide emissions were increased with the increase in pentanol percentage. Lowest hydrocarbon and carbon monoxide emissions were obtained for diesel and pentanol at 10% of pentanol in injected fuels [33]. The 75% load variation exhibit lower hydrocarbon and carbon monoxide emissions as compared with 50% load variation. Out of the various fuel combinations studied diesel and compressed natural gas resulted into lower amount of hydrocarbon and carbon monoxide emissions [34]. Among various injection timings studied 50°ATDC exhibit fewer hydrocarbon and carbon monoxide emissions. Biodiesel powered RCCI combustion exhibit more number of hydrocarbon and carbon monoxide emissions [35]. The hydrocarbon and carbon monoxide engine-out emissions increased as the amount of energy from gaseous fuels increased. The hydrocarbon and carbon monoxide engine-out emissions were higher for compressed biogas as compared with compressed natural gas as low reactive fuel [36]. The hydrocarbon and carbon monoxide emissions increased with the increase in n-butanol percentage. Lowest hydrocarbon and carbon monoxide emissions obtained for diesel and n-butanol at 10% of n-butanol in injected fuels [37]. The hydrocarbon and carbon monoxide emissions diminished for diesel and producer gas powered RCCI combustion mode when compared with other fuel combinations [38].

* **Carbon di-oxide**

The carbon dioxide emissions were decreased with the increase in gasoline content. Higher emissions were found for diesel fuel because it burnt effectively as compared to B20 and B100 blends [31].

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

RCCI mode of combustion resulted in higher in-cylinder pressure as compared with other type of combustion mode. The pressure rise for diesel and hydrogen fuel combination was higher as compared with other fuel combinations in both combustion modes [32]. The highest pressure was found for diesel and pentanol fuel combination mode as compared with other tested fuels [33]. At all injection timings higher in-cylinder pressure was found for diesel and compressed natural gas fuel combination. The engine fuelled with biodiesel showed a lower rate of in-cylinder pressure rise as compared with diesel [35]. Higher peak pressure rise was found for diesel and compressed natural gas fuel combination as compared with other fuel combinations. For biodiesel operated RCCI engine peak pressure rise was lower as compared with diesel [36]. Highest pressure was found for diesel and producer gas fuel combination mode as compared with other tested fuels [38].

* **Heat release rate**

RCCI mode of combustion resulted in higher heat release rate as compared with other combustion mode. The heat release rate for diesel and hydrogen fuel combination was higher as compared with other fuel combinations in both combustion modes [32]. Highest heat release rate was found for diesel and pentanol fuel combination mode as compared with other tested fuels [33]. Among various fuel combinations, diesel and compressed natural gas showed higher heat release rate as related with other combinations of fuels. The engine fuelled with compressed biogas resulted in lower heat release rate as compared with compressed natural gas [35]. Higher heat release rate was found for diesel and compressed natural gas fuel combination as compared with other fuel combinations. Heat release rate for biodiesel operated RCCI engine was lower as compared with diesel [36].

**3. Conclusion**

From the exhaustive literature survey it is clear that, RCCI combustion is an effective ignition control maintaining high efficiency and low emissions. RCCI combustion is better promising low temperature combustion mode than others. In RCCI combustion, by adjusting ratio of high cetane fuel and high octane fuel the ignition timing can be controlled. In RCCI combustion nitric oxide emissions were decreased with the introduction of high reactive fuel. The hydrocarbon and carbon monoxide emissions were slightly increased for RCCI combustion. Biodiesel enables partial replacement for fossil diesel, decreasing the need for petroleum fuel and offers ecological energy supply.

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