Impact of low temperature combustion attaining strategies on diesel engine emissions for diesel and biodiesels: A review


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Abstract

Simultaneous reduction of particulate matter (PM) and nitrogen oxides (NO\textsubscript{x}) emissions from diesel exhaust is the key to current research activities. Although various technologies have been introduced to reduce emissions from diesel engines, the in-cylinder reduction techniques of PM and NO\textsubscript{x} like low temperature combustion (LTC) will continue to be an important field in research and development of modern diesel engines. Furthermore, increasing prices and question over the availability of diesel fuel derived from crude oil have introduced a growing interest. Hence it is most likely that future diesel engines will be operated on pure biodiesel and/or blends of biodiesel and crude oil-based diesel. Being a significant technology to reduce emissions, LTC deserves a critical analysis of emission characteristics for both diesel and biodiesel.

This paper critically investigates both petroleum diesel and biodiesel emissions from the viewpoint of LTC attaining strategies. Due to a number of differences of physical and chemical properties, petroleum diesel and biodiesel emission characteristics differ a bit under LTC strategies. LTC strategies decrease NO\textsubscript{x} and PM simultaneously but increase HC and CO emissions. Recent attempts to attain LTC by biodiesel have created a hope for reduced HC and CO emissions. Decreased performance issue during LTC is also being taken care of by latest ideas. However, this paper highlights the emissions separately and analyzes the effects of significant factors thoroughly under LTC regime.

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1. Introduction

The diesel engine is the most efficient type of internal combustion engine, offering good fuel economy and low carbon dioxide (CO\textsubscript{2}) emission [1]. Unfortunately, it is also a source of particulate matter (PM) and nitrogen oxides (NO\textsubscript{x}), both of which are now subjected to legislative limits because of their adverse effects on the environment and human health [2]. In the last few years, diesel engines have been subjected to progressively stringent emission control standards; especially as far as NO\textsubscript{x} and PM emissions are concerned. Fig. 1 shows this trend for Europe (Euro 2, 1996–Euro 5, 2008), the United States (US04–US10) and Japan. In order to meet the requirements of future emission standards, emission of these substances, as well as carbon monoxide (CO) and hydrocarbon (HC) emissions must be reduced significantly. Three general methods can be applied to the engines to meet lower regulated emission limits, viz. alternation of fuels [3,4], alternation of combustion processes and after-treatment of the exhaust [5]. Considerable progress has been made on both combustion and catalyst control pathways to reduce emission. Diesel particulate filters (DPF) for PM filtration and selective catalytic reduction (SCR) of NO\textsubscript{x} are now available for after-treatment of engine out emissions. Nevertheless, to minimize the cost and complexity of exhaust after-treatment systems as well as for potential fuel economy penalties—considerable research efforts have also focused on the in-cylinder control of emissions through the application of low-temperature combustion (LTC) techniques.

LTC is now widely demonstrated covering light-duty [7–11] to heavy-duty [12–14] engines. It is the concept at the heart of advanced diesel combustion. LTC is a general term for Homogeneous Charge Compression Ignition (HCCI) combustion, and Premixed Charge Compression Ignition (PCCI) combustion [5]. To explain the theory of LTC, Akihama et al. [15] simulated combustion by a compression ignition (CI) 3D-CFD KIVA2 model and plotted local equivalence ratio (Φ) vs. flame temperature (T) for the stratified combustion process. This particular figure showed the NO\textsubscript{x}–PM trade-off related to conventional diesel combustion, where at the edge of spray flame, fuel lean zones produce abundant NO\textsubscript{x} and fuel rich zones inside the spray flame produce abundant soot (an...
element of PM). With their model and $\Phi$–$T$ map they explained that LTC takes place at temperatures below the formation regime of NOx and at local equivalence ratios below the formation regime of diesel soot. As mentioned earlier, these systems can be divided into two categories [16]. Those in which the combustion phasing is decoupled from the injection timing and the kinetics of the chemical reactions dominate the combustion, are in the first category which is known as HCCI mode. In the second category, combustion phasing is closely coupled to the fuel injection event which is termed as PCCI mode. In the former category, air and fuel are thoroughly premixed in such a way that at the start of the combustion, the mixture is nearly homogeneous and characterized by an equivalence ratio, which is lower than 1 everywhere. For the second category, pre-mixing occurs between the fuel injection and start of combustion event, but significant regions exist where the equivalence ratio is greater than unity at the start of the combustion. Fig. 2 shows the plot of local equivalence ratio ($\Phi$) vs. flame temperature ($T$) with different combustion mechanisms. It can be seen that, NOx forms in the lean mixture zone where flame temperature is above 2200 K, whereas soot forms in the rich mixture zone above 1800 K. Conventional combustion overlaps the formation zones of NOx and soot, but LTC techniques like HCCI and PCCI avoid these zones and reduce NOx and soot simultaneously. Recently, a new approach of LTC, Reactivity Controlled Compression Ignition (RCCI) has been proposed by several authors [17–19]. This technology has the potential to overcome some of the limitations of HCCI and PCCI.

The objective of this article is to present the state of the art of the effects of different LTC mode (HCCI, PCCI, RCCI) attaining strategies on particular diesel emissions (NOx, PM, CO, UHC) using both petroleum diesel and biodiesel. The attainment of these strategies primarily depends on some factors like, application of exhaust gas recirculation (EGR), change in injection timing (IT) & injection pressure (IP), variation in compression ratio (CR) hence operating load, changes in fuel blends, etc. Therefore the analysis has been governed by these significant factors surely. To provide a complete overview of the whole scenario, more than 150 technical articles have been reviewed to collect significant information related to this article’s objective. At first, the article briefly introduces the LTC strategies and then analyzes how the attainment of these strategies may affect the emissions for petroleum diesel and biodiesel respectively. Though LTC mode has a positive impact on NOx and PM emissions but many of the researchers have reported reduced performance during LTC modes [20,21] due to higher rates of EGR and incomplete combustion. Impact of LTC modes on engine performance is also briefly presented here in this article.

2. LTC strategies

2.1. Homogeneous Charge Compression Ignition (HCCI)

HCCI engine is a combination of SI (homogeneous charge spark ignition) and CI (stratified charge compression ignition) engines with a sense that it uses premixed charge like SI engine but depends on autoignition like CI engine [22]. In HCCI, fuel is injected well before the combustion event which allows the homogeneous mixture of air–fuel. This homogeneous mixture initiates combustion simultaneously at different sites of the combustion chamber unlike SI (flame propagation) or CI (locally rich flame front) engines. With diesel fuel, HCCI combustion shows two-stage heat release. The first stage is low temperature kinetic reactions and the second stage is main heat release regime [23]. HCCI autoignition is controlled by low temperature chemistry and the main heat release is dominated by CO oxidation [24]. The main advantage of the HCCI combustion over conventional combustion mode is the reduction of NOx and soot in the exhaust. Though the concept gives higher indicated thermal efficiency, inability to control the combustion phasing has led the researchers to try different combustion control strategies e.g. port fuel injection [25,26], early direct injection [27,28], multiple fuel injection [29,30], compound combustion technology [31,32], narrow angle injection [33–35], late direct injection [36,37], variable inlet temperature, variable valve timing, internal or external EGR, etc. [22]. In addition, use of alternative fuels and fuel blends according to compression ratios and operating conditions have much potential to control the combustion phasing [22,38,39]. Actually, fuels with different autoignition points can be blended at varying ratios to control the ignition point at various load–speed regions [40]. This has yield alternative fuels to be tested in HCCI engines [41–51]. In diesel–fueled HCCI engines, these combustion control technologies are not often used alone. The combination of several strategies helps in achieving better effects on the combustion mechanism.

2.2. Premixed Charge Compression Ignition (PCCI)

Premixed charge compression ignition or the partially premixed charge compression ignition (PCCI) evolved from the HCCI combustion mode for the sake of better control over the start of combustion (SOC). In-cylinder homogeneity causes rapid combustion by simultaneous ignition throughout the cylinder space and produces great combustion noise in the HCCI mode. It is also very tough to control the combustion phases in HCCI mode. PCCI process is introduced to solve these problems. It is not fully homogeneous like HCCI. It achieves desired ignition delay through enhanced charge motion,
reduced compression ratio, higher injection pressure and extensive use of EGR. In the PCCI combustion process, fuel can be injected into the combustion chamber in three ways, they are, advanced direct injection, port fuel injection and late direct injection. Advanced direct injection and port fuel injection suffer from fuel spray impingement on the cylinder walls and incomplete fuel evaporation. Consequently HC and CO emissions increased [52,53]. However, narrow spray angle injectors and EGR reduce the wall impingement [35,54,55]. Late direct injection avoids the fuel-wall impingement and gives a way to switch the combustion style to the conventional at higher loads. Researchers have tried to increase the high load limits and reduce the emissions of PCCI by applying additives and tuning fuel properties [56,57], variable valve timing, multiple injections [58], and fuel–air mixing enhancement [10,59]. A new approach in PCCI introduces air–fuel premixing by early injection followed by a late injection of fuel pulse in the compression stroke, which governs the onset of ignition. Early injected fuel stratifies in the cylinder with the air and as the compression stroke reaches near the TDC (top dead center) it creates HCCI like condition. When the late direct injection occurs, the fuel-rich area of the late injection burns before the fuel-lean homogeneous mixture. This variable fuel–air mixture prevents the entire charge from igniting instantaneously which gives a better control over the combustion phase and rate. Moreover adoption of higher EGR permits longer ignition delay. It permits better premixing of air–fuel, results in less fuel-rich pockets followed by a low temperature combustion, which simultaneously reduces NOx and soot level [60].

2.3. Reactivity Controlled Compression Ignition (RCCI)

Reactivity controlled compression ignition is the newest approach where multiple fuels of different reactivity are injected at scheduled intervals which governs the reactivity of the charge in the cylinder for the desired combustion duration and magnitude. Mainly, in this approach, relatively low reactive fuel is injected (port injection) very early in the engine cycle which mixes with the air homogeneously. Later on, a higher reactive fuel is injected directly into the cylinder; it creates pockets of different air–fuel ratios and reactivity, which govern the onset of combustion at different times and rates.

This process of combustion originated from the effort of the researchers to reduce the EGR at higher loads while working on the PCCI regime. Inagaki et al. [61] investigated PCCI with two different reactive fuels and they succeeded to run the engine at higher loads (up to 12 bar) with minimal EGR. They reported, stratification of fuel reactivity made it possible to reduce the heat release rate and they achieved control over the combustion phasing beyond PCCI combustion. In RCCI combustion process, the combustion is staged [62] and proceeds from locally high reactivity fuel areas to low reactivity fuel areas. Such staging results in significant expansion of the premixed combustion duration and consequently produces high thermal efficiency, low pressure rise rate, low emission for higher loads up to 16 bar IMEP [63,64]. Therefore, as the combustion parameters are governed by the degree of reactivity of the charge in RCCI process, it is likely that, different operating conditions will need different fuel blends. For this reason, capability to operate with fuel blends covering the spectrum from neat gasoline to neat diesel fuel (low reactive to high reactive) is mandatory to get the best output from this kind of strategy.

3. Emission analysis under LTC modes

This section investigates emission characteristics for diesel and biodiesels under LTC modes. Results are summarized in Tables 1 and 2.

3.1. NOx emission analysis

3.1.1. Formation of NOx

NO (nitric oxide) and NO2 (nitrogen dioxide) are generally grouped under the NOx emission. But among the nitrogen oxides, NO is the predominant oxide produced inside the engine cylinder. Oxidation of the atmospheric nitrogen (molecular) is the main source of NO and this is called thermal NOx. Strong triple bond of nitrogen molecules breaks down by high combustion temperature and disassociated atomic state of nitrogen takes part in series of reactions with oxygen which results in thermal NOx. This mechanism is also known as Zeldovich mechanism [61,65]. However if the nitrogen content of the fuel is higher, then the nitrogen containing compounds get oxidized and become a potential source of NOx, which is also called fuel NOx. Formation of fuel NOx is quite complex because numerous intermediate species are there. Several hundred reversible reactions take place and still the true rate constant values are unknown. Another process of NOx formation is prompt mechanism. By this mechanism, the amount of NOx is quite lower than fuel and thermal NOx [66]. Mainly, free radicals formed in the flame front of the hydrocarbon flame generate this rapid production of NOx.

Formation of NOx generally depends on oxygen concentration, in-cylinder temperature, air surplus coefficient and residence time. NOx forms both in the flame front as well as in the post flame gases [67]. In engines, flame reaction zone remains extremely thin, as the combustion pressure is very high. In addition, residence time is short within this zone. On the other hand, the burned gases, which are produced early in the combustion process, are compressed to a higher temperature than they reached just after the combustion. That is why NO formation on the post flame gases usually dominates over the flame-front-produced NOx.

3.1.2. NOx emission under LTC modes for diesel

In LTC modes, the combustion temperature is reduced by premixed or leaner mixture with moderated use of EGR, consequently NOx emission reduces [68]. EGR hinders the O2 flow rate into the engine and results in reduced local flame temperature, which helps to reduce thermal NOx. Again EGR extends the ignition delay which indicates delayed start of combustion. It results in lower pressure and temperature rise during the combustion. The effect of late injection strategy on NOx emission is just like ignition delay [69]. Many researchers have attained LTC modes like PCCI, HCCI or RCCI, optimizing various parameters such as fuel reactivity (Cetane number, CN of fuel), injection timing and pressure, dilution of charge by EGR, controlling the operating load. Effects of these parameters for attaining the LTC modes are discussed below concerning the literature review.

Valentino et al. [68] tested blends of fuels having lower cetane number, higher resistance to autoignition and higher volatility than diesel fuels to reach partially premixed LTC mode. The fuels were neat diesel, 20% and 40% blend of n-butanol with diesel. Along with EGR, late injection and higher injection pressure gave them LTC mode for neat diesel. They reported that higher injection pressure allowed better mixing before the combustion and sufficient ignition delay provided by the use of EGR gave them a partially premixed LTC mode, which resulted in lower NOx. Again, blends of n-butanol with diesel gave them premixed LTC mode by elongating the ignition delay which can be attributed to the lower CN of n-butanol. Longer ignition delay permitted earlier injection as well as lower injection pressure with lower EGR rate to achieve the LTC mode and obviously lower NOx. Zhang et al. [21] also attempted lower cetane numbered gasoline and diesel fuel mixture (50:50) to reach premixed LTC mode. They used single advanced injection (up to 28° BTDC, before top dead center) with
Table 1  
Emission for diesel at LTC.

<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>UHC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S,4-cylinder, DI, TC</td>
<td>RS: 1500 rpm DV: 1.7 L IP: 1000 bar Injection system: common rail</td>
<td>PCI Load: 3.75 bar</td>
<td>D 8 to 25° BTDC</td>
<td>Up to 50%</td>
<td>E ↓ as EGR ↓, Very low (0.3 ppm) at 50% EGR</td>
<td>Increased</td>
<td>Increased</td>
<td>E ↓ as EGR ↓, Very low (0.03FSN) at 50% EGR and 25° BTDC</td>
<td>[72]</td>
</tr>
<tr>
<td>4S,4-cylinder, DI, TC</td>
<td>DV: 4.5 L CR: 16.57:1 RS: 2400 rpm RP: 115 kW/2400 rpm Injection system: common rail</td>
<td>Retarded injection LTC Speed: 1400 rpm Variable torque: 54–80 Nm</td>
<td>D Sweep of injection timing –8° to –2° ATDC</td>
<td>56%</td>
<td>E ↓ as IT retarded, 53% j as IT swept from –8 to –2° ATDC</td>
<td>Increased</td>
<td>Increased</td>
<td>E ↓ as IT retarded, 91% j as IT swept from –8 to –2° ATDC</td>
<td>[73]</td>
</tr>
<tr>
<td>4S,1-cylinder, DI super charged,</td>
<td>DV: 2.022 L RS: 1500 rpm CR: 14:1 Injection system: electronically controlled injector</td>
<td>LTC 25% load, charge air pressure: 1.3 bar (abs.)</td>
<td>D Sweep of IT 9 to 20° BTDC</td>
<td>Up to 60%</td>
<td>0 g/kW h</td>
<td>E ↓ as IT advanced, 28% j as IT swept from 9° to 20° BTDC</td>
<td>E ↓ as IT advanced, 71% j as IT swept from 9° to 15° BTDC</td>
<td>E ↓ as IT advanced, 92% j as IT swept from 9° to 20° BTDC</td>
<td>[20]</td>
</tr>
<tr>
<td>4S,1-cylinder, super charged, DI, WC DV: 781.7 cm³ CR: 13 RS: 1000 rpm Injection system: common rail</td>
<td>PCCI Injection pressure: 140 MPa Low sulfur diesel.</td>
<td>Sweep of IT 15 to 25° BTDC</td>
<td>0% and 40% Became 0 g/kW h as 40% EGR applied</td>
<td>E ↓ with advancement of IT and increment of EGR</td>
<td>E ↓ with advancement of IT and increment of EGR</td>
<td>Very high for such EGR level 56 g/kW h at 20° BTDC</td>
<td>Very high for such EGR level, 28.126 g/kW h at 20° BTDC</td>
<td>E ↓ as IT advanced, 93% j as IT swept from 9° to 15° BTDC</td>
<td>[74]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 2.34 L CR: 11.2 RS: 1200 rpm Injection system: common rail</td>
<td>Late injection LTC 152° injection angle 124° injection angle 160° injection angle</td>
<td>71% n-heptane, 29% iso-octane, 1% toluene</td>
<td>12.7% O₂ concentration N/A</td>
<td>12.7% O₂ concentration N/A</td>
<td>12.7% O₂ concentration N/A</td>
<td>Higher emission Higher emission Higher soot emission</td>
<td>Comparatively Lower emission than 152° injection angle Comparatively Lower emission than 152° injection angle Lower soot than 152° injection angle</td>
<td>[124]</td>
</tr>
<tr>
<td>Engine setup</td>
<td>Operating condition</td>
<td>Fuel</td>
<td>Injection timing</td>
<td>Percentage of EGR/O₂ concentration</td>
<td>NOₓ</td>
<td>CO</td>
<td>UHC</td>
<td>PM/Soot</td>
<td>Refs.</td>
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<tr>
<td>4S, 1-cylinder, DI DV: 373 cm³ CR: 15:1 RS: 1500 rpm IP: 100 MPa Injection system: Bosch common rail</td>
<td>HCCI Injection angle: 60°</td>
<td>D</td>
<td>Dual injection. Sweep of 1st injection: 50 to 70° BTDC. Sweep of 2nd injection: TDC – 20° ATDC</td>
<td>N/A</td>
<td>47% decrement as the 1st IT sweep from 50° to 60° BTDC while 2nd injection at 20° ATDC</td>
<td>360% increment as the 1st IT sweep from 50° to 60° BTDC while 2nd injection at 20° ATDC</td>
<td>40% increment as the 1st IT sweep from 50° to 60° BTDC while 2nd injection at 20° ATDC</td>
<td>N/A</td>
<td>[35]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI DV: 416 cm³ CR: 15:1 Injection system: Bosch common rail 2nd generation</td>
<td>HCCI</td>
<td>Fischer–Tropsch fuel, FAME, 20% ethanol</td>
<td>About 50%</td>
<td>On average 0.06 g/kW h</td>
<td>E ↓ as speed and load ↑, Almost 58% decrement.</td>
<td>E ↓ as speed and load ↑, Almost 67% decrement.</td>
<td>E ↓ as speed and load ↑, Almost 40% increment.</td>
<td>N/A</td>
<td>[77]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI, WC DV: 1.08 L CR: 16:1 RS: 1400 rpm Injection system: Common rail</td>
<td>EGR and toluene in the fuel paved LTC 70% n-heptane + 30% toluene, 80% n-heptane + 20% toluene, Commercial diesel fuel – 10° ATDC</td>
<td>12% O₂ concentration</td>
<td>Quite similar for all the fuels. On average 26 ppm.</td>
<td>E ↓ as % of n-heptane ↑, Commercial diesel gave highest emission</td>
<td>E ↓ as % of n-heptane ↑, Commercial diesel gave highest emission</td>
<td>E ↓ as % of n-heptane ↑, Commercial diesel gave highest emission</td>
<td>E ↓ as % of n-heptane ↑, Commercial diesel gave highest emission</td>
<td>Not affected by the change of speed and load</td>
<td>[166]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI DV: 2.44 L CR: 11.6:1 RS: 1200 rpm IMEP: 4.14 bar</td>
<td>RCCI Port injected fuel: iso-octane Direct injected fuel: n-heptane</td>
<td>Single direct injection: Sweep from 150° to 10° BTDC</td>
<td>N/A</td>
<td>Remained lower than 0.1 g/kW h, advancement after 60° BTDC caused rapid increment</td>
<td>Remained lower than 17.7 g/kW h, advancement after 70° BTDC caused rapid increment</td>
<td>Remained lower than 5 g/kW h, advancement after 50° BTDC caused rapid increment</td>
<td>Remained lower than 5 g/kW h, advancement after 40° BTDC caused rapid increment</td>
<td>N/A</td>
<td>[79]</td>
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</tbody>
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<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>UHC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 2.44 L</td>
<td>RCCI</td>
<td>N/A</td>
<td>Port injected fuel: gasoline</td>
<td>Direct injected fuel: gasoline + variable percentage of DTBP (di-tert-butyl peroxide)</td>
<td>N/A</td>
<td>43%</td>
<td>Remained quite lower than 0.1 g/kW h, No significant impact of DTBP percentage.</td>
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<td></td>
<td>CR: 16:1:1</td>
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<td>Lowest E was 4.5 g/kW h, E with the fraction of port fuel</td>
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<td>Direct IP: 400 bar</td>
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<td>No significant impact of% of DTBP, On average 3 g/kW h.</td>
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<tr>
<td></td>
<td>Port IP: 5.17 bar</td>
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<td></td>
<td>0.75% DTBP gave the lowest E, Average E level was below 0.005 g/kW h.</td>
<td></td>
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</tr>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 1.9 L</td>
<td>PPCI with advanced IT</td>
<td>Commercial diesel</td>
<td>26.6° BTDC</td>
<td>Sweep of O₂ concentration from 15 to 9%</td>
<td>N/A</td>
<td>E</td>
<td>288% as the load ↓ to 1.5 bar</td>
<td>E</td>
<td>430% as the load ↓ to 1.5 bar</td>
<td>N/A</td>
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<td>CR: 16:7:1</td>
<td></td>
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<td></td>
<td></td>
<td></td>
<td>E</td>
<td>140% as the%O₂ ↓ to 9% from 15%</td>
<td>E</td>
<td>260% as the%O₂ ↓ to 9% from 15%</td>
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<td></td>
<td>RS: 1500 rpm</td>
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<td></td>
<td>No significant impact of DTBP percentage.</td>
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<td></td>
<td>Rated IMEP: 4.5 bar</td>
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<td></td>
<td>4S, 4-cylinder, DI DV: 1.7 L CR: 16:1 RS: 1500 rpm BMEP: 4 bar Injection system: common rail</td>
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<tr>
<td>4S, 4-cylinder, DI</td>
<td>DV: 1.7 L</td>
<td>PCI</td>
<td>Low sulfur diesel</td>
<td>High injection rail pressure.</td>
<td>Advanced injection</td>
<td>Up to 50%</td>
<td>PCCI gave almost 86% ↓ than CC</td>
<td>PCCI gave almost 90% ↓ than CC</td>
<td>PCCI gave almost 46% ↓ than CC</td>
<td>PCCI strategy gave almost 51% ↓ than CC</td>
<td>[75]</td>
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<tr>
<td></td>
<td>CR: 16:1</td>
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<tr>
<td></td>
<td>RS: 1500 rpm</td>
<td>Commercial diesel</td>
<td>Commercial diesel</td>
<td>Ultra low sulfur Swedish diesel</td>
<td>Sweep from 8.5° to 4.5° BTDC</td>
<td>48%</td>
<td>[EGR and retarded IT gave ↓ NOₓ, 48% ECR gave lower than 1 g/kg-fuel all over the running condition.</td>
<td>[EGR and retarded IT gave ↓ E, About 28% ↓ for IT sweep from 6.5° to 4.5° BTDC at 48% EGR]</td>
<td>[EGR and retarded IT gave ↓ E, About 80% ↓ for IT sweep from 6.5° to 4.5° BTDC at 48% EGR]</td>
<td>Retarded IT gave ↓ E, About 53% ↓ as IT sweep from 6.5° to 4.5° BTDC at 48% EGR</td>
<td>[53]</td>
</tr>
<tr>
<td></td>
<td>IMEP: 2.6 bar</td>
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</tr>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 1.7 L</td>
<td>PCI</td>
<td>Low sulfur diesel</td>
<td>High injection rail pressure.</td>
<td>Start of ignition is at 6° ATDC on average</td>
<td>35–50% for 1.5 bar intake pressure 40–60% for 2 bar intake pressure</td>
<td>N/A</td>
<td>E</td>
<td>284% as the% of the gasoline ↑</td>
<td>N/A</td>
<td>[146]</td>
</tr>
<tr>
<td></td>
<td>CR: 15:1</td>
<td>Intake pressure 1.5 bar to 2 bar</td>
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<tr>
<td></td>
<td>RS: 1500 rpm</td>
<td>Commercial diesel</td>
<td>Commercial diesel</td>
<td>European diesel.</td>
<td>N/A</td>
<td>O₂ concentration 12.1%</td>
<td>N/A</td>
<td>66% ↓ as IT advanced from –30° ATDC to –33° ATDC</td>
<td>185% ↓ as IT advanced from –30° ATDC to –33° ATDC</td>
<td>[125]</td>
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<tr>
<td></td>
<td>IMEP: 7.5 bar</td>
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<td>IP: 1000 bar</td>
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<td></td>
<td>Injection system: common rail</td>
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<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 1.806 L</td>
<td>PCI</td>
<td>Commercial diesel</td>
<td>−24° ATDC to −33° ATDC</td>
<td>O₂ concentration 12.1%</td>
<td>N/A</td>
<td>N/A</td>
<td>66% ↓ as IT advanced from –30° ATDC to –33° ATDC</td>
<td>185% ↓ as IT advanced from –30° ATDC to –33° ATDC</td>
<td>[125]</td>
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<tr>
<td></td>
<td>CR: 14:4:1</td>
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<tr>
<td></td>
<td>RS: 1200 rpm</td>
<td>Commercial diesel</td>
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<tr>
<td></td>
<td>IMEP: 7 bar</td>
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<td>IP: 1450 bar</td>
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<td></td>
<td>Injection cone angle: 120°</td>
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<tr>
<td>Engine setup</td>
<td>Operating condition</td>
<td>Fuel</td>
<td>Injection timing</td>
<td>Percentage of EGR/O&lt;sub&gt;2&lt;/sub&gt; concentration</td>
<td>NO&lt;sub&gt;x&lt;/sub&gt;</td>
<td>CO</td>
<td>UHC</td>
<td>PM/Soot</td>
<td>Refs.</td>
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<tr>
<td>4S, 4-cylinder, DI, WC, TC DV: 1910 cm³ RS: 2500 rpm CR: 17.5:1 BMEP: 0.8 MPa Variable IP: 100–160 MPa Injection cone angle: 148°</td>
<td>PPCI</td>
<td>Low sulfur diesel, 20% and 40% blend of n-butanol with diesel</td>
<td>Retarded IT Sweep from to 2° ATDC</td>
<td>O&lt;sub&gt;2&lt;/sub&gt; concentration 19.5%</td>
<td>Retarded IT, lower IP and EGR gave ↓ NO&lt;sub&gt;x&lt;/sub&gt; for D Blends permitted slight advancement and reduction of IT and EGR respectively. N/A</td>
<td>↑ IP and ↑ EGR increased E Blends gave higher E than pure D</td>
<td>Higher% of n-butanol, retarded IT, higher IP and EGR gave ↓ PM</td>
<td>[68]</td>
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</tr>
<tr>
<td>4S, 4-cylinder, DI DV: 2198 cc RS: 3500 rpm CR: 16.6:1 Injection system: common rail Injection cone angle: 153°</td>
<td>PPCI 1800 rpm BMEP: 2.95</td>
<td>50% blend of gasoline with D Advanced injection at 28° BTDC</td>
<td>50%</td>
<td>Very low like 0.06 g/kW h</td>
<td>Higher value like 10.5 g/kW h</td>
<td>Higher value like 2.5 g/kW h</td>
<td>Very low like 0.008FSN</td>
<td>[21]</td>
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</tr>
<tr>
<td>4S, 1-cylinder DV: 500 cc CR: 18.4:1/16:1 RS: 1500 rpm IMEP: 3/7.7/10.8 bar Injection system: high pressure pump injection</td>
<td>PCCI Variable CR and IMEP</td>
<td>Ultra-low sulfur diesel</td>
<td>Sweep from to 3° ATDC</td>
<td>Up to 45.4%</td>
<td>E ↓ about 20% as CR ↓</td>
<td>N/A</td>
<td>N/A</td>
<td>Very high E for higher load. For lower load, E ↓ irrespective of CR</td>
<td>[76]</td>
<td></td>
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</tr>
<tr>
<td>4S, 1-cylinder, DI, WC RV: 1800 rpm CR: 17.8:1 IP: 120 MPa Injection system: common rail</td>
<td>PCCI Dual stage injection</td>
<td>15–20% blend of ethanol with diesel First injection: 60° BTDC Second injection: sweep from TDC to 15° ATDC</td>
<td>25%</td>
<td>E ↓ significantly as load ↓</td>
<td>E ↓ up to 71% as 2nd IT retarded from TDC to 15° ATDC at a fixed Φ E ↓ up to 100% as the 2nd IT retarded from TDC to 15° ATDC at a fixed Φ E ↓ up to 64% as 2nd IT retarded from TDC to 15° ATDC at a fixed Φ E ↓ about 160% as the 2nd IT retarded from TDC to 15° ATDC at a fixed Φ</td>
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<td>[71]</td>
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<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of ( \text{EGR/O}_2 ) concentration</th>
<th>( \text{NO}_x )</th>
<th>CO</th>
<th>UHC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S, 1-cylinder, DI DV: 638 cc CR: 17.7:1 RS: 140/180 MPa Injection system: common rail</td>
<td>PCI</td>
<td>20% blend of ethanol with diesel</td>
<td>Sweep from 20° to 40° BTDC</td>
<td>N/A</td>
<td>As IT advanced eventually got zero ( \text{NO}_x )</td>
<td>N/A</td>
<td>Very advanced and retarded IT gave ↑ E</td>
<td>N/A</td>
<td>[70]</td>
</tr>
<tr>
<td>4S, 1-cylinder, WC DV: 422 cm³ CR: 18.7:1 RS: 1500 rpm Injection system: common rail</td>
<td>Late injection premixed LTC</td>
<td>Diesel</td>
<td>Sweep from –30.25° to 7.75° ATDC</td>
<td>Up to 65%</td>
<td>As IT retarded and EGR ↑, E ↓ drastically. EGR higher than 60% gave quite zero level E</td>
<td>As IT retarded and EGR ↑, E ↓</td>
<td>N/A</td>
<td>As IT retarded and EGR ↑, E ↓ drastically. EGR higher than 60% gave quite zero level E</td>
<td>[16]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI DV: 2.44 L CR: 16.1:1 Injection system: common rail Direct IP: 800 bar Port IP: 4.14 bar</td>
<td>RCCI operating loads: 9.6–16.5 bar Port injected fuel: 15% gasoline + 85% ethanol Direct injected fuel: Diesel</td>
<td>Diesel</td>
<td>1st direct injection: 55° BTDC 2nd direct injection: 36° BTDC</td>
<td>Up to 47%</td>
<td>Remained lower than 0.15 g/kWh all through the running conditions. At higher loads increased value.</td>
<td>Decreased as load increased</td>
<td>Constant low E throughout the operating conditions.</td>
<td></td>
<td>[19]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI DV: 300 cc CR: 19.5:1 RS: 1500 rpm IP: 600–1000 bar IMEP: 3 bar Injection system: common rail Intake pressure: 1.4 bar</td>
<td>Retarded injection assisted premixed homogeneous combustion.</td>
<td>ULSD</td>
<td>Retarded single injection close to TDC</td>
<td>25%</td>
<td>↑ IP caused ↑ E, As the IT retarded emission ↓</td>
<td>N/A</td>
<td>N/A</td>
<td>↑ IP caused ↓ E, At retarded injection simultaneous ↓ of soot and ( \text{NO}_x )</td>
<td>[59]</td>
</tr>
</tbody>
</table>

E = Emission, CR = Compression ratio, RP = Rated power, RT = Rated torque, IP = Injection pressure, IT = Injection timing, BD = Biodiesel, D = Diesel, DV = Displacement volume, 4S = 4 stroke, DI = Direct injection, WC = Water cooled, TC = Turbo charged, NA = Naturally aspirated, CC = Conventional combustion, ELTC = Early LTC, LLTC = Late LTC.
<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>HC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>45,1-cylinder, DI</td>
<td>DV: 708 cm³</td>
<td>Diesel, 100%, 65% and 30% blend of colza biodiesel.</td>
<td>10° BTDC</td>
<td>Up to 32%</td>
<td>E ↓ as BD content ↓; E ↓ as EGR ↓; Up to 32% increment for BD than D</td>
<td>E ↓ as BD content and EGR ↓; Average 16% increment for BD than D</td>
<td>E ↓ as BD content and EGR ↓; Up to 52% increment for BD than D</td>
<td>E ↓ as BD content ↓; And E ↓ as EGR ↓; Up to 61% decrement for BD than D</td>
<td>[110]</td>
</tr>
<tr>
<td>45,1-cylinder, DI</td>
<td>DV: 857 cm³</td>
<td>100% soy biodiesel</td>
<td>17° BTDC</td>
<td>Up to 60%</td>
<td>Same trend like soy but relatively lower emission</td>
<td>Increased</td>
<td>Increased</td>
<td>Same trend like soy but relatively higher emission</td>
<td>[111]</td>
</tr>
<tr>
<td>45,4-cylinder, DI</td>
<td>DV: 4.5 L</td>
<td>Various blends of soybean oil derived biodiesel.</td>
<td>Sweep from −20° ATDC to 5° ATDC</td>
<td>30%</td>
<td>E ↓ as IP and BD content ↓; E ↓ as IT retarded. Up to 12% increment for higher IP for same blend</td>
<td>E ↓ as IP and BD content ↓; E ↓ as IT retarded. Up to 22% decrement for higher IP for same blend</td>
<td>E ↓ as IP and BD content ↓; Huge ↓ at IT beyond −5° ATDC</td>
<td>Pure BD and higher IP gives ↓ emission, Up to 33% decrement for higher IP for same blend.</td>
<td>[103]</td>
</tr>
</tbody>
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<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>HC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S, 4-cylinder, DI Late injection premixed LTC mode.</td>
<td>20, 50 and 100% Soy-based methyl ester</td>
<td>5°−7° and 9° BTDC</td>
<td>50%</td>
<td>E ↑ as IT advanced and BD content ↑. Up to 50% increment for BD than ULSD at retarded IT. Higher than 1500 ppm for all the cases.</td>
<td>E ↑ for retarded IT and ↓ for ↑ BD portion. Up to 42% decrement for BD than ULSD at retarded IT.</td>
<td>Very low especially for 100% BD.</td>
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<tr>
<td>4S, 4-cylinder, DI, TC Late and early injection partially premixed LTC</td>
<td>100% soy based methyl ester and 50% blend of soy based methyl ester with ULSD</td>
<td>5.9°−7.1° BTDC (for LLTC) 17°−24.1° BTDC (for ELTC)</td>
<td>45% (LLTC) 55% (ELTC)</td>
<td>N/A</td>
<td>N/A</td>
<td>E ↓ as BD content ↓. ELTC, LTC and CC gave 64%, 25% and 66% ↓ respectively for B100 than ULSD.</td>
<td>ELTC gave the highest E, 94% ↓ when used B100 than ULSD. CC gave lowest emission.</td>
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</tr>
<tr>
<td>4S, 1-cylinder, DI Late injection EGR assisted single injection LTC and pilot ignited HCCI combustion Variable BMEP: 3.3−8 bar</td>
<td>100% yellow grease based biodiesel.</td>
<td>17° BTDC (conventional single shot)</td>
<td>Up to 32%</td>
<td>As load ↓, EGR and BD content ↑ emission ↓ for single injection. Pilot-ignited HCCI ↓ emission.</td>
<td>Comparatively low E for pilot injection.</td>
<td>Same trend as CO. E ↑ as load and EGR ↑ for single injection. Pilot-ignited HCCI gave very low E.</td>
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<tr>
<td>4S, 1-cylinder, DI Late injection LTC</td>
<td>20%, 50% and 100% soy biodiesel</td>
<td>Sweep of injection timing from −25° ATDC to 3° ATDC</td>
<td>N/A</td>
<td>Retarded IT gave ↓ E than early IT; except early IT, E ↑ as BD content ↑. Up to 68% decrement than CC.</td>
<td>Very decreased emission for retarded injection.</td>
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<tr>
<td>4S, 4-cylinder, DI, TC Late injection EGR assisted LTC 1600 rpm and 25% loading.</td>
<td>40% biodiesel blended with ultralow sulfur diesel.</td>
<td>Single injection (6° BTDC) Double injection (pilot: 25° BTDC, main: 2° ATDC)</td>
<td>Up to 38%</td>
<td>As EGR ↑ E ↓, BD blend showed slight ↑ E than ULSD. Single injection with EGR showed better results than double injections.</td>
<td>N/A</td>
<td>N/A</td>
<td>N/A</td>
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**Table 2 (continued)**

DV: 1.7 L; CR: 16:1; RS: 1500 rpm; Load: 400 kPa;
DV: 1.7 L; CR: 16:1; RS: 1500 rpm; IP: on average 870 bar;
DV: 1.7 L; CR: 16:1; RS: 1500 rpm; IP: 800, 1000, 1200 bar;
DV: 17.3–24.1° BTDC (for ELTC) 17° BTDC (for LTC)
DV: 857 cm³; CR: 17.8:1; RP: 12.5 kW@2400 rpm; Injection system: pump injection.
DV: 2.5 L; CR: 17.5:1; RP: 103 kW@4000 rpm; IP: up to 1600 bar;
DV: 1.7 L; CR: 16:1; RS: 1500 rpm; IP: 800, 1000, 1200 bar; Injection system: common rail.
DV: 3.3−8 bar; Pilot (4−8) injections starting at 17° BTDC.
DV: 300 cc; CR: 19.5:1; RS: 1500 rpm; IP: 600 bar; Injection system: common rail
DV: 300 cc; CR: 19.5:1; RS: 1500 rpm; IP: 600 bar; Injection system: common rail.
DV: 3.3−8 bar; Single injection (6° BTDC) Double injection (pilot: 25° BTDC, main: 2° ATDC).
DV: 300 cc; CR: 19.5:1; RS: 1500 rpm; IP: 600 bar; Injection system: common rail.
<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>HC</th>
<th>PM/Soot</th>
<th>Refs.</th>
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</thead>
<tbody>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 0.477 L CR: 14: 1 RS: 1500 rpm IP: 860 bar Injection system: common rail</td>
<td>PPCI (1) IMEP: 3 bar, 1500 rpm, intake pressure: 1.5 bar (2) IMEP: 6 bar, 2000 rpm, intake pressure: 1.85 bar</td>
<td>20, 50 and 100% palm oil methyl ester and sunflower methyl ester blended with diesel fuel.</td>
<td>Sweep from 32° BTDC to 4° BTDC</td>
<td>O₂ concentration 9–10%</td>
<td>N/A</td>
<td>At 19° BTDC E was lowest for all the fuels. PME gave less E than SME. At ↑ load Minimum E shifted to advanced CA°</td>
<td>N/A</td>
<td>[163]</td>
</tr>
<tr>
<td>4S, 4-cylinder, DI</td>
<td>DV: 425 cm³ (1-cylinder) CR: 15:1 RS: 1500 rpm IP: 100 MPa Intake pressure: 120/150 kPa Injection system: common rail</td>
<td>Late injection premixed LTC Variable loading (IMEP: 0.25–0.65 MPa)</td>
<td>ULSD, 100% soybean methyl ester, biodiesel–ethanol (80–20%) blend.</td>
<td>Sweep from 8.3° to 7.5° BTDC (for 100% biodiesel) 13° to 10.5° BTDC (for biodiesel–ethanol)</td>
<td>40% for high load Up to 50% for low loads.</td>
<td>E less than 1 g/kg-fuel at each loading for BD-ethanol for other fuels at 0.4–0.65IMEP</td>
<td>N/A</td>
<td>N/A</td>
<td>For BD and BD-ethanol blend less than 0.25FSN found almost all over engine load.</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 522 cm³ CR: 17.5:1 RS: 1500 rpm Load: BMEP1.6 bar Injection style: common rail</td>
<td>HCCI Variable injection pressure (400–700 bar)</td>
<td>Rapeseed methyl ester</td>
<td>Advanced injection to create HCCI</td>
<td>N/A</td>
<td>E ↓ as IP ↑, Up to 33% decrement than D.</td>
<td>E ↓ as IP ↑ up to 43% decrement than D at higher IP</td>
<td>Tends to zero level of emission.</td>
<td></td>
</tr>
<tr>
<td>4S, 1-cylinder</td>
<td>Constant speed IMEP: 3 bar CR: 10.5:1 Injection system: Heated port atomization fuel system.</td>
<td>HCCI Variable intake air temperature (30–300 °C) 50% burn point constant to 365°CA</td>
<td>20% blend of palm, coconut, rape, soy and mustard oil methyl ester with ULSD</td>
<td>N/A</td>
<td>N/A</td>
<td>E ↓ with ↑ intake air temperature</td>
<td>E ↓ with ↑ intake air temperature</td>
<td>E ↓ with ↑ intake air temperature</td>
<td>N/A</td>
</tr>
<tr>
<td>4S, 1-cylinder, AC</td>
<td>DV: 517 cc CR: 10:5:1 RP: 7.9 kW/3000 rpm Injection system: air-assisted partial-vaporization port fuel injection.</td>
<td>HCCI Variable intake air temperature using 6 kW heater.</td>
<td>Soy biodiesel blended with ULSD up to 50%</td>
<td>N/A</td>
<td>N/A</td>
<td>E ↓ as the intake temperature ↓. At 160–170 °C intake temperature E was &lt; 1 ppm</td>
<td>E ↓ as intake temperature ↓, E quite same for all the blends.</td>
<td>N/A</td>
<td>N/A</td>
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<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>NOₓ</th>
<th>CO</th>
<th>HC</th>
<th>PM/Soot</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S, 1-cylinder, DI, AC</td>
<td>DV: 662 cm³</td>
<td>Fuel vaporizer with port fuel injection assisted HCCI. Variable loading</td>
<td>100% biodiesel</td>
<td>23° BTDC (for direct injection)</td>
<td>N/A</td>
<td>BD vapor induction gave very low E, Up to 87% decrement at 2–4 bars BMEP than DI system</td>
<td>As load ↓, E ↓, up to 20% decrement for BD vapor induction</td>
<td>As load ↓, E ↓, BD vapor induction emitted lowest</td>
<td>[113]</td>
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<tr>
<td></td>
<td>CR: 17.5:1</td>
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<td>N/A</td>
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<td></td>
<td>RP: 4.4 kW</td>
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<td>As load ↑, E ↑, [113]</td>
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<tr>
<td></td>
<td>RS: 1500 rpm</td>
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<td>As load ↑, E ↑, [113]</td>
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<tr>
<td></td>
<td>IP: 2 bar</td>
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<td></td>
<td>As load ↑, E ↑, [113]</td>
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</tr>
<tr>
<td>Injection system: port fuel injection with fuel vaporizer, direct injection</td>
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<tr>
<td>4S, 1-cylinder, DI</td>
<td>DV: 300 cc</td>
<td>Late injection HCCI</td>
<td>Neat soybean biodiesel and 20–50% blend of biodiesel with low sulfur diesel.</td>
<td>–25° ATDC, –10° ATDC and 3° ATDC</td>
<td>N/A</td>
<td>At IT – 25° ATDC very ↓ E, At 3° ATDC the lowest E for all fuel blends, E ↑ as BD content ↑</td>
<td>N/A</td>
<td>N/A</td>
<td>At 3° ATDC simultaneous reduction of soot and NOₓ</td>
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<td>CR: 19.5:1</td>
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<td>RS: 1500 rpm</td>
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<td>IP: 600 bar</td>
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<td>Injection system: common rail</td>
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<td>Injection cone angle: 150°</td>
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<td>4S, 4-cylinder, DI</td>
<td>DV: 1.7 L</td>
<td>Single late injection</td>
<td>Neat soy-based methyl ester.</td>
<td>Late injection</td>
<td>45%</td>
<td>Very low E, Within the range of 26–35 ppm</td>
<td>About 17% less emission than ULSD</td>
<td>About 30% less emission than ULSD</td>
<td>Engine out PM was over an order of magnitude higher than ULSD</td>
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<td>CR: 16:1</td>
<td>Premixed LTC</td>
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<td></td>
<td>RS: 1500 rpm</td>
<td>Intake pressure: 100 kPa</td>
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<td>RP: 11.3 bhp</td>
<td>IP: 1007 bar</td>
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<td>4S, 4-cylinder, DI, TC</td>
<td>DV: 2 L</td>
<td>Early injection LTC and late injection Nissan MK type combustion engine with European diesel</td>
<td>50% and 90% blend of rapeseed methyl ester with European diesel</td>
<td>–30° to –10° ATDC for low load, At TDC for higher load.</td>
<td>Up to 60% Two stage EGR cooler configuration.</td>
<td>For early IT, E ↑ as IP and BD content ↑, At ↑ loads retarded IT gave low E like 30 ppm at high EGR</td>
<td>For early IT, E ↑ as IP ↑, For late IT, E ↑ as EGR and BD content ↑, For late IT, E ↑ as EGR and BD content ↑</td>
<td>At higher load, ↑ EGR and retarded IT gave ↑ E</td>
<td>[104]</td>
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<td>CR: 14.4:1</td>
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<td>RS: 1600 rpm</td>
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<td>IP: 1600 bar</td>
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340

extensive EGR (50%) to gain PPCI mode and reported very low amount of NOx (0.06 g/kW h). Masuda and Chen [70] also got same type of results by the use of ethanol. Mohammadi et al. [71] tried two-stage injection with 15–20% blend of ethanol with diesel fuel to reach PCCI condition. First injection was at 60° BTDC and they reported that retarding the second injection along with 25% EGR improved the NOx emission. Low cetane number of ethanol permitted very early injection giving a long ignition delay and moderated use of EGR suppressed the possible NOx emission from the second injection.

Not only the lower CN fuel blend but also late injection can elongate the ignition delay to reach premixed LTC mode. Jacobs and Assanis [72] experimented with retarded injection and high EGR rate which gave them increased ignition delay and combustion duration. Such experimental data confirmed the achievement of PCCI (premixed compression ignition) which showed reduced NOx. As the EGR rate increased and injection time retarded, emission of NOx decreased. Bittle et al. [73] reported application of immense EGR gave them about 94% decrement of NOx while they were trying to get a universal determination of LTC mode attainment criteria. Retardation of injection timing with EGR gave them even better results of emission. Han et al. [53] also reported same results by the use of cooled EGR. However, Kiplimo et al. [74] reported lower NOx even in early injection (20° BTDC) during PCCI combustion strategy. With EGR, they got about 75% decrement of NOx emission and negligible difference of IMEP. They worked out an optimum spray-targeting zone where they got simultaneous reduction of CO, HC and soot but could not manage to reduce the NOx without EGR. Without EGR, lower injection-pressure (80 MPa) with late injection (2–15° BTDC) gave reduced amount of NOx while for higher injection pressure (140 MPa) advanced injection (20–40° BTDC) resulted in reduced NOx. This lower NOx for higher injection pressure at advanced injection timing can be attributed to the achievement of PCCI regime. PCCI regime ensured lower in-cylinder temperature and longer premixing time, which resulted in lower NOx [75]. On the contrary, Alikronsson and Denbratt [20] reported higher NOx when they tried advanced injection timing to keep the BSFC (brake specific fuel consumption) low. However, Kook et al. [16] investigated the effect of dilution and injection timing very precisely on low temperature combustion emission. In a fixed SOI (start of injection) they observed that NOx emission decreased as the dilution increased. NOx emission was actually correlated with the adiabatic flame temperature. Again, in a fixed level of dilution, retardation of injection timing gave lower NOx. They commented that earlier injection timing assisted by high level of dilution, generated greater adiabatic flame temperature than the late injection even with less amount of dilution. From NOx formation point of view, we can infer that, as late injection LTC mode generates lower temperature than early injection LTC mode, the former one is better in this regard.

Laguitton et al. [76] observed the effect of compression ratio on NOx emission under a wide range of PCCI like combustion styles. Lowering the compression ratio gave them lowered NOx. This effect was more pronounced at higher loads. They also observed that at higher loads, combustion style proceeded to premixed-charge from the combination of premixed and diffusion type combustion as the injection timing was swept from very early to retarded. Hence, NOx emission was converged as the diffusion combustion suppressed. They also concluded that below a certain combustion temperature, as in fully premixed charge combustion, NOx emission was dominated by the air fuel ratio and the local oxygen concentration rather than in-cylinder pressure and temperature.

However, HCCI combustion mode has also been cited for reduced NOx emission. Pidol et al. [77] evaluated that ethanol–diesel–biodiesel blend could be used to keep NOx under the HCCI acceptance criteria (<0.1 g/kW h) even at higher loads. They used two types of ethanol–diesel–biodiesel blends. In one type of blend fossil diesel was used and to another synthetic Fischer–Tropsch diesel was used. In both cases 20% ethanol was used. Use of ethanol expanded the ignition delay which helped to reach HCCI mode. They succeeded to increase the IMEP to 12.2 bar and 11 bar for fossil diesel and Fischer–Tropsch diesel blend respectively while for both of the cases NOx emission was below 0.10 g/kW h. Reason for sustaining higher load with lower NOx of fossil diesel–biodiesel–ethanol blend can be attributed to the lower cetane number of the blend while Fischer–Tropsch diesel–biodiesel–ethanol blend sustained at bit lower load for the reason of higher CN of Fischer–Tropsch diesel. Again higher volatility contributed by the ethanol permitted lower injection pressure to reach homogeneity which reduced the combustion rate hence lowered NOx. Kim and Lee [35] also cited very low NOx at HCCI combustion style.

While researchers were trying to increase the operating load under LTC mode keeping the NOx lower, RCCI combustion mechanism gave them very good results. Splitter et al. [19] got very reduced NOx (below than 0.1 g/kW h) while they tried gasoline as the low reactive fuel and diesel as the high reactive fuel at higher loads like 14.5 bar. Though use of ethanol–gasoline blend (85% ethanol) as the low reactive fuel gave a bit higher NOx emission, but due to its lower cetane number it sustained even higher load (16.5 bar) than the gasoline. This dual-fuel mechanism needed comparatively lower EGR to keep the NOx lower, which increased the thermal efficiency as well. Splitter et al. [76] also tried single fuel stock as the basis for both high and low reactive fuels. They reported lower NOx but the load level was lower than the former experiments. They reported that at about 9 bar load the emission characteristics were just similar to the dual fuel approach of RCCI. Again Splitter et al. [79] tried to reveal the impacts of injection timing on emission within RCCI combustion process. They injected iso-octane by port injection and n-heptane by direct injection. They reported, regardless of the single or double direct injection, injection timing had minimal effect on reduced NOx emission except beyond −60° ATDC (after top dead center). NOx increased beyond −60° ATDC for both injection style and that can be attributed to the less available mixing time. Rapid ramp on the heat release rate at that injection timing clarified the scenario. For double direct injection, they succeeded to retard the injection timing 10 more crank angle degrees as for such injection, fuel mass was more mixed.

However, after such discussion we can come to some salient points which are following:

- Regardless of PCCI, HCCI or RCCI combustion modes, lower NOx depends on higher ignition delay and lower combustion rate which result in lower in-cylinder temperature and pressure rise rate.
- Below a certain combustion temperature, NOx emission is controlled by air–fuel ratio and local oxygen concentration more than in-cylinder temperature.
- Late injection premixed LTC is better than early injection premixed LTC mode regarding NOx emission.
- Advanced injection assisted HCCI combustion mode needs higher EGR to control NOx emission as advanced injection causes higher in-cylinder temperature.
- RCCI combustion mode has succeeded to keep the NOx level lower at higher loads even with low EGR rate by the help of fuel reactivity gradient inside cylinder.

### 3.1.3. NOx emission under LTC modes for biodiesels

Increased NOx is an established phenomenon while using biodiesel in internal combustion engines [80–82]. This increment is not solely controlled by the change of a single fuel property, rather some coupled mechanisms which may strengthen or cancel one another in various circumstances depending on combustion and fuel properties [83]. Potentially contributing factors to make
differences in NO\textsubscript{x} emission for biodiesel can be summarized as injection timing, injection pressure-spray-mixing, ignition delay, combustion stages and heat release, heat radiation from soot, combustion temperature, fuel unsaturation and system response issues [84].

Due to higher densities, bulk modulus of compressibility and speed of sound, start of fuel injection is advanced for biodiesel relative to petroleum diesel in rotary/distributor-style fuel injection systems [85,86]. An advance in injection timing is considered as a main reason for observed increases in NO\textsubscript{x} emissions with biodiesel as it helps to elevate diffusion reaction temperatures and ultimately post flame gas temperature. Of course, this incident is not present in common rail fuel injection system [87]. Szybist et al. [88] investigated NO\textsubscript{x} emission characteristics of different fuels including biodiesel, altering injection timings at high and low load conditions. They observed at higher loads, relation between NO\textsubscript{x} emission and injection timing was independent of fuel types but at low loads emission characteristics were unique for each fuel types. Therefore, it can be said that, for higher loads increase in NO\textsubscript{x} emission is due to the advancement of injection timing but this is not true for low loads. This confirms the existence of other factors for increased NO\textsubscript{x} for biodiesels along with the advanced injection timing. Such as, biodiesels have higher cetane number which depicts shorter ignition delay [89—92]. A short ignition delay reduces the premixed burn, consequently increases the fraction of diffusion burn [93]. In the diffusion stage, the equivalence ratio at the flame front is essentially always at a stoichiometric value [94]. Therefore, once the fuel is largely being consumed in a diffusion flame, it is more relevant to consider the oxygen fraction within it. It is well-known that higher oxygen fractions yield higher diesel combustion temperatures and NO\textsubscript{x} formation rates for diffusion flame [95—97]. Ullman et al. [98] has confirmed that, because of increased oxygen content and decreased sulfur content, PM formation is comparatively low in the biodiesel combustion than petroleum fuel. Less PM depicts less radiation heat transfer which increases post-flame gas temperature therefore increased NO\textsubscript{x} emission [84]. Again biodiesels have got higher degree of unsaturation [99,100], and Graboski et al. [101] reported increase in NO\textsubscript{x} emission, with the increase in unsaturation and decrease in mean carbon chain length. Finally, the changes in NO\textsubscript{x} emission for biodiesel are largely dependent on pre-combustion chemistry of hydrocarbon free radicals [102]. It incorporates prompt mechanism of NO\textsubscript{x} formation more in consideration, because it is more sensitive to radical concentration within the reaction zone whereas thermal mechanism remains quite unaffected by fuel chemistry.

Low temperature combustion is a promising technique for NO\textsubscript{x} reduction not only for petroleum diesel but also for biodiesels, though they produce much higher NO\textsubscript{x} than petroleum diesel as discussed earlier. Veitman et al. [103] experimented sweep of SOI from −20\textdegree ATDC to TDC with a common rail injection with moderated EGR to gain premixed LTC. Electronically controlled injection system ensured same injection timing for all the fuels. Still higher biodiesel content showed higher NO\textsubscript{x} emission which contradicts the so called general clarification (advanced injection for higher density) of the higher NO\textsubscript{x}. They got reduced NO\textsubscript{x} (less than 0.5 g/kWh) at 30% EGR at very retarded SOI due to lower combustion temperature. Though higher injection pressure caused higher emission for increased combustion temperature, it was insignificant at higher EGR as the emission was already low. Weall and Collings [104] also reported higher NO\textsubscript{x} emission for higher injection pressure at premixed LTC. Along with EGR and injection pressure, intake pressure has also been cited for having command on NO\textsubscript{x} emission in premixed LTC. NO\textsubscript{x} Emission decreases as intake pressure increases for biodiesels [105,106]. Better premix of charge was responsible for such results.

Fang et al. [107] claimed that even in premixed LTC mode, oxygen content in biodiesel dominated the NO\textsubscript{x} emission more than ignition delay while they tried a sweep of SOI from −25\degree ATDC to 3\degree ATDC. They observed higher ignition delay of biodiesel than European low-sulfur diesel, which attributed to lower cetane number and higher boiling point of biodiesel that slowed down the droplet evaporation rate hence preparation of the ignitable air–fuel mixture. In spite of higher ignition delay, increasing portion of biodiesel showed increasing NO\textsubscript{x} at the conventional and late SOI. They attributed this phenomenon to the higher oxygen content of the biodiesel. They suggested a trade-off between ignition delay and oxygen concentration was responsible for this incident and concluded commenting that late SOI was better to reduce the NO\textsubscript{x} emission than early SOI. Similarly, Zheng et al. [108] observed lower NO\textsubscript{x} for late injection but unlike Fang et al. [107], they observed higher cetane number of biodiesels and commented that for this reason biodiesels sustained late SOI as well as EGR-incurred LTC better.

Along with the oxygen concentration, injection timing and ignition delay, combustion phasing has influence on NO\textsubscript{x} emission in the case of premixed low temperature combustion. From a common baseline condition of combustion, created by keeping the 50% mass fraction of the fuel burned point constant, Northrop et al. [109] got the NO\textsubscript{x} emission curves more or less same for all the fuels they tested. It proves the command of combustion phasing on NO\textsubscript{x} emission in premixed LTC modes. They also observed the combustion location as a dominant factor of NO\textsubscript{x} emission.

To reduce NO\textsubscript{x} emission from biodiesel combustion, HCCI has also been tried by the researchers. Jiménez-Espadafor et al. [110] showed the effect of EGR on NO\textsubscript{x} emission at late injection HCCI. They got the same story of EGR and biodiesel content like premixed LTC. Interestingly they observed that even on higher EGR, higher biodiesel content showed lowest ignition delay and they suggested that ignition delay relied more on chemical kinetics mechanism than the temperature reduction made by EGR. However, shorter ignition delay may also produce less NO\textsubscript{x} if the ignition delay is short enough to make a weak mixture [111]. If the mixture gets close to the stoichiometry then again the NO\textsubscript{x} will be higher.

EGR assisted single injection LTC and pilot ignited HCCI combustion were investigated by Zheng et al. [112]. They reported that EGR was the instrumental factor to reduce the NO\textsubscript{x} at single shot injection by reducing in-cylinder flame temperature and diluting the oxygen concentration. Pilot ignited HCCI with immense EGR reduced the NO\textsubscript{x} emission even more by helping to overcome mixing problem which led to homogeneity of the mixture. Later, the same authors [111] experimented with various biodiesels with the same setup and got the same trend of results including lower emission for lower loads and higher emission for the higher loads. This can be attributed to the higher flame temperature for higher loads and vice versa. However, they [105] got an improvement at higher loads while they tried two early injections with higher boost pressure. They mentioned better combustion process and improved combustion phasing due to enhanced fuel–air mixture responsible for such improvement. Recently, Ganesan et al. [113] tried a unique technique to reach HCCI like combustion and they succeeded to keep the NO\textsubscript{x} substantially low. They used a fuel vaporizer with port fuel injection to achieve the mixture homogeneity as well as to attain HCCI like combustion process which gifted low level of NO\textsubscript{x}. Pidol et al. [114] used ethanol–diesel blend, stabilized by biodiesel, and they got quite...
low NO\textsubscript{x} all over the engine loading range. Low soot tendency of ethanol permitted higher EGR which helped to reduce NO\textsubscript{x} drastically. Again low ignitability let the mixture to be homogeneous which was conducive to the LTC.

However, some points can draw a summary to this discussion:

- Increment of NO\textsubscript{x} for biodiesel is a matter of aggregated factors. Only advancement of injection timing or lower ignition delay is not responsible for this.
- Higher percentage of EGR or late injection timing increases ignition delay which can reduce the NO\textsubscript{x}.
- Ignition delay relies more on chemical kinetics mechanism of biodiesel than temperature reduction made by EGR.
- Keeping combustion phasing constant for different biodiesels by tuning concerning parameters gives similar trend of NO\textsubscript{x} emission. It establishes that combustion phasing is one of the dominant parameters regarding NO\textsubscript{x} emission of premixed LTC.
- Late SOI is better than early SOI for lower NO\textsubscript{x} emission in biodiesel combustion. Higher percentage of oxygen and higher cetane number of biodiesel can sustain higher EGR and late SOI incurred LTC better. Therefore, better NO\textsubscript{x} reduction.

3.2. PM emission analysis

3.2.1. PM formation

Diesel particulates are principally combustion generated carbonaceous material (soot) where some organic compounds remain absorbed as well [67] and grow via gas to particle conversion process [115]. Diesel engines significantly emit particulate matter. To be precise, particulate matter is a highly complex mixture of fine particles and liquid droplets including soot, ash, hydrocarbon soluble organic fraction (SOF) and water SOF [99]. PM varies in size, shape, number, surface area, solubility, chemical composition and origin [99,116,117]. Size distribution of the PM has three modes consisting coarse particles, fine particles, and ultrafine particles [118,119]. These particles exist in various shape and densities in the air thus aerodynamic diameter is used to define the size of the particle [120]. Soot particle size can be as small as 1–2 nm at initial state [66]. Collision of rings causes coagulation and clustering together similar to a chain, making the soot grow to agglomerates with size ranging 100–1000 nm. Soot content in the exhaust gas is indicated by the smoke opacity; hence, this parameter can be correlated with fuels tendency to form PM during combustion.

Incomplete combustion of fuel hydrocarbons produces most of the particulate matter with little contribution of lubricating oil. It sources from the rich combustion zones where the equivalence ratio is higher than 1. This is the reason for the highest particulate concentrations in the core region of each fuel spray in direct injection diesel engines [67]. Generally, soot formation takes place at higher than 1800 K temperature in diesel combustion environment. Net soot release is commonly defined as the difference between formation and oxidation of soot. Formation and oxidation of soot are strongly coupled with the combustion temperature just like the NO\textsubscript{x} formation. So, conventionally soot and NO\textsubscript{x} formation have got an inverse relation known as soot–NO\textsubscript{x} tradeoff [73].

3.2.2. PM emission under LTC modes for diesel

In LTC, simultaneous reduction of soot and NO\textsubscript{x} are achieved by reducing the combustion temperature lower than soot formation level. Once LTC is attained, soot formation loses its strong dependence even on equivalence ratio [72]. In LTC mode soot formation occurs primarily downstream in the head of the jet [121]. This is in contrast to the upstream soot-producing core, in conventional diesel jets [122]. This shift is due to the charge dilution employed in LTC and mixing between the end of injection and second-stage ignition [123].

Many researchers have reported successful reduction of soot when they attained LTC. Actually more complex relationship exists in soot reduction than that of NO\textsubscript{x}. Soot oxidation process is more sensitive to temperature than the soot formation process [73]. Therefore, when application of EGR reduces the combustion temperature, oxidation rate falls dramatically and emission of soot increases. Further reduction of temperature by higher level of EGR or retarding the SOI, below the soot formation level gives very low amount of soot. For example, PCI combustion condition attained by late injection have been reported to produce very low amount of PM at about 48% EGR [53]. Alriksson and Denbratt [20] also reported that they needed almost 50% EGR even on 25% loading when the peak combustion temperature was comparatively low. They observed an increment of soot up to 50% EGR and then it suddenly decreased, which supports the soot oxidation and formation relationship discussed previously. Genzale et al. [124] studied the effect of spray targeting on engine out soot emission of a heavy-duty optical diesel engine in LTC regime achieved by late injection. They measured fuel-vapor concentration by fuel-tracer (toluene) fluorescence and measured OH, PAH fluorescence and combined formaldehyde to evaluate combustion, mixing and formation of pollutants. They observed that spray pattern with included angle of 152° injected fuel towards the vertical center of the piston bowl and wall impingement occurred. Also, merging with neighboring jets happened before the peak heat release. Near the piston bowl floor, jet–jet interaction created fuel-rich regions which were the main sources of PAH (poly aromatic hydrocarbon, precursor of soot) formation. They concluded that narrow injection angle (124°) or wider injection angle (160°) could avoid this problem.

It is reported that PCI or PCCI achieved by advanced injection and extensive use of EGR also produced lower soot emission. Jacobs and Assanis [72] experimented with an AF (air–fuel) ratio which is lower than stoichiometric (14.7) and the injection timing was 25° BTDC, confirming the low peak combustion temperature in the combustion. They reported tremendous low soot (0.03 FSN) emission. Though, lower global AF ratio indicated rich mixture, soot decreased. Therefore, within the PCI combustion regime net soot release has very little sensitivity to local equivalence ratio. Parks li et al. [75] also reported the same result when they attained PCCI by advanced injection timing and extensive EGR. Although PM emission decreased by about 51% than conventional combustion, PM size was smaller and the soluble organic fraction was increased. But opposite results are also been reported by Benajes et al. [125]. They studied the PM emission with advanced injection timing within the premixed LTC regime. Advancing the injection timing empowered them to control ignition delay and local equivalence ratio of the injected fuel. Higher value of the ignition delay and maximum local equivalence ratio for advanced injection timing were supposed to give less PM formation. Surprisingly they observed increased PM with injection advancement. Most drastic increment of PM (185%) happened when they advanced the injection from −30° ATDC to −33° ATDC. Though here the end of injection occurred before the start of combustion as in premixed LTC mode, PM formed because of fuel deposition on the surface of piston bowl. As the injection was advanced, the fuel spray trajectory crossed the piston bowl surface at a higher point. It caused increased liquid fuel spray and combustion chamber surface interaction, hence more deposition. This was the reason behind increased PM formation, though it was a premixed LTC mode.

Researchers have also reported decreased PM for HCCI combustion strategy [14,77,126]. Cracknell et al. [126] investigated the ef-
fect of a broad range of fuel properties on HCCI combustion strategy. They observed several fuels, at certain speeds and loads, broke the NOx-PM trade-off curve and produced simultaneous reduction of NOx and soot. But as the load increased, all fuels tended to revert to classic diesel NOx-PM trade-off curve. Singh et al. [14] studied the effect of dual injection strategy in the HCCI combustion process. They started the first injection early (22° BTDC) with low load condition which helped the fuel to be premixed well. Earlier injection caused cool-flame heat release followed by second stage combustion. As the fuel was premixed well, soot formation was low because there were less fuel-rich soot pockets. At the second injection (15° ATDC), the combustion was much like conventional diesel combustion process. Significant mixing-controlled combustion occurred and soot was formed in less well-mixed zones.

Therefore, concerning the above discussion, some prominent issues can be noted regarding soot/PM emission for LTC conditions. They are following:

- PM/soot emission increases with the increase of EGR rate, advancement or retardation of injection timing to achieve LTC mode, until the combustion temperature is above the formation temperature of soot.
- Under soot formation temperature, soot emission does not depend on local equivalence ratio.
- Narrow angle of injection is conducive to lower soot emission during late injection premixed LTC.
- Although very advanced injection can create premixed charge in HCCI mode, deposition of fuel can increase PM/soot emission.
- Very late injection during premixed LTC creates fuel rich pockets which can increase soot emission.

### 3.2.3. PM emission under LTC modes for biodiesels

Though some of the researchers have reported increased PM emissions for biodiesel time to time [127–129], it is almost unanimous that biodiesel reduces PM emission significantly [130–134]. Researchers claiming increased PM emission, have a common explanation that, reduction of the soluble fraction (ISF) of the PM is compensated by the increase of soluble organic fraction (SOF) which increases with the use of biodiesel [101,135–137]. Main reasons for reduced PM emission for biodiesel than diesel can be summarized as, increased oxygen content, lower stoichiometric need of air, absence of aromatics and sulfur, combustion advance and soot structure formed while using biodiesel [94]. Higher oxygen content of biodiesel molecule ensures complete combustion even on the fuel-rich zones and reduces PM emission [138]. Possibility of fuel-rich zones reduces as the stoichiometric need of air is less for biodiesel. On the other hand advanced combustion elongates the residence time which confirms better oxidation of soot particles hence reduces the emissions [139,140]. Structure of soot particles of biodiesel also have been reported as a reducing factor of PM as it helps in oxidation of soot [141]. Aromatics, considered as the precursors of soot, reduce PM emission by its absence in biodiesel [85].

Though biodiesel reduces PM emission, LTC with biodiesel reduces PM emission even in a better form as it reduces simultaneously NOx and PM. Veltman et al. [103] reported higher injection pressure reduced PM emission when they achieved late injection premixed LTC. They commented that as higher injection pressure had less significance on NOx emission at higher EGR, it could be used to achieve simultaneous reduction of PM and NOx. Zheng et al. [112] also reported that higher injection pressure, boost pressure, multi-pulse injection with higher EGR reduced PM and NOx simultaneously. To explain higher emission of soot with EGR, they emphasized on not achieving the threshold temperature of soot formation as EGR was not increased to that level. Later, the same authors [108] experimentally proved the previous explanation by increasing EGR, to much higher level, and got reduced soot emission. They also attempted retarded multiple injections with higher EGR and got much reduced emission. In a separate experiment, Zheng et al. [105] showed multiple injection reduced soot emission even in higher loads while other authors [113] claimed higher PM emission for higher loads in premixed LTC. Fang et al. [107] reported that retarding injection timing until after the TDC was a potential way for simultaneous reduction of soot and NOx. In a separate experiment [142] they again claimed post-TDC injection provided ultra-low soot. They attributed this to the combination of low soot formation characteristics of biodiesel and low temperature combustion feature of the retarded post-TDC injection strategy. Weall and Collings [104] also reported very decreased amount of smoke emission as they retarded injection to the TDC. However, opposite findings are also there about PM emission during premixed LTC. Northrop et al. [115] reported over an order of higher magnitude of PM emission than diesel fuel while they experimented premixed LTC with biodiesel. Though the engine out soot was lower for biodiesel, they mentioned that PM emission increased due to the conversion of unreacted biodiesel to PM through condensation. In a separate experiment of partially premixed LTC with early and late injection, Northrop et al. [143] again found very high emission of PM due to the same factor.

Along with premixed LTC, HCCI combustion can reduce PM emission to a satisfactory level. Mancaruso and Vaglieco [144] reported very low amount of PM when they achieved HCCI combustion with rapeseed methyl ester. Homogeneous lean charge in the combustion chamber helped to reduce PM. Moreover, higher injection pressure facilitated the atomization of fuel and higher oxygen content of biodiesel ensured complete oxidation of soot. They attributed such reduction to less soot formation and enhanced rate of oxidation. Jiménez-Espadafor et al. [110] corroborated this finding in their article. However, overall emission was quite low and pilot-ignited HCCI gave them very low amount of soot. Zhu et al. [106] explored the potential of very low soot emission by using ethanol-biodiesel blend with late injection HCCI strategy. However, advanced injection timing can also reduce PM emission but it depends on achieving HCCI like combustion mode [144]. Ganesan et al. [113] reported, along with early and late fuel injection, port fuel injection with a fuel vaporizer reduced PM emission because of absence of diffusion combustion and localized fuel rich mixture.

Nevertheless, salient points of PM emission during LTC of biodiesel are as follows:

- Inherent properties of biodiesel, like increased oxygen content, lower stoichiometric need of air, absence of aromatics and sulfur, combustion advance and soot structure spontaneously reduces PM emission irrespective of conventional or low temperature combustion.
- Higher injection pressure, retarding injection timing, multi-pulse injection with higher boost pressure along with higher level of EGR are instrumental factors of PM reduction in premixed LTC for biodiesel.
- Due to conversion of unreacted biodiesel into PM through condensation, premixed LTC can increase PM emission time to time.
- HCCI combustion is very efficient in PM reduction for biodiesels as it permits higher injection pressure and EGR. Consequently, ignites homogeneous mixture.

### 3.3. UHC and CO emissions analysis

#### 3.3.1. UHC and CO formation

In conventional diesel combustion the main reasons for HC emission are trapping of fuel in the crevice volumes of the
combustion chamber, low temperature bulk quenching of the oxidation reactions, locally over-lean or over-rich mixture, liquid wall films for excessive spray impingement and incomplete evaporation of the fuel [67]. Turns [65] described two ways to form CO and UHC, viz. overly lean mixture and overly rich mixture. In the case of overly lean mixture, flame cannot propagate through the mixture and fuel pyrolysis with partial oxidation causes CO and UHC. For overly rich mixture, fuel cannot mix with sufficient amount of air, or even mix, does not get sufficient amount of time to get oxidized. This results in significant amount of CO and UHC.

3.3.2. UHC and CO emissions under LTC modes for diesel

In LTC regime, to reduce the locally fuel-rich region, intake charge and fuel are mixed more thoroughly before combustion. With reduced combustion temperature, though this mode helps to provide simultaneous reduction of NOx and soot [16,20,145] on contrary due to the reduction of in-cylinder combustion temperature and oxygen concentration, incomplete combustion products like HC and CO increase. Actually, this is one of the primary challenges of applying HCCI, PCCI or PPCI. Homogeneous or partially homogeneous mixtures are formed in these combustion processes and a significant amount of fuel is stored in the crevices at the time of compression and escape combustion. Since, the burned gas temperature is not that much high in these processes to consume the fuel, when they are back into the cylinder during expansion, CO and UHC formation becomes inevitable.

Han et al. [55] experimented with increased EGR and late injection timing assisted premixed LTC mode. Long ignition delay over mixed the air-fuel and in some regions the charge became too lean to burn which caused unburned fuel. Actually, when the injection occurs just before the TDC, lower peak bulk temperature hinders the oxidation process of the fuel. Because in that case, combustion occurs after TDC, when with every crank angle, the contents of cylinder are getting cooler for expansion. Therefore, for late injection LTC, bulk quenching mechanism and over mixing of the charge are responsible for HC and CO emissions. Another experiment was conducted by Han et al. [146] confirming the effect of ignition delay on HC emission. They experimented with diesel and gasoline blends. Greater percentage of gasoline in the fuel blend extended the ignition delay. Therefore, they observed that on same equivalence ratio, HC emission increased with the higher percentage of gasoline in the blend. This one is a clear indication of the effect of ignition delay on HC emission. However, they also observed that unlike HC, CO emission was primarily dependent on equivalence ratio. As the equivalence ratio was higher, they got higher CO emission even for the same fuel. Kook et al. [147] reported that CO emission showed a rapid decrease from the maximum as SOI was advanced, particularly at the highest swirl ratios. The numerical simulations showed, at a fixed swirl ratio, earlier injection timing enhanced pre-combustion mixing hence lower peak in-cylinder CO mass. The enhanced mixing was not only for increased ignition delay, but also for increased mixing rates under high-swirl conditions. A reduction in CO emission with increased pre-mixing implies that, CO emission stems predominantly from under-mixed fuel (rich mixtures), a finding which is supported experimentally by a strong tendency towards reduced CO emission with increased ignition delay and injection pressure.

Opat et al. [148] reported that mixing time effect or temperature effect does not control the HC and CO emissions entirely. They mentioned about fuel impingement in the piston bowl. Numerous researchers have accused liquid fuel film on piston bowl, generated by spray impingement as the primary reason for higher HC emission in early injection premixed LTC [53,149–153]. As the injection advances, the fuel spray trajectory crosses the piston bowl surface at a higher point, causes more deposition consequently higher CO and UHC. But wall impingement can be avoided by some changes with the injection angle. Genzale et al. [124] showed the solution which is mentioned earlier. However, Kim and Lee [35] investigated the impact of injection with narrow fuel spray angle and a dual injection idea on the emission characteristics. They tuned the first and second injection timing with a narrow fuel spray angle (injection angle 60°) and compared the results with conventional injection parameters. Dual fuel injection strategy, consisting an early injection (50° BTDC) and a late second injection (20° ATDC) with narrow angle injection gave them reduced CO and HC emission. Again under the LTC mode, higher load emits lower HC and CO. Because higher load means higher peak bulk temperature and it helps in oxidation and hence complete combustion of the fuel. Alriksson and Denbratt [20] got an experimental proof regarding this.

Besides experimenting about the reasons of HC and CO emissions, some researchers have worked out the regions of HC and CO generation during premixed LTC. Ekoto et al. [151] worked out the sources of UHC and CO applying PPCI in a light duty diesel engine with low load and engine speed. UHC and CO emissions are most significant under low load and speed while the engine is running in the LTC condition. They spotted out three regions where UHC and CO originate primarily in the PPCI strategy. At centerline and squish-volume, the got experimental evidence but at bowl and central clearance volume they only predicted by simulation. Kim et al. [152] tried to reveal the UHC and CO emissions regions in the cylinder under premixed LTC. They came to a decision that during expansion stroke the spatial distribution of UHC in the clearance volume was dominated by a region near the cylinder centerline and by a region near the cylinder wall, the latter likely due to UHC released from the top ring-land crevice. When injection timing is advanced or retarded, UHC extends throughout the squish volume, due to formation of over-rich or over-lean mixtures, respectively. At light load, UHC mainly generates at the centerline from lean mixture. Increased dilution slows down oxidation throughout the cylinder, but most noticeably within the squish volume. On the other hand, CO is generally observed near the cylinder centerline and broadly distributed within the squish volume. Advanced injection timing increases squish volume CO; retarded injection also does so but later in the cycle. Increased dilution generally increases CO throughout the cylinder, but especially within the squish region, just like UHC. Musculus et al. [154] observed that when end of injection (EOI) was shorter than ignition delay, UHC and CO were increased significantly. Fuel vapor measurement analysis revealed that maximum portion of HC emission came from the incomplete combustion of lean mixture near the injector after EOI. Ekoto et al. [151] claimed that late-cycle addition of partially oxidized liquid and vapor phase fuel through nozzle dribble or ejected fuel droplets were found to be a major contributor of UHC and CO emissions in the PPCI regime. Singh et al. [14] also observed, near the injector the mixture did not undergo complete combustion which resulted in higher UHC emission during HCCI like combustion mode. It is not enough to mention that, HCCI combustion strategies are also have been reported for higher HC and CO emissions [35,77]. Basic principles of higher emission of HC and CO are more or less similar for PPCI, HCCI or RCCI combustion modes which are already discussed here. However, some striking points about UHC and CO emissions under LTC modes can be noted here.

- Reduction in in-cylinder combustion temperature and oxygen concentration are the main reasons for higher HC and CO emissions for LTC modes.
- Higher ignition delay of LTC modes over mixes the charge. Thus it becomes too lean to burn, consequently HC and CO emissions increase.
Advanced injection for both PCCI and HCCI causes spray impingement on piston bowl which results in higher emission of HC and CO.

HC emission primarily depends on ignition delay whereas CO emission mostly depends on equivalence ratio.

3.3.3. UHC and CO emissions under LTC modes for biodiesels

Biodiesel with conventional diesel combustion process reduces UHC and CO emissions significantly [99,155–161]. The most important reason behind is the oxygen content and the cetane number. As the oxygen content is higher in biodiesel, it helps attaining complete oxidation and comparatively complete and efficient combustion than diesel thus reduces UHC and CO [162]. Higher cetane number reduces the ignition delay that means advances the combustion and also decreases the possibility of fuel-rich zones [99] which reduces the UHC and CO emissions. Therefore, using biodiesel in LTC mode can be an attractive way to mitigate the obvious increment of UHC and CO [163] which are general effect of LTC described earlier. In spite of this, some researchers have mentioned higher UHC and CO emissions for biodiesel than diesel fuel under LTC mode [110]. They attributed it to poor air–fuel mixture formation of biofuels.

However, during premixed LTC, Tormos et al. [164] reported 27% and 45% decrement of CO and UHC respectively for biodiesel than diesel. It establishes, during premixed LTC, biodiesel is better than diesel fuel regarding UHC and CO emissions. But within the biodiesels, at LTC mode UHC and CO emissions depend on several factors like injection pressure and timing [104], operating load, injection style [112], intake air temperature [165], in-cylinder temperature and combustion phasing [163], etc. Besides these factors, Petersen et al. [163] observed short premixed combustion duration of biodiesel resulted in low UHC and CO emissions which was stated earlier by Northrop et al. [109]. They mentioned addition of biofuels resulted in shorter combustion duration, consequent of higher ignition quality, which reduced UHC and CO emissions. In a separate experiment of premixed combustion of biodiesel, Northrop et al. [115] again mentioned shorter ignition delay for reduced UHC and CO emissions.

However, in premixed LTC mode Northrop et al. [109] also reported higher emission of UHC and CO at late injection. They attributed this to incomplete combustion and fuel passing unreacted through the cycle. Later on, some authors [142] reported that in the premixed mode, both early and late injection emitted higher UHC and CO than conventional combustion of biodiesel. Extended ignition delay of LTC mode created over-lean regions and thus increased the quantity of injected fuel species outside lean flammability limits hence increased UHC. It implies that, though biodiesel reduces UHC and CO at premixed LTC than diesel fuel, still its higher than conventional combustion of biodiesel. Opposite results are also there. Zheng et al. [105] reported very low UHC and CO emissions at late injection premixed LTC for low load.

Researchers have also tried multiple injections and got comparatively reduced UHC and CO emissions than conventional single shot injection [105,108,112]. However, at multiple injections, very advanced SOI showed increased UHC and CO emission due to poor mixing of fuel–air [105]. Same result was achieved by Veltman et al. [103] at advanced pilot injection. Along with SOI, injection pressure has a significant effect on UHC and CO emissions [103,104]. Veltman et al. [103] got reduced UHC and CO emissions as they increased the injection pressure to 180 MPa from 150 MPa on premixed LTC.

However, regarding UHC and CO emissions, premixed LTC and HCCI give more or less same trends for biodiesel. Still there are some evidences where researchers have got lower UHC and CO for HCCI. Mancaruso et al. [144] observed quite low UHC for biodiesel than diesel fuel while they experimented HCCI mode and attributed this reduction to the increased oxygen content. Ganesan et al. [113] got reduced emission for higher loads during HCCI. Bunting et al. [165] reported increased intake air temperature helped to reduce the UHC and CO emissions as it advanced the combustion phasing of HCCI combustion mode with biodiesel.

Some salient points regarding this discussion are as follows:

- LTC of biodiesel reduces UHC and CO than LTC of diesel, but still gives higher emission than conventional combustion of biodiesel.
- Both late and early injection timing of biodiesel give higher emission in premixed LTC than conventional combustion.
- Comparative reduction of UHC and CO emissions during premixed LTC of biodiesel than diesel can be attributed to the short premixed combustion duration and higher oxygen content of biodiesel.
- Higher injection pressure, intake pressure or multiple injection methods are conducive to reduce UHC and CO emissions during biodiesel combustion both for PCCI and HCCI.
- HCCI mode emits higher UHC and CO when the mixture is over mixed and beyond the lean flammability zone.

4. Performance analysis under LTC

Though LTC modes reduce NOx and PM simultaneously, unfortunately it is accompanied by higher fuel consumption and lower thermal efficiency for both diesel and biodiesels [20,21,73,109,110]. Tables 3 and 4 summarize the results respectively for diesel and biodiesels. Considerable increase of unburned fuel (in the form of unburned HC and CO in the exhaust) at higher EGR percentage or retarded injection timing to achieve LTC are responsible for this. Not only in the case of late injection for PCCI like combustion but also at advanced injection timing for HCCI like combustion, fuel consumption increases drastically. Poor evaporation, formation of the air–fuel mixture outside the combustion chamber or too much lean charge to burn due to over mixing are the issues responsible for such increment of fuel consumption [35]. Though biodiesels have got higher oxygen content which permit a higher percentage of EGR keeping the CO and HC lower, many researchers have reported higher fuel consumption and lower thermal efficiency when they used biodiesel in LTC modes [105,108,109,111,113]. However, some exceptions have also been reported [107,167].

Researchers have tried many ideas to reduce the fuel consumption during LTC. Kim and Lee [35] reported lower fuel consumption applying the narrow fuel injection system in HCCI combustion. Fang et al. [142] reported lower BSFC for all the tested fuels in a HCCI like combustion process when they tried injection after TDC. However, during PCCI combustion process, most of the researchers have reported increased fuel consumption due to late injection timing (primary requirement for premixed combustion process) [68,74,108,109] whereas, a few researchers have got a bit different result. Lilik and Boehman [5] reported lower fuel consumption even in late injection when they tried a synthetic fuel produced in a low temperature Fischer–Tropsch process. They explained this reduction by the combustion phasing of this high reactive fuel (CN: 81), which was maintained near the TDC. Kook et al. [16] reported maximum fuel conversion efficiency at moderate EGR and slight late injection timing. Fang et al. [107] reported lower fuel consumption for biodiesel blends as they retarded injection timing. Zhang and Boehman [167] reported low fuel consumption even at higher EGR level at 6° BTDC injection timing. Hence, proper...
Table 3  
Performance for diesel at LTC.

<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>Power/torque/IMEP</th>
<th>BSFC/ISFC</th>
<th>BTE/ITE</th>
<th>Refs.</th>
</tr>
</thead>
</table>
| 4S,4-cylinder, DI, TC  
DV: 4.5 L  
CR: 16.5:1  
RS: 2400 rpm  
RP: 115 kW@2400 rpm  
Injection system: common rail | Retarded injection  
LTC  
Speed: 1400 rpm  
Variable torque: 54–80 N m | D | Sweep of injection timing  
–8° to –2° ATDC | 56% | Torque ↓ as the IT retarded | ↑ as the IT retarded | N/A | [73] |
| 4S,1-cylinder, DI, super charged,  
DV: 2.022 L  
RS: 1500 rpm  
CR: 14:1  
Injection system: electronically controlled injector | LTC  
25% load, charge air pressure: 1.3 bar (abs.)  
50% load, charge air pressure: 2.4 bar (abs.) | D | Sweep of IT 9° to 20° BTDC  
Sweep of IT 9° to 20° BTDC | Up to 60%  
Up to 65% | N/A | Value ↑ as the IT retarded after 50% EGR  
Advanced IT like 20° BTDC gave notable ↓ value | N/A | [20] |
| 4S,1-cylinder, super charged, DI, WC  
DV: 781.7 cm³  
CR: 13  
RS: 1000 rpm  
Injection system: common rail | PCCI  
Injection pressure: 140 MPa | Low sulfur diesel | Sweep of IT 15 to 25° BTDC | 0% and 40%  
40% EGR gave higher IMEP than 0% EGR | N/A | 40% EGR gave ↑ value than 0% EGR | N/A | [74] |
| 4S,1-cylinder, DI  
DV: 373 cm³  
CR: 15:1  
RS: 1500 rpm  
IP: 100 MPa  
Injection system: Bosch common rail | HCCI  
Injection angle: 60° | D | Sweep of injection from 40° to 70° BTDC | N/A | Narrow angle injection gave higher IMEP than conventional injection  
Narrow angle injection gave lower ISFC than conventional injection all through | N/A | [35] |
| 4S,4-cylinder, DI, TC  
DV: 2.5 L  
CR: 17.5  
RS: 4000 rpm  
RP: 103 kW@4000 rpm  
Injection system: Bosch common rail | Advanced PCCI  
Fuel processed by high and low temperature Fischer–Tropsch process (HTFT and LTFT) | D | Sweep of injection  
–8° to 0° ATDC | 40% | N/A | ↑ as IT retarded,  
LTFT gave lowest BSFC | LTFT gave ↑ value at late IT | [5] |
| 4S,4-cylinder, DI, WC, TC  
DV: 1910 cm³  
CR: 17.5:1  
BMEP: 0.8 MPa  
Variable IP: 100–160 MPa  
Injection cone angle: 148° | PPCI | Low sulfur diesel,  
20% and 40% blend of n-butanol with diesel | Retarded IT  
Sweep from to 2° ATDC | O₂ concentration 19.5% | N/A | For each condition 5–7% higher value  
Lower value when LTC achieved. | N/A | [68] |

(continued on next page)
<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>Power/torque/IMEP</th>
<th>BSFC/ISFC</th>
<th>BTE/ITE</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S, 4-cylinder, DI DV: 2198 cc RS: 3500 rpm RP: 96 kW CR: 16.6:1 Injection system: common rail Injection cone angle: 153°</td>
<td>Optimized PPCI 1800 rpm IMEP: 2.93</td>
<td>50% blend of gasoline with D</td>
<td>Advanced injection at 28° BTDC</td>
<td>50%</td>
<td>N/A</td>
<td>N/A</td>
<td>2.5% lower value than CC [21]</td>
<td></td>
</tr>
<tr>
<td>4S, 1-cylinder, WC DV: 422 cm³ CR: 18.7:1 RS: 1500 rpm IP: up to 1350 bar Injection system: common rail</td>
<td>Late injection premixed LTC</td>
<td>Diesel</td>
<td>Sweep from −30.25° to 7.75° ATDC</td>
<td>Up to 65%</td>
<td>Power ↓ as EGR ↑</td>
<td>N/A</td>
<td>Maximum at moderate EGR and slight late IT (−5° ATDC) [16]</td>
<td></td>
</tr>
<tr>
<td>4S, 1-cylinder DV: 2.44 L CR: 16.1:1 IMEP: 9 bar Direct IP: 400 bar Port IP: 5.17 bar</td>
<td>RCCI</td>
<td>Port injected fuel: gasoline Direct injected fuel: gasoline + variable percentage of DTBP (di-tert-butyl peroxide)</td>
<td>N/A</td>
<td>41%</td>
<td>IMEP: 6–9 bar</td>
<td>N/A</td>
<td>57% at 3.5% DTBP in gasoline [78]</td>
<td></td>
</tr>
<tr>
<td>Engine setup</td>
<td>Operating condition</td>
<td>Fuel</td>
<td>Injection timing</td>
<td>Percentage of EGR/O₂ concentration</td>
<td>Power/torque</td>
<td>BSFC/ISFC</td>
<td>BTE/ITE</td>
<td>Refs.</td>
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<tr>
<td>4S, 1-cylinder, DI DV: 708 cm³ CR: 18.4:1 RP: 11 kW@3000 rpm RT: 45 N m@2100 rpm IP: 650 bar Injection system: pump injection</td>
<td>HCCI</td>
<td>Diesel, 100%, 65% and 30% blend of colza biodiesel</td>
<td>10⁰ BTDC</td>
<td>Up to 32%</td>
<td>N/A</td>
<td>BSFC ↑ with the percentage of EGR</td>
<td>N/A</td>
<td>[110]</td>
</tr>
<tr>
<td>4S, 1-cylinder, DI DV: 857 cm³ CR: 17.8:1 RP: 12.5 kW@2400 rpm IP: 4 bar Injection system: injection pump</td>
<td>High EGR enabled HCCI Variable loading</td>
<td>100% soy biodiesel</td>
<td>17⁰ BTDC</td>
<td>Up to 60%</td>
<td>Notable power loss with the increment of EGR</td>
<td>N/A</td>
<td>↓ BTE with increased EGR</td>
<td>[111]</td>
</tr>
<tr>
<td>4S, 4-cylinder, DI DV: 1998 cm³ CR: 18.2:1 RS: 1500 rpm IP: 950 bar Injection system: common rail</td>
<td>Variable loading (IMEP 5–10 bar). Variable boost pressure.</td>
<td>Biodiesel blend of Soy, Canola, Yellow grease and Tallow biodiesel.</td>
<td>Single and multiple injections with wide range sweep of IT.</td>
<td>Up to 70% according to the condition.</td>
<td>Power ↓ as% of EGR ↓ Single shot injection gave better power than multiple injection.</td>
<td>N/A</td>
<td>N/A</td>
<td>[105]</td>
</tr>
<tr>
<td>4S, 4-cylinder, DI DV: 1998 cm³ CR: 18.2:1 RS: 1500 rpm IP: 950 bar Injection system: common rail</td>
<td>EGR and late injection assisted LTC mode Variable intake pressure (1.2/1.5 bar)</td>
<td>Biodiesel blend of Soy, Canola, Yellow grease and Tallow biodiesel.</td>
<td>Single shot injection (IMEP 8 bar) Multi-pulse injection (IMEP 6 bar) Wide range (347–367°C) sweep of IT</td>
<td>Up to 70%</td>
<td>N/A</td>
<td>Above 50% EGR and at late IT very high fuel consumption. Multiple injection gave ↓ value than single shot injection.</td>
<td>N/A</td>
<td>[108]</td>
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<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>Power/torque</th>
<th>BSFC/ISFC</th>
<th>BTE/ITE</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>4S,4-cylinder, DI DV: 1.7 L CR: 16:1 RS: 1500 rpm Load: 400 kPa IP: 800, 1000, 1200 bar Injection system: common rail</td>
<td>Late injection premixed LTC mode</td>
<td>20%, 50% and 100% Soy-based methyl ester</td>
<td>5°, 7° and 9° BTDC</td>
<td>50%</td>
<td>Retarded IT gave ↑ BSFC ↓ BTE at retarded IT</td>
<td>[109]</td>
<td></td>
<td></td>
</tr>
<tr>
<td>4S,4-cylinder, DI, TC DV: 1.7 L CR: 16:1 RS: 1500 rpm RP: 75 kW@4400 rpm IP: on average 870 bar Injection system: common rail</td>
<td>Late and early injection partially premixed LTC</td>
<td>100% soy based methyl ester and 50% blend of soy based methyl ester with ULSD</td>
<td>5.9° to 7.1° BTDC (for LLTC) 17.3° to 24.1° BTDC (for ELTC)</td>
<td>45%(LLTC) 55%(ELTC)</td>
<td>Lower power for LLTC and ELTC than CC. ELTC gave lowest.</td>
<td>N/A</td>
<td>N/A</td>
<td>[143]</td>
</tr>
<tr>
<td>4S,1-cylinder, DI DV: 300 cc CR: 19.5:1 RS: 1500 rpm IP: 600 bar Injection system: common rail</td>
<td>Late injection LTC</td>
<td>20%, 50% and 100% soy biodiesel</td>
<td>Sweep of injection timing from – 25° ATDC to 3° ATDC</td>
<td>N/A</td>
<td>N/A</td>
<td>ISFC ↓ up to 11.1% as the IT was retarded.</td>
<td>N/A</td>
<td>[107]</td>
</tr>
<tr>
<td>4S,4-cylinder, DI, TC DV: 2.5 L CR: 17.5:1 RP: 103 kW@4000 rpm IP: up to 1600 bar Injection system: common rail</td>
<td>Late injection EGR assisted LTC 1600 rpm and 25% loading</td>
<td>40% biodiesel blended with ultralow sulfur diesel</td>
<td>Single injection (6° BTDC) Double injection (pilot: 25° BTDC, main: 2° ATDC)</td>
<td>Up to 38%</td>
<td>N/A</td>
<td>Reduced BSFC at 6° BTDC IT even at higher EGR. 6° BTDC IT gave high thermal efficiency even at higher EGR.</td>
<td>[167]</td>
<td></td>
</tr>
</tbody>
</table>
Table 4 (continued)

<table>
<thead>
<tr>
<th>Engine setup</th>
<th>Operating condition</th>
<th>Fuel</th>
<th>Injection timing</th>
<th>Percentage of EGR/O₂ concentration</th>
<th>Power/torque</th>
<th>BSFC/ISFC</th>
<th>BTE/ITE</th>
<th>Refs.</th>
</tr>
</thead>
<tbody>
<tr>
<td>45,4-cylinder, DI DV: 425 cm³ (1-cylinder) CR: 15:1 RS: 1500 rpm IP: 100 MPa Intake pressure: 120/150 kPa Injection system: common rail</td>
<td>Late injection premixed LTC Variable loading</td>
<td>Biodiesel–ethanol (80–20%) blend.</td>
<td>13 to 10.5° BTDC</td>
<td>40% for high load Up to 50% for low loads.</td>
<td>IMEP range was 0.35–0.82 MPa</td>
<td>N/A</td>
<td>Maintained at least 96% combustion efficiency</td>
<td>[106]</td>
</tr>
<tr>
<td>45,1-cylinder, DI, AC DV: 662 cm³ CR: 17.5:1 RP: 4.4 kW RS: 1500 rpm IP: 2 bar Injection system: port fuel injection with fuel vaporizer, direct injection</td>
<td>Fuel vaporizer with port fuel injection assisted HCCI Variable loading</td>
<td>100% biodiesel</td>
<td>23° BTDC (for direct injection)</td>
<td>N/A</td>
<td>N/A</td>
<td>3–5% higher value than CC system</td>
<td>About 5.5% ↓ value than CC system</td>
<td>[113]</td>
</tr>
<tr>
<td>45,1-cylinder, DI DV: 300 cc CR: 19.5:1 RS: 1500 rpm IP: 600 bar Injection system: common rail Injection cone angle: 150°</td>
<td>Late injection HCCI</td>
<td>Neat soybean biodiesel and 20–50% blend of biodiesel with low sulfur diesel.</td>
<td>−25° ATDC, −10° ATDC and 3° ATDC</td>
<td>N/A</td>
<td>N/A</td>
<td>For all the fuels late injection (3° ATDC) gave lower value than even CC condition.</td>
<td>N/A</td>
<td>[142]</td>
</tr>
</tbody>
</table>
optimization of the operating conditions and fuel chemistry can keep the fuel consumption low even in the LTC modes.

However, regarding load level, LTC modes were always confined to the low to mid region as higher load means higher in-cylinder temperature which drives NOx higher. Fortunately, experiments regarding fuel reactivity have unwrapped higher load region for LTC strategies with higher efficiency. Bessonette et al. [168] reported 60% increment of the operating load during HCCI combustion while they experimented with a fuel having autoignition quality in between diesel and gasoline. Inagaki et al. [61] used dual fuel of different reactivity in PCCI combustion and succeeded to increase the IMEP up to 12 bar. Being inspired by these works Splitter et al. [62] worked regarding reactivity gradient of fuel inside cylinder during PCCI combustion and experimentally demonstrated that combustion proceeded from areas of locally high fuel reactivity to areas of locally low fuel reactivity. Such staged combustion process extended the duration of the premixed combustion event which resulted in high thermal efficiency, low pressure rise rate, for loads as high as 16 bar IMEP [63] keeping the emissions low. Later on, the same authors tried RCCI combustion using gasoline as the low reactive fuel and gasoline + di-tert-butyl peroxide as high reactive fuel. They got up to 57% indicated thermal efficiency [78]. Application of ethanol–gasoline blend as low reactive fuel gave even higher efficiency like 59% [19]. Nieman et al. [17] also reported higher efficiency with higher load level when they used natural gas as the low reactive fuel. Therefore, regarding performance, though lower power, higher fuel consumption with lower thermal efficiency are the inevitable effects of LTC modes, we have some potential methods as well to avoid these drawbacks. Low emission level with permeable fuel consumption and moderate load level are achievable for HCCI, PCCI and obviously for RCCI combustion processes.

Some highlighting points of the above discussion are as follows:

- Unburned fuel (in the form of HC and CO in the exhaust) due to higher EGR rate or late injection timing of premixed LTC mode, is responsible for higher fuel consumption.
- In the case of advanced injection during HCCI like combustion, over mixing or poor evaporation and sometimes formation of charge outside the combustion chamber are responsible for higher fuel consumption.
- Proper optimization of the combustion conditions and fuel chemistry can reduce the fuel consumption of LTC modes.
- Fuel reactivity stratification method used in RCCI combustion has unfastened a new era of LTC by reducing BSFC as well as increasing the operating load with high thermal efficiency.

5. Uncovered gaps and probable solution of the drawbacks of LTC

It is not enough to mention that, to meet the future emission regulations, along with the after-treatment methods, in-cylinder emission reduction technologies like LTC modes are of immense necessity. Though researchers are trying to improve LTC modes day by day, still there are some drawbacks or some problems of LTC modes which need to be solved. New ideas or methods should come to light regarding the improvement of this marvelous idea of low emitting combustion system. Some points are noted below to highlight the uncovered gaps and probable solutions regarding this system:

- Higher emission of UHC and CO is a huge drawback of this system. Use of oxygenated fuels like ethanol, n-butanol, diethyl ether or biodiesel can be a good solution to this problem. There is a huge gap of research using such fuels during LTC modes. Compositional differences between biofuels are responsible for a change in CO oxidation behavior especially at late injection. These issues should be explored visibly to make oxygenated fuels more suitable for LTC modes.
- In the case of late injected PCCI like combustion, degree of atomization and mixture quality are vital issues for CO and HC emissions. High injection pressure with in-cylinder swirl formation can be used to mitigate these emissions.
- Use of fuel vaporizer has made a new development for using heavy fuel like biodiesel in HCCI like combustion. However, research should be conducted thoroughly to improve its efficiency level for different biodiesels.
- New concepts should come to increase the operating load level of PCCI and HCCI.
- RCCI combustion mode has not been tested with biodiesel as the higher reactive fuel yet. Different fuels should be tested covering a wide range of reactivity. It may bring some new concepts for better emission characteristics.

6. Conclusion

Diesel engines are significant power sources for numerous applications because of stability, fuel flexibility, and higher thermodynamic efficiency. Over the past two decades, diesel engine pollutant emission regulations have progressively become more stringent. To cope with the order-of-magnitude reductions of both PM and NOx, required by upcoming regulations, engine developers will have to rely on alternative in-cylinder strategies that use some form of LTC. Compared to conventional diesel combustion, LTC strategies generally employ enhanced pre-combustion mixing, which helps to avoid locally rich regions. Less locally fuel rich region means low PM formation. Similarly, using EGR results in dilution by pre-combustion mixing which reduces the peak combustion temperature. The peak combustion temperature at LTC can be reduced to about 2000 K or even lower, which drastically reduces the thermal NOx formation. Engine researchers and developers have proposed several practical strategies of LTC like HCCI, PCCI, PCI, and RCCI. Researchers have attained these strategies using both diesel and biodiesels. As the objective of this article was to analyze the effects of LTC strategies on emissions for diesel and biodiesels, from review, the following points are summarized.

- LTC mode visibly reduces NOx emission. Irrespective of combustion strategies, low NOx emission basically depends on higher ignition delay and lower combustion rate, which result in lower in-cylinder temperature. Higher use of EGR and optimized SOI control these parameters on LTC to keep the NOx emission low during diesel combustion. LTC modes have also succeeded to reduce the usual higher emission of NOx for biodiesels. Higher percentage of oxygen and cetane number of biodiesels enable them to sustain higher EGR and late SOI to attain better LTC. Therefore, better NOx reduction. Regarding combustion modes, irrespective of fuels, premixed LTC reduces NOx better than HCCI. However, considering efficiency, RCCI is the best one.
- LTC mode simultaneously reduces NOx and PM emissions. Generally LTC modes take the combustion temperature below the formation temperature of PM. Below the formation temperature even the equivalence ratio has no command on PM emissions. Consequently, PM emissions decrease for diesel fuel combustion during LTC modes. For biodiesels, condensation of unburned fuel sometimes increases PM emissions during LTC modes. However, generally, inherent properties of biodiesels like increased oxygen content, lower stoichiometric need of
air, absence of aromatics and sulfur, combustion advance and soot structure spontaneously reduce PM emissions during all the LTC strategies.

- UHC and CO emissions increase during LTC of diesel for the reason of reduction in in-cylinder combustion temperature and oxygen concentration. Biodiesels reduce UHC and CO emissions than diesel during LTC modes because of their short premixed combustion duration and higher oxygen content. Still, the emission level remains higher than the conventional combustion system.

- Generally, LTC strategies show higher fuel consumption. Unburned fuel (in the form of HC and CO in the exhaust) due to higher EGR rate or late injection timing is responsible for higher fuel consumption. Proper optimization of injection timing and fuel chemistry can reduce it to a satisfactory level.

Since, LTC method is an emerging idea of modern combustion science, new ideas or strategies should come to light to eliminate or decrease its inherent drawbacks. Using oxygenated fuels or additives, high injection pressure with in-cylinder swirl formation, optimized use of fuel vaporizer and finally, use of different fuels having a wide range of reactivity for the RCCI combustion system can be potential fields for future studies. However, taking stringent emission policies into concern and continuous improvement regarding the performance of LTC, it is sure that LTC is on the verge of being the most suitable and acceptable technology of future combustion.

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