



Review article

A landscape review on biodiesel combustion strategies to reduce emission

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ABSTRACT

The study presents state-of-the-art literature on biodiesel combustion strategies for improving engine performance and reducing emissions. The application of those strategies in compression ignition engines is a prominent approach to reducing harmful emissions. The literature reveals that biodiesel can significantly reduce CO, HC, and PM emissions. However, the higher NO_x emission characteristics of biodiesel compared to fossil fuel is a concerning factor for its application. In addition, its physicochemical properties have also shown some complications during the combustion process. This article covers fundamental aspects, challenges, prospects, and the pros and cons of advanced biodiesel combustion strategies such as low-temperature combustion (LTC), chamber modifications, and injection modifications that directly influence combustion rate and emission formation. In addition, the exhaust gas after-treatment systems and the effect of operating parameters are briefly discussed and presented in this article. The study found that LTC strategies and retarded injection timing showed significant reductions in NO_x emission in the leaner mixtures but with poor combustion control. Besides, chamber modifications and higher injection pressure showed better combustion through high turbulence and swirl rates, but leading to more NO_x emission. The review critically analyzed those strategies and concluded that incorporating leaner mixture combustion with chamber modifications will form better combustion under low in-cylinder temperature conditions, favoring reducing the NO_x emission. Finally, the study recommends that more studies are needed to obtain a low in-cylinder temperature atmosphere by altering the equivalence ratio and optimizing the chamber (geometry and nozzle) modifications and varying injection parameters with biodiesel to reduce emissions.

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1. Introduction

Diesel engines play a crucial aspect in our daily lives in the industrial, agricultural, and transportation sectors. A compression ignition (CI) engine is effective in the transportation sector in terms of engine performance, emission, and load-carrying capacities. Almost 95% of the world’s transportation depends on the petroleum-based fuels that run the internal combustion (IC) engines. The main detriments of the use of fossil fuels in CI engines are global warming and environmental pollution v. According to Our World Data (OWD), the energy sector is responsible for 73% of the total greenhouse gas (GHG) emissions (Ritchie, 2020). Under the energy sector, the transportation sector alone is responsible for 16.2% of the global GHG emissions (Ritchie, 2020), as shown in Fig. 1. Road transportation accounts for 11.9% of global GHG emissions in the transportation sector. According to the International Energy Agency (IEA), passenger and road freight vehicles release more CO₂ emissions than other transport sectors, as shown in Fig. 2. In 2019, for instance, around 8.5 Giga tons of CO₂ were released into the atmosphere. More robust policies are needed to support decarbonizing the transportation sector (IEA, 2022).

At the Glasgow climate pact, COP26, around 200 countries negotiated and agreed to limit the global temperature rise to 1.5 °C by mid-century (Pact, 2021; Sarker et al., 2023). In the case of transportation, for delivering these challenging commitments, several countries have agreed to an energy transition from fossil sources to electrification technologies that can commit to net zero emissions. These agreements have placed the diesel engine future in a tough place; however, carbon-neutral electrification has a long way to go in many countries. Optimistic scenarios are being analyzed, indicating that electric vehicles (EVs) might significantly cut GHG emissions in the transportation sector. If considering the case, most countries depend on electric energy generated by using fossil sources such as coal, natural gas and diesel, which generate significant GHG emissions (Azad et al., 2015a). Energy security is also crucial in developing and maintaining electric vehicles (Thomas et al., 2022). When considering the life cycle assessment of EV production, CO₂ emissions will be 59% higher than fossil-fueled internal combustion engines (Goel et al., 2021). This emission percentage difference will be more when compared with biodiesel fueled vehicles because biodiesel emits less GHG emissions than fossil fuels (Azad et al., 2015b). Considering these scenarios, biodiesel can compete with EVs and be a suitable replacement for fossil fuels, and its historical development over the years suggests that biodiesel can be a viable source of reducing GHG emissions (Azad et al., 2016b; Bhuiya

et al., 2016b). Like fossil fuels, biodiesel can run diesel engines without any modifications, but a few alterations to the working components can help achieve better engine characteristics with biodiesel (Azad et al., 2020, 2017, 2019; Karthickeyan et al., 2020; Verma et al., 2020).

This study supports the statement that biodiesel can help reduce GHG emissions to a greater extent than petroleum fuels. Moreover, several results showed improved engine efficiency with biodiesel compared to diesel fuel (Bhuiya et al., 2016a). Biodiesel’s physical properties and higher NO_x emissions are the major concerns associated with biodiesel research (Azad, 2017). Direct implementation of biodiesel in diesel engines is not a good idea as the physical properties such as higher density and viscosity of the biodiesel cause improper fuel mixings, injector choking, piston ring deposits, and many more difficulties. Despite all these, biodiesel can contribute a viable path to a sustainable energy source to meet transportation needs. According to reports from the international energy agency (IEA), biodiesel production and consumption trends are rising yearly (IEA, 2021), as presented in Fig. 3. Moreover, it is predicted that the production and consumption rates will be higher in the future (IEA, 2021). A category of technologies, including machine learning, chemical kinetic modeling, and several optimization tools, are being used to investigate and improve the chemical kinetics of biodiesel in internal combustion engines. Advanced exhaust gas treatment systems are also implemented in automobile vehicles that can minimize CO, HC, soot, and NO_x emissions (Doppalapudi et al., 2023). The European Environmental Agency has reported a decrease in GHG emissions between 2018 and 2019 because of the implementation of biodiesel in the transportation sector (agency, 2019). Several upcoming automobile vehicles in the market are meeting the stringent EURO 6 emission standards and these vehicles can limit the transportation sector pollutants remarkably. Hence, the primary motivating factor to present this study is to discuss the advanced combustion strategies for biodiesel fuels. Indeed, there are few existing review articles focusing on individual strategies, but this review shines a light over other review articles by presenting detailed outcomes on several combustion modification technologies with biodiesel. Indeed, there are some gaps in understanding the potential for the combination of these viable techniques with biodiesel fuel. The study also provides a brief review of the combination of these strategies.

The study is mainly focused on the advanced combustion strategies irrespective of the biodiesel type, and the organization of the report is as follows: Firstly, low-temperature combustion strategies are discussed and several viable techniques in reducing the NO_x emissions from biodiesel are mentioned. Secondly, the

List of abbreviations

| | |
|--------------------------------|--|
| AC | Activated carbon |
| Al ₂ O ₃ | Aluminum oxide |
| BHA | Butylated hydroxy anisole |
| BP | Break power |
| BSFC | Brake specific fuel consumption |
| BTDC | Before top dead center |
| BTE | Brake thermal efficiency |
| BHT | Butylated hydroxytoluene |
| CeO ₂ | Cerium oxide |
| CI | Compression ignition |
| CN | Cetane number |
| CNG | Compressed natural gas |
| CO | Carbon monoxide |
| CO ₂ | Carbon di-oxide |
| CR | Compression ratio |
| DOC | Diesel oxidation catalyst |
| DCC | Direct combustion chamber |
| DLDC | Double layer diverging combustion |
| DMF | Diesel methanol fuel |
| DPF | Diesel particulate filter |
| DSPB | Double swirl piston bowl |
| ED | Ethanol-diesel |
| EGR | Exhaust gas recirculation |
| EGT | Exhaust gas temperature |
| EHN | 2-ethylhexyl nitrate |
| FSCS | Forced swirl combustion system |
| GD | Gasoline- diesel |
| GE | Geraniol |
| HC | Hydrocarbons |
| HCC | Hemispherical Combustion Chamber |
| HCCI | Homogeneous charge compression ignition |
| HRF | High reactivity fuel |
| HRR | Heat release rate |
| ICP | In-cylinder pressure |
| ID | Ignition delay |
| LHR | Low heat rejection engine |
| LRF | Low reactivity fuel |
| LTC | Low temperature conditions |
| N ₂ O | Nitrous oxide |
| NO ₂ | Nitrogen di-oxide |
| NO | Nitrogen oxide |
| NO _x | Nitrogen oxides |
| OEC | Oxygen enriched combustion process |
| PCCI | Premixed charge compression ignition |
| PFI | Port fuel injection |
| PMDI | Porous medium direct injection system |
| PPPD | P-phenylene di amine |
| PRR | Pressure rise rate |
| PM | Particulate emissions |
| PPCC | Partially premixed charge compression |
| PRR | Pressure rise rate |
| PY | Pyrogallol |
| RCCI | Reactivity controlled compression ignition |

| | |
|-------------------------------|--|
| SCC | Shallow depth combustion chamber |
| SCCI | Stratified charge compression ignition |
| SCR | Selective catalytic reduction |
| SI | Spark ignition |
| SOF | Soluble organic fraction |
| SOI | Start of injection |
| SO _x | Sulphur oxides |
| SSCS | Separated swirl combustion chamber |
| TCC | Trapezoidal combustion chamber |
| TDC | Top dead center |
| TRCC | Toroidal re-entrant Combustion Chamber |
| UBHC | Unburned hydrocarbons |
| ULSD | Ultra-low sulfur diesel |
| V ₂ O ₅ | Vanadium oxide |
| WDE | Water diesel engine emulsion |
| WFGCS | Wall-flow-guided combustion systems |
| ZrO ₂ | Zirconium dioxide |

effects of making alterations to the engine combustion chamber have been presented and discussed. In this, the effect of turbulence on biodiesel physical properties and its impact on NO_x emissions are analyzed. Thirdly, advanced injection strategies are presented, and their impact on biodiesel combustion is analyzed and discussed. Finally, the exhaust gas after-treatment system, which includes EGR, DOC, DPF and SCR were studied, and their effect on biodiesel emissions are discussed.

2. Advanced combustion strategies

Low-temperature strategies, combustion chamber design, injection strategies and exhaust gas retreatment systems are promising solutions for biodiesel fueled CI engines and are briefly discussed in this section.

2.1. Low-temperature combustion strategies (LTC)

LTC is an advanced alternative combustion process that can reduce NO_x by 85%, soot by 95% and BSFC by 15% in CI engines (Krishnamoorthi et al., 2019; Zheng and Caton, 2012). The working process in an LTC system involves the substantial premixing of the fuels. The main idea is to create a degree of dilution by delaying the start of combustion in the required space and allowing time for reducing the combustion temperature using a premixed charge (Azad et al., 2016a). LTC has produced better results in engine performance, reduced heat transfer rates, lean mixture combustion, and unthrottled operation and is able to work at high compression ratios (CR) (Reitz, 2013; Yan et al., 2021). Homogeneous charge compression ignition (HCCI), Premixed charge compression ignition (PCCI) and Reactivity controlled compression ignition (RCCI) techniques were widely recognized as LTC strategies with controlled combustion to limit NO_x emissions and improve indicated thermal efficiency. Shi et al. (2020c) studied the early stages of the LTC strategies and illustrated the combustion phasing process.

Fig. 4 illustrates the NO and soot distribution regions for the conventional and advanced combustion strategies as a function of local equivalence ratio (Φ) and temperature (K). The LTC cycles closely resemble the Otto cycle. The fuel gets burned at the premixed phase, and undergoes combustion of homogeneous lean mixtures inside the chamber. This phenomenon hinders the NO_x and soot emissions compared with the conventional CI

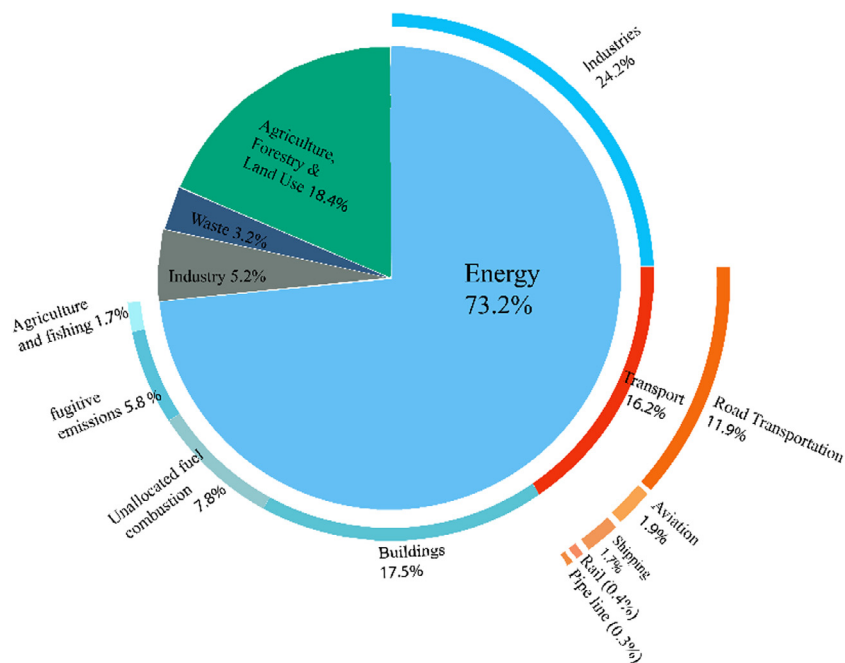


Fig. 1. Global GHG emissions by sector (Ritchie, 2020).

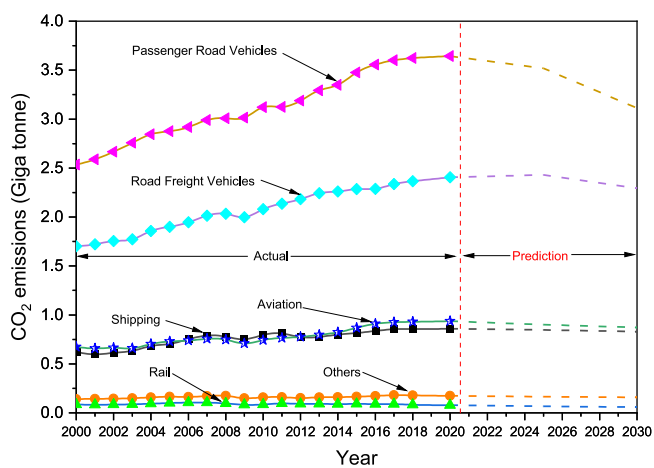


Fig. 2. Transport sector CO₂ emissions by mode in the Sustainable Development Scenario, 2000–2030 (IEA, 2022).

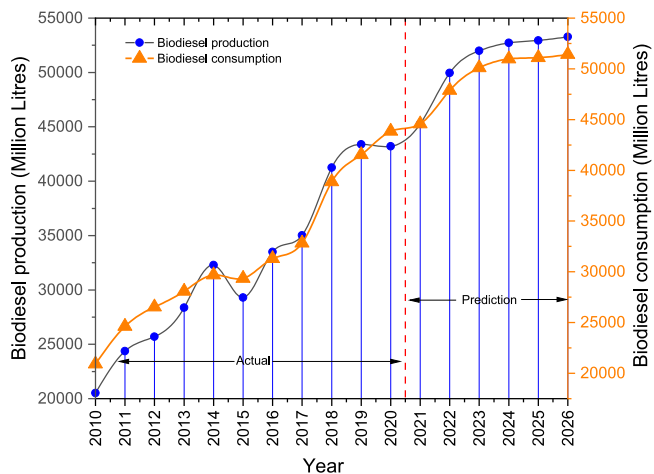


Fig. 3. Global biodiesel production rates from 2010 to 2022 in million tonnes (IEA, 2021).

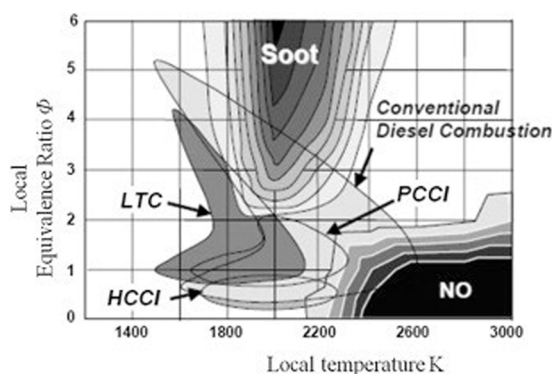


Fig. 4. Conventional and advanced strategy emission distribution under $\phi - T$ parameters (Lee and Huh, 2014).

engine emissions, as shown in Fig. 4. However, implementing the fundamental LTC principle faces several challenges, and the LTC concept has been gradually modified to attain the optimum LTC conditions for an engine. The HCCI combustion concept is ideal for the LTC principle because of the homogeneous combustion, whereas the air-fuel mixture formation is not homogeneous in PCCI (Agarwal et al., 2017). The overlap of these concepts is represented in the $\phi - T$ chart shown in Fig. 4.

2.1.1. Homogeneous charge compression ignition (HCCI)

In HCCI, a well premixed air-fuel mixture is compression ignited and initiates the combustion phenomenon (Duan et al., 2021). In short, the working principle of HCCI is the combination of working procedures involved in CI and spark ignition (SI) engines. Similar to SI engines, in HCCI, the homogeneous premixed charge helps to reduce NO_x and PM emissions by stabilizing the fuel-rich diffusion combustion process (Pipitone and Genchi, 2016). A stratified air-fuel charge acts as a homogeneous mixture of air and fuel. It undergoes two stages of heat release rates during the combustion process. The first stage is the low-temperature regime; here, only 10% of heat is released, and

the additional combustion is processed in a high-temperature regime (Yao et al., 2009). In between these two stages, a negative temperature coefficient (NTC) regime will be formed, creating a low reactivity atmosphere inside the chamber along with the rise in in-cylinder temperature. These temperature changes trigger HC and CO emissions (Tao et al., 2019; Yao et al., 2009).

The combustion process in the HCCI engine is difficult to control, whereas, in SI and CI engines, the start of combustion (SOC) can be controlled through ignition spark and fuel injection timing (Bendu and Murugan, 2014). Combustion in HCCI is very complicated to manage because it is operated through chemical kinetics from premixed fuel composition, the thermal state of the mixture and equivalence ratio (You-cheng et al., 2016). The combustion process in HCCI involves different operating conditions compared to CI engines. For instance, the compression ratio of the HCCI engine is usually high because of the lean mixture inside the chamber. Thereby, the equivalence ratio plays a critical role in combustion phasing. Several models are also proposed for advanced control in the combustion phasing of HCCI (Yang et al., 2020). The operating range for HCCI engines is minimal because the operational range is narrower than the conventional diesel engine. Knocking behaviors, high-pressure rate, and vibrations are observed at high loads because of the early SOC. At low loads, misfire tendencies will occur due to longer ignition delay and low exhaust temperatures (Calam, 2020; Lee et al., 2015a). Several strategies, like fuel stratification, variable valve timing (VVT), EGR stratification, open valve injection, etc., were used to extend the operating range limits for the HCCI engine (Lee et al., 2015a).

2.1.2. Premixed charge compression ignition (PCCI)

PCCI is the compromised representation of both HCCI and conventional diesel engine combustion (El Shenawy et al., 2019). The PCCI concept brings more practicality to engine combustion than HCCI because PCCI's combustion phasing is linked with fuel injection (Imtenan et al., 2014). Premixed combustion in LTC strategies creates a more effective combustion process. The in-cylinder pressure inside the chamber will build effectively compared to the conventional charge compression ignition (Jain et al., 2017a; Tarabet et al., 2014). PCCI conditions will create lean premixed regions that reduce emissions simultaneously but with limited operating conditions (Soloiu et al., 2013). For example, Salahi and Gharehghani (2019) assisted the control in combustion phasing by adding the ozone proportions to the intake air. Advancements in the combustion phasing with the increasing ozone proportions were observed, along with reductions in HC, CO, and NO_x emissions (Salahi and Gharehghani, 2019).

Limited operating range, higher HC and CO emissions, and lack of combustion control are major drawbacks of the PCCI concept. The auto-ignition behavior of the fuel is the prime feature of controlling the combustion phasing in PCCI. For instance, neat diesel exhibits better auto-ignition characteristics, and it is not easy to control the combustion phasing at increasing loads (Hanson et al., 2010). Alternative fuels with high resistance to auto-ignition and with higher volatility, perform better homogeneous mixing rate inside the combustion chamber and extends the operating range in the PCCI engine (Chen et al., 2020). PCCI combustion shows excellent results in reducing HC and CO emissions compared to HCCI, except for NO_x and soot emissions.

2.1.3. Reactivity controlled compression ignition (RCCI)

RCCI is a dual fuel combustion mode in which two fuels of different reactivity rates are sent into the chamber proportionally at various stages to conduct combustion at low temperatures (Duraismy et al., 2020; Kakoei et al., 2020). A low reactivity-high octane number fuel (LRF) is injected through port injection along with the intake air and EGR. Proceeding, the high reactivity fuel

(HRF) is directly injected into the combustion chamber. The auto-ignition phase starts after the injection of HRF into the chamber, and the fuel-rich compressed charge will undergo sequential staged combustion. In RCCI, this staged combustion helps in controlling the heat release rate (HRR) better than other LTC strategies (Kokjohn et al., 2015).

Most importantly, the direct-injected fuel controls the local reactivity inside the chamber. RCCI is showing negligible NO_x and soot emissions. For instance, Splitter et al. (2012) stated that, with RCCI, 56% gross thermal efficiency was achieved, and it can be extended to 60%. Besides, Huang et al. (2020) observed the dual fuel reactivity combustion by supplying both low and high reactivity fuels into the combustion chamber through two direct injectors. The authors observed NO_x emissions below 0.6 g/kWh followed by acceptable HC and CO emissions, and also noted 53.5% of indicated thermal efficiency. Duraisamy et al. (2020) performed combustion analysis on an RCCI engine with Methanol-PODE and Methanol-diesel dual fuels. The authors concluded that the higher-octane fuel and latent heat of vaporization of methanol are resisting the auto ignition and extending the ignition delay. Also, a shorter ignition delay (ID) was observed with Methanol-PODE operation because of the oxygen content and the higher cetane number in PODE achieving a better fuel combustion process compared to a conventional diesel engine. RCCI combustion is showing excellent results in NO_x reduction when compared to conventional operation of HCCI and PCCI. Approximately a three times reduction of NO_x emissions, a decrease in soot by over 6 times and around a 16.4% higher gross indicated efficiency were achieved using RCCI compared to conventional CI engines (Reitz and Duraisamy, 2015).

Though RCCI is showing promising results in performance and emissions, yet these benefits are accessible for mild-load conditions only (Kavuri and Kokjohn, 2017). At low loads, the engine's thermal efficiency is low, and HC and CO emissions will be high due to low-temperature combustion near the cylinder wall (Fu et al., 2020; Xu et al., 2019). At high loads, the rate of pressure increases, and it becomes challenging to control combustion even with the fuel stratifications. If the LRF proportions increase, the pressure rise rate (PRR) exceeds the limit. On the other hand, if more HRF fuel is added through direct injection, the soot and NO_x emissions will increase (Wang et al., 2020). Several alternative strategies are proposed to extend the operating conditions, such as lower compression ratio (Xu et al., 2019), variable valve actuation (Xu et al., 2019), alcohol as fuel and EGR. These strategies show improving outcomes at high loads, but the results are still unsatisfactory at low loads. On the contrary, the direct dual fuel stratification combustion mode shows consistent results at low loads compared with the RCCI combustion (Fu et al., 2020).

2.1.4. Discussion on low-temperature strategies

LTC strategies have several advantages, such as delayed combustion with reduced throttle losses, significant reductions in NO_x and soot emissions and fuel flexibility that are able to cope with the SI and CI engine working habits (Goldsborough et al., 2017). Early fuel injection and delayed start of combustion with the lean mixture proportions opened the opportunity for the low-temperature atmosphere inside the combustion chamber (Drews et al., 2010). All LTC modes showed low-temperature heat release followed by the main heat release. The low-temperature combustion phasing varies for these modes as it occurs early for HCCI and later for PCCI and RCCI (Murugesu Pandian and Anand, 2018). Injecting a lean mixture inside the chamber progressed through the high air to low fuel ratio (equivalence ratio) and, with the use of EGR near TDC, caused the delayed combustion. As a result of the early injection, fuel impingement will occur inside the chamber and causes more extended jet penetration,

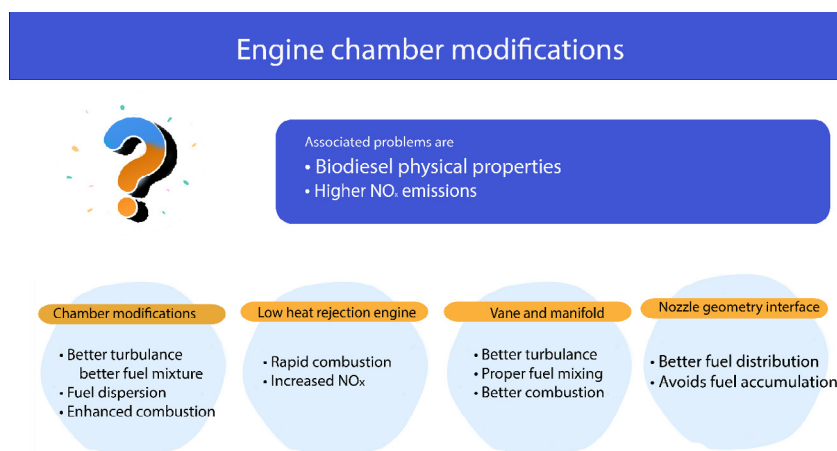


Fig. 5. Effect of chamber modifications on combustion.

leading to delayed combustion (Saxena and Bedoya, 2013). On the other hand, early injection of fuels with a higher boiling range in CI engines can lead to wet cylinder liners that influence the oil dilution, formation of fuel films and pool fires.

At the same time, a longer ignition delay (ID) in LTC helps achieve homogeneous mixture formations for better combustion. Concepts like late injection and EGR applications are responsible for the longer ID. Also, longer ID and lean mixture combustion condition can lead to incomplete combustion and accounts for unburned hydrocarbons (HC). EGR flue gases encompassed combustion also lead to longer ID because of the lower oxygen content charge supply into the chamber. This dilution directly affects the chemical kinetics inside the combustion chamber by slowing down the formation of active radicals and reducing their concentration (Dulbecco et al., 2009). Fuel-based control over combustion also helps in achieving a longer ID. Fuels with high volatility and lower cetane number provide low-temperature combustion with reduced emissions (Atmanli and Yilmaz, 2020).

From the discussion of the above two alternatives, it is understood that fuel reactivity mainly controlled ignition, is one of the best ways to introduce the LTC strategies in to CI engines. Several studies presented multiple control concepts to control combustion. These strategies include variable valve timing, variable CR, water injection, tailoring fuel mixtures with different reactivities, and thermal stratification by delayed fuel injection.

Other LTC concepts such as stratified charge compression ignition (SCCI) and partially premixed charge compression (PPCC) are also derived from the HCCI concept. A stratified fuel injection in HCCI combustion is considered as the SCCI (Fathi et al., 2017). In PPCC, control over combustion is achieved by utilizing different injection events to reduce the peak pressure rates (Ma et al., 2013). Li et al. (2020) proposed a new combustion mode called intelligent charge combustion mode (ICCI). In this mode, the charge is entered into the combustion chamber through two common rail direct injection systems that are independently controlled by the electronic control unit. Li et al. found that NO_x emissions are extremely low and indicated a significant thermal efficiency increase. The premixed proportions and low-temperature atmosphere inside the chamber influences fuel vapor and create non-oxidized portions near liners and crevice regions that lead to HC and CO emissions (Li et al., 2016b; Pachiannan et al., 2019). Table 1 presents the effects of LTC strategies on the performance and emission characteristics of CI engines.

2.2. Modification of engine chamber

Modification of the combustion chamber significantly impacts the combustion process. Modifications to the engine chamber are one active technique that directly affects the combustion process through air-fuel mixings. The engine chamber design greatly influences the local air-fuel mixture and heat transfer rates (Leach et al., 2018). The shape of the combustion space, valve and port design modifications, nozzle holes, and spray angle modifications mainly affect turbulent motions inside the chamber and influence the combustion process. The overall effects of the chamber modifications are outlined in Fig. 5.

2.2.1. The shape of the combustion chamber

Combustion chamber geometry influences the air-fuel charge by generating squish, swirl, and tumble motions inside the chamber (Prasad et al., 2011; Venu et al., 2019). Recent works in the literature revealed that the turbulence motions are still active even after closing the intake valve (Micklow and Gong, 2007). Combustion chamber modifications play a crucial role in in-cylinder fluid dynamics, especially for the CI engines (Gnana Sagaya Raj et al., 2013). Squish formation occurs during the compression stroke, where the injected fuel flows radially into the geometry and helps in the air-fuel composition reformations (Fu et al., 2020). The swirl and tumble motions occur radially inside the chamber in parallel and perpendicular directions to the cylinder axis (Bari et al., 2020). These motions generate high turbulence inside the chamber and increase the in-cylinder pressure and temperature (Abdul Gafoor and Gupta, 2015). Chamber modifications improved the BTE and reduced the BSFC, along with HC, CO, smoke, and soot emissions (Karthickeyan, 2019; Sener et al., 2020; Singh Varun et al., 2017).

Fig. 6 represents some of the piston bowl geometry modifications. Also, several investigations are conducted on different combustion chambers such as Double-layer diverging combustion chamber (DLDC) (Fu et al., 2020), Rotary engine (Cihan et al., 2020; Shi et al., 2020a), hemispherical Combustion Chamber (HCC) (Khan et al., 2019; Vedagiri et al., 2020), Trapezoidal combustion chamber (TCC) (Jyothi and Reddy, 2020; Vedagiri et al., 2020), toroidal re-entrant Combustion Chamber (TRCC) (Jyothi and Reddy, 2020), shallow depth combustion chamber (SCC) (Vedagiri et al., 2020), wave-shaped piston bowl (Zhang et al., 2020a), double swirl piston bowl (DSPB), multiple swirl piston bowl (MSPB), Nebula combustion chamber, wave-shaped piston bowl geometry (Zhang et al., 2020b), omega combustion chamber (Li et al., 2014), separated swirl combustion chamber

Table 1
Effect of LTC strategies on performance and emission characteristics of CI engines.

| LTC mechanism | Engine parameters | Performance | | Emissions | | Reference | |
|---------------|---|-------------|-----------|-----------|-----------|-----------|------------------------------------|
| | | BTE | BSFC | CO | HC | | NO _x |
| HCCI | 4-stroke, 4- cylinder, RP: 105 kW, CR: 17.2:1, RS: 1800 rpm, IP: 1200 bar, CRDI | i | – | – | d | d | Mathivanan et al. (2016) |
| HCCI | Single cylinder, RP: 4.4 kW, CR: 17.5:1, RS: 1500 rpm, IP: 200 bar, CRDI | d | – | i | i | d | Ganesh and Nagarajan (2010) |
| HCCI | 4-stroke, 1- cylinder, CR: 13:1, RS: 1200 rpm, Injection pressure: 0.3 MPa (port) and 5 MPa, (Direct), Injection: Direct and port injection | – | – | d | d | i | Yang et al. (2011) |
| HCCI | 4-stroke, 1- cylinder, RP: 7.4 kW, CR: 17.5:1, RS: 1500 rpm, IP: 200 bar, pilot and main, Premixed ratio: 0%–80% | – | d | i | i | d | Das et al. (2015) |
| HCCI | Four-stroke, four-cylinder, SI, CR: 12:1 | d | i | No change | d | d | Hussaini et al. (2016) |
| HCCI | Single cylinder, CR: 10.5:1, Direct and port injection | – | – | – | – | d | Lee et al. (2015b) |
| PCCI | 4-stroke, 1- cylinder, CR: 15.5:1, RS: 1500 rpm, IP: 75 MPa, CRDI | – | No change | i | i | d | Lee et al. (2015c) |
| PCCI | Single cylinder, RP: 6.25 kW, CR: 17.5:1, RS: 4200 rpm, IP: 700 bar, CRDI | i | d | i | i | d | Jain et al. (2017b) |
| PCCI | Twin cylinder, CR: 17:1, RS: 1500 rpm, CRDI | d | i | i | i | d | Wang et al. (2013) |
| PCCI | 4-stroke, 2- cylinder, RP: 21.9 kW, CR: 17:1, RS: 2300 rpm, CRDI | d | i | i | i | d | Zhao et al. (2014) |
| PCCI | 4-stroke, 1- cylinder, CR: 17.5:1, RS: 1500 rpm, RP: 4.4 kW, IP: 21 MPa, direct injection | – | i | No change | No change | d | Rajesh Kumar and Saravanan (2016) |
| PCCI | 4-stroke, 1- cylinder, AC, RP: 4.4 kW, CR: 17.5:1, RS: 1500 rpm, IP: 220 bar, CRDI | d | i | i | i | d | Saravanan et al. (2015) |
| RCCI | 4-stroke, 1- cylinder, CR: 19.5:1, RS: 1500 rpm, IP: 250–260 bar, port and CRDI | – | i | i | i | d | Kalsi and Subramanian (2017) |
| RCCI | 4-stroke, 1- cylinder, CR: 19.5:1, RS: 1500 rpm, IP: 250–260 bar, port and CRDI | d | i | i | i | d | Singh Kalsi and Subramanian (2016) |
| RCCI | Single cylinder, CR: 17.8:1, RS: 1200 rpm, IP: 0.4 MPa-biogas, 3 MPa (gasoline), port and CRDI | i | – | i | i | d | Park and Yoon (2016) |
| RCCI | 4-stroke, 4- cylinder, CR: 17:1, RS: 1500 rpm, port and CRDI | i | i | i | i | d | Işık and Aydın (2016) |
| RCCI | Single cylinder, CR: 18.5:1, RS: 800–2000 rpm, port and CRDI | i | d | d | d | d | Zhu et al. (2015) |
| RCCI | 4-stroke, 4- cylinder, CR: 17.5:1, RS: 1000–2200 rpm, IP: 100 bar, port and CRDI | – | – | i | i | d | Benajes et al. (2016) |

i – increase
d – decrease

(SSCS) (Zhou et al., 2020) and grooved piston surfaces. All these combustion chambers exhibited varying combustion rates and demonstrated that chamber modifications do play a significant role in the engine combustion process and emissions formation.

Slight modifications to the combustion chamber geometry affect the combustion behavior and directly influence the combustion and emission parameters (Doppalapudi et al., 2021). Chan-nappagoudra et al. (2019) observed an 8.42% and 6.21% decrease respectively in CO and HC emissions at mid load, a 5.5% increase in BTE, and a 6.89% reduction in BSFC at full loads for a modified TRCC chamber when compared to a HCC chamber. Jaichandar et al. (2012) conducted an experimental analysis on different bowl geometries of TRCC, SCC and HCC types with 20% POME biodiesel. Outcomes revealed that the changes in geometry and fuel resulted in CO, UBHC, and PM being lower in TRCC than for the other two. On the other hand, peak pressure at high loads and high NO_x emissions are the associated drawbacks during the combustion process. The injected fuel passes around the bowl periphery and gets ignited inside the bowl geometry. Thus, chamber modifications revealed reduced combustion duration, shortened ignition delay, high cylinder pressures and temperatures and high heat release rates. The chamber geometry shapes,

lip corners, bowl diameter, depth and throat diameter play a significant role in combustion formation and fluid distribution. At the same time, the asymmetric shapes of these chambers revealed asymmetric behaviors created by the geometric protrusions inside the chamber. In some cases, the injected fuel gets diverted into the squish and crevice regions where the chances for incomplete combustion are high. In addition, these behaviors will magnify at higher loads and speeds, and it is difficult to control the combustion at this stage due to high turbulent kinetic energy formation rates. Overall, combustion chamber modifications need more design considerations as appropriate changes to these chambers can greatly improve the combustion process without major retrofitting to the engine.

2.2.2. Low heat rejection engine (LHR)

Providing a thermal barrier coating inside the chamber is termed as a low heat rejection engine. It is always advantageous to make the system adiabatic to utilize the heat energy released from the combustion process to improve the system's thermal efficiency (Karthickeyan et al., 2020). The thermal resistance of a combustion chamber is improved using a ceramic coating and

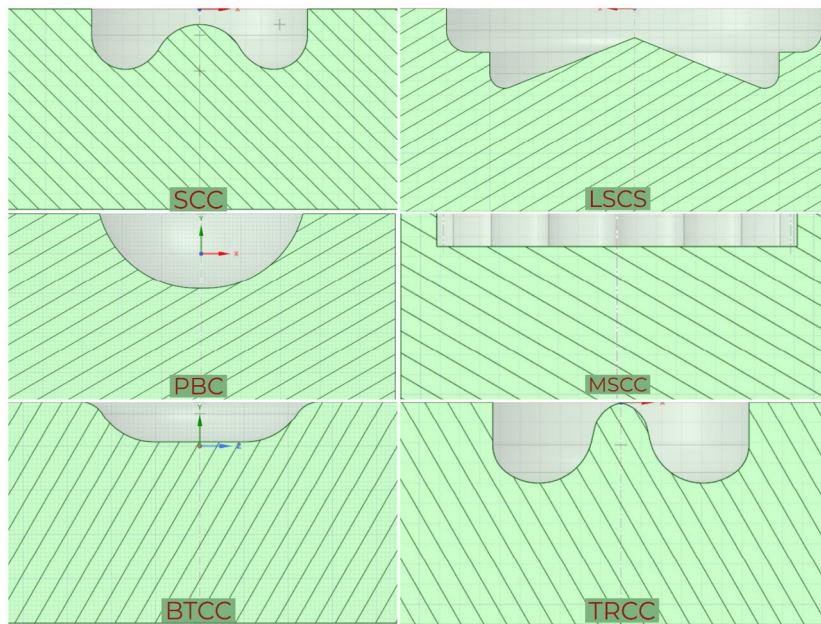


Fig. 6. Different shapes of the combustion chamber.

air gap insulation (Krishna et al., 2014). Also, an insulated coating technique is used to reduce the heat transfer rate from combustion chamber elements to the cooling jacket. In this way, the heat energy produced in the combustion is converted to useful piston work without major heat losses, and this will increase the in-cylinder gas temperature by 200 °C compared to the base engine (Ellappan and Rajendran, 2021). These differences will affect the combustion process through reduced ID, shortened combustion duration (CD) and diffused combustion (Öztürk et al., 2019). The heat retained in the cylinder will help oxidize the emissions at higher temperatures, reduce the CO and UBHC emissions, and increase the brake thermal efficiency (Elumalai et al., 2021a; Murali Krishna M et al., 2014). Biodiesel fuels with a low cetane number can be used in LHR engines. Prasath et al. (2010) observed LHR operation with Zircona (PSZ) with 0.005 m thickness and observed high-temperature conditions and high heat release rate for both diesel and biodiesel but, on the other hand, NO_x emissions were relatively high, which were subsequently stabilized by EGR and water injection into the chamber. For instance, Karthickeyan et al. (2019a) observed good performance and reduction in NO_x emissions for the LHR engine using SCR systems. Different ceramic coatings such as CaZrO₃ (Taymaz, 2006), ZrO (Mittal et al., 2013), MgO-ZrO₂ (Öztürk et al., 2019), PSZ (Ellappan and Pappula, 2019; Karthickeyan, 2020; Karthickeyan et al., 2019b), YSZ (Reddy et al., 2019), Al₂-TiO₂ (Öztürk et al., 2019), mullite (Rao et al., 2016), TiO₂ and many more (Cao et al., 2004) are studied as the possible in-cylinder liners for the LHR engine. The volumetric efficiency of the combustion chamber is very challenging in LHR engines and directly influences the energy conversion efficiency (Elumalai et al., 2021b; Gangula et al., 2021; Karthickeyan et al., 2019a).

2.2.3. Vane and intake manifold modifications

Vanes and intake manifold designs are the other modifications that help to improve the turbulence in the intake air. Turbulence is created in the form of swirl and tumble during the intake stroke, which enhances the fuel evaporation and mixing process. For example, Kui et al. (2019) conducted a simulation study with different virtual guide vanes by changing the vane size and angles at each stage and observed an improvement in the swirl ratio of about 24.69%. In addition, Bari et al. (2015a) observed

a slight increase in the in-cylinder pressure and temperatures with modifications to the guide vanes. Moreover, Ramadan (2003) observed that decreasing the port diameter did not affect the swirl ratio, but the pumping losses are high. Guide vane design parameters such as vane angle (Bari and Saad, 2013), shape (Bari et al., 2015b; Dhingra et al., 2015), length (Bari and Saad, 2014) and number (Bari and Saad, 2015) have shown improved engine combustion characteristics with improved turbulence (see Fig. 7).

A well-designed intake manifold greatly influences the inflow characteristics of the air-fuel mixture (Xu, 2017; Yang et al., 2017). Maintaining the low pressure, equal static pressure distribution, minimizing the sudden bends and curvatures and maximizing the air velocity in the system are some of the major considerations to be taken while designing the intake manifold. The manifold pressure variations also play a crucial role in the engine's volumetric efficiency (Silva et al., 2019; Souza et al., 2019). For instance, Shrirao et al. (2014) conducted a simulation study with different internal threads grooved inside the intake manifold and noted an 11.62% reduction of BSFC for the grooved manifold when compared to the normal intake manifold. Also, the authors observed the reduction of NO_x by 3.6%, CO by 26.66% and HC by 12.32% for the buttress internal threaded manifold in comparison with the baseline manifold. A well-designed manifold influences the flow rates and improves the engine's efficiency (Bari et al., 2020).

2.2.4. Nozzle geometry interface inside the combustion chamber

In the combustion chamber, the nozzle interface plays a significant role in the rate of combustion process through atomization and fuel dispersion. Nozzle interface modifications involve nozzle diameter, nozzle orifices, and spray targeting positions that affect the in-cylinder combustion efficiency. Shivashimpi et al. (2019) observed that, with the increased nozzle holes from 6 to 7 at constant injection pressure (IP), injection timing (IT), speed and compression ratio (CR), the BTE and HRR were increased by 1.5% and 2.4%, respectively. Similarly, Yaliwal et al. (2016) observed improved combustion characteristics with the HOME biodiesel with a 5-holed nozzle compared to the 3-holed nozzle results. Moreover, flame propagation, energy dispersion, emissions, HRR,

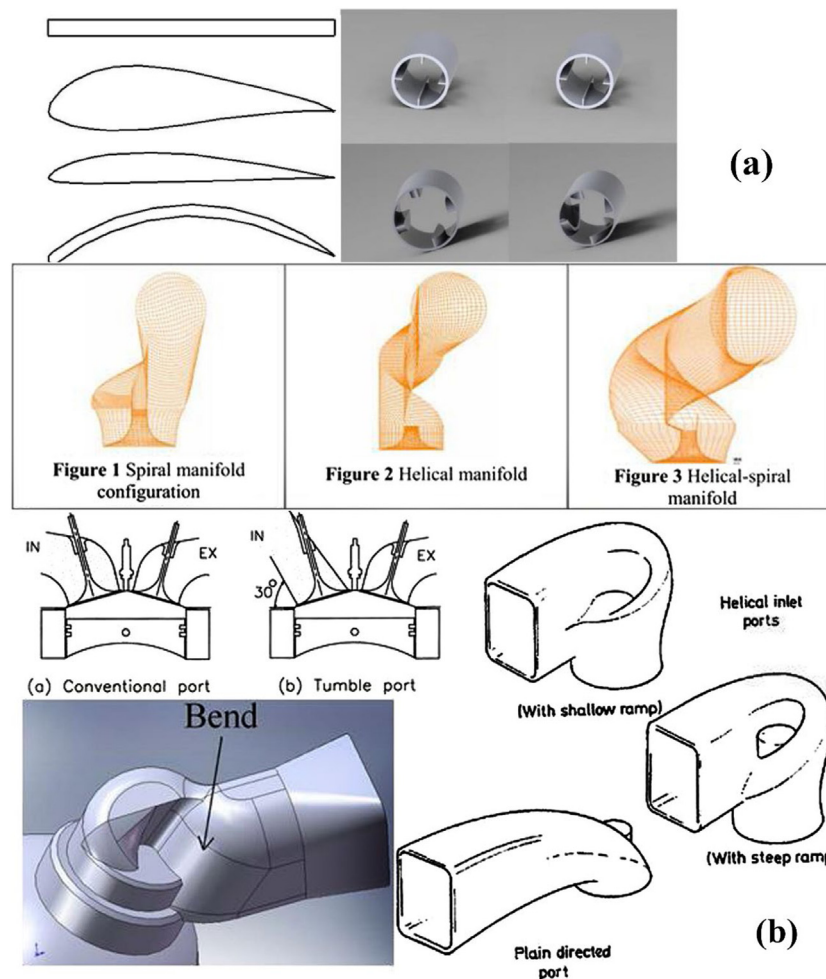


Fig. 7. Vane design and manifold design modifications (Bari et al., 2020).

and in-cylinder pressures are affected by spray behavior, which is also associated with fuel droplets, air–fuel mixture, and cavitation.

Spray angle is the most critical parameter for characterizing the spray distribution inside the chamber. Khan et al. (2018) observed an increase in cylinder pressure, in-cylinder temperature and HRR with the increase of spray angle. Zhou et al. (2020) performed a numerical study on the SSCS combustion chamber with different spray angles. The study noted decreased BSFC and soot emissions with improved thermal efficiency. Poorghasemi et al. (2017) observed a decrease in NO_x emissions while narrowing the spray angle. On the other hand, the study found an increase in HC and CO emissions and a decrease in BTE. Sener et al. (2020) observed a decrease in soot emissions by 81% with the increase of spray angle and IP. It has been understood that the increase of spray angle provides a wider distribution to the fluid particles thereby enhancing the combustion rate. However, the excess angles will cause the fuel impingements onto the wall and pass the fuel to squish and crevice regions. This causes incomplete combustion and leads to higher CO, HC and soot emissions.

Another concept of modifications to the CI engine is the porous medium direct injection system (PMDI). The PMDI concept was initially proposed by Durst and Weclas (2001) for preparing the steady-state homogeneous mixture formation inside the cylinder chamber. These authors added a porous medium after the injection and before the combustion chamber in such a way that the injected fuel enters the combustion chamber by dispersing through the small pores from the PMDI. This allows the free expansion of the fuel droplets inside the porous matrix and

results in rapid vaporization of the fuel. The authors observed ultra-low NO_x emissions, lower HC, CO emissions, noticeable reduction in soot emissions, low combustion noise and higher cycle efficiency. Moreover, the temperature released from the combustion chamber adds an additional heat source to the PMDI for rapid evaporation (Saghaei and Mohammadi, 2019).

2.3. Modification of injection strategies

Fuel injection systems play a consequential function during combustion. Modern fuel injection systems are simple yet complex operating devices, that influence the potential combustion events managed with multiple injections. For instance, an electrically operated common rail direct injection (CRDI) system offers flexible operations and precise control over the fuel injection rates. Conventional injectors have limitations on injection pressures that provide a maximum pressure of 400 bar with associated shims (Khandal et al., 2017), whereas CRDI provides up to 1600 bar (Agarwal et al., 2014). Increased combustion performance and decreased BSFC, PM and NO_x emissions can be observed with CRDI compared with engines with mechanical injectors (Ghaedi et al., 2020; Senatore et al., 2000).

2.3.1. Injection pressure (IP) and injection timing (IT)

IP is a vital parameter associated with the fuel injectors and is practically responsible for the fuel impingements and accumulations inside the chamber. IP denotes the pressure of the fuel at the entrance of the injection. Higher injection pressure leads to

the better mixing of air and fuel through the formation of finer atomized particles with superior dispersion. First of all, biodiesel itself exhibits higher IP due to higher bulk modulus and causes a shorter ignition duration (usually with a lower volumetric flow rate) (Szybist et al., 2007). That means an increase in IP with biodiesel results in more mass delivery and density. The Sauter mean diameter which is used to analyze the quality of atomization of injected fuel varies with the fuels (Suh et al., 2008; Yuan et al., 2005). Moreover, the fuel adhesive nature of biodiesel is shortened with the increase in IP, resulting in rapid vaporization. Thus, the physical properties of biodiesel are influenced by the injection pressure in terms of spray penetration (Huang et al., 2017; Shi et al., 2020b).

Yao et al. (2017) observed the longer flame lift-off length with the reduced combustion duration and ignition delay period achieved by increasing the injection pressure. Belagur and Chitmini (2010) increased the IP from 200 bar to 240 bar and noticed an increase in BTE and NO_x emissions and a reduction in CO, HC, and SO emissions. Many studies reported that emissions like CO, HC, PM and smoke were reduced with the advanced IP, but the CO₂ and NO_x emissions increased (Emiroğlu, 2019; Gumus et al., 2012; Saravanan et al., 2020; Yesilyurt, 2019). Sarıdemir et al. (2020) conducted an experimental analysis with the methyl-ester diesel blends by increasing the IP from 210 bar to 230 bar. The study observed a decrease in CO and HC emissions by 66.67% and 52.38% respectively and an increase in NO_x by 22.45%. Shrivastava and Verma (2020b) observed a decrease in NO_x emissions and an increase in CO₂ with the rise of IP. Rajak et al. (2019) observed a 22.9% decrease in PM with increased IP using microalgal-diesel fuel. Yoon et al. (2019) observed a decrease in BSFC, HC, CO, and PM emissions with increasing fuel IP. Injection pressure and engine load are the functions of in-cylinder pressure (ICP) because ICP is increased with the increasing load and IP (Shrivastava and Verma, 2020a). Gumus et al. (2012) observed the effect of IP with different biodiesel blends and noted the decrease in BSFC in the higher biodiesel portion blends and a decrease in UBHC and CO emissions. As discussed above, in most cases, NO_x emissions are increased with advancing IP which increases in-cylinder temperatures and triggers the oxides of nitrogen. An increase in IP for biodiesel influences the fuel mixture through atomization which can help the LTC strategies to perform the combustion at leaner mixtures. This can help to reduce the NO_x emissions with biodiesel. At the same time, reduced IP leads to retardation of ID and this will help in the case of biodiesel with lower heating value and higher viscosity. Rapid combustion with the higher viscous biodiesel fuels causes poor combustion because the viscous biodiesel forms fuel films inside the chamber. Hence the injection pressure adjustment needs to be more associated with the fuel properties than with the fuel type.

IT plays an essential role in the combustion rate because it correlates directly with the full combustion process. As discussed before, biodiesel itself has a shorter ignition timing due to its bulk modulus. Monyem et al. (2001) observed that biodiesel has an artificial advance in the injection of about 2.3 °C compared with diesel despite changes to the injection setting. These differences clearly suggest that the controlled adjustments in IT are necessary for controlled biodiesel fueled combustion. Advanced IT in biodiesel combustion has revealed higher NO_x (Raeie et al., 2014). A typical LTC strategy follows the late injection technique (near TDC) to achieve low in-cylinder temperatures (Zheng and Caton, 2012). During the early injection, the temperature and combustible pressure inside the chamber provide distinctive space for the fuel mixings and cause longer ID. Retarded injection timing leads to a longer ignition delay and processes with reduced heat transfer exergy rate and increased net flow exergy (Zheng and Caton, 2012). Jaichandar and Annamalai (2012) noted that, with

the retardation of IT, marginal decrements in in-cylinder temperatures, NO_x emissions, and maximum heat release rate occurred. Hu et al. (2017) observed an increase in NO_x emissions and a decrease in BSFC and soot emissions with the increase in IT. The overall response is that advances in IT happen due to the usual biodiesel property activity and changes to the injection timing setting (Sun et al., 2010). Considering the cases of IP and IT, it is understood from the findings that increased IP and retarded IT is an efficient way to get better control over combustion and can reduce NO_x emissions. For instance, Kannan and Anand (2012) observed an increase in BTE by 1.44% and a decrease in smoke and NO_x by 17.2% and 6.5% respectively with an increase in IP by 280 bar and at an IT of 25.5° BTDC (Kannan and Anand, 2012).

2.3.2. Multiple injection system

In multiple injections, the fuel is injected into the combustion chamber at different crank angles. Increasing the IP and IT will affect the air-fuel mixing rates, and it is difficult in some cases to control the combustion with these parameters. Combustion inside the chamber can be controlled with different injection modes proportional to the fuel quantity. For instance, with multiple injection strategies, one can create ordered combustion by the fuel at regular intervals. Fig. 8(a) and (b) illustrate the correlation between the mass of the fuel injected and the apparent heat release rate (HRR). Each peak on the HRR graph represents heat release rates with respect to the injected masses at that respective crank angle. The multiple injections are classified into two sections, single-stage and multi-stage injections, and are discussed in the subsections below.

2.3.2.1. Split injection. Split injection reduces the peak pressure rates inside the chamber with reduced NO_x and PM emissions (Nehmer and Reitz, 1994). Fuel is injected into the chamber in portions to avoid excessive lean or rich mixtures. Many authors reported that this two-stage injection performs a smooth transition with lean mixtures and creates stratified combustion at full loads (Matsushita et al., 1997; Zhao et al., 1999). Yang and Anderson (1998) observed an increase in output torque at full loads with split injection. Kim et al. (2008) performed an experimental analysis with a split injection strategy with neat soybean biodiesel and observed an extreme reduction in NO_x emissions. How et al. (2018) observed the lower NO_x emissions for B50 and B20 fuels with triple split injections. In addition, Park et al. (2011) performed an experimental analysis with split injection and multistage injection strategies. The authors observed that the first injection showed an increase in indicated mean effective pressure (IMEP) compared to the split injection. Also, multistage injection combustion exhibited a decrease in soot, CO and HC emissions by trading off the NO_x emissions. Yehliu et al. (2012) noted that the soot generated with split injection formed smaller carbon layers than a single injection. Moreover, split injection caused a longer ignition delay irrespective of the injected mass and injection pressure (Wang et al., 2015). Performance wise, BSFC has reduced with the split injection strategies compared to the single injection (Huang et al., 2016; Li et al., 2016a; Minami et al., 1995; Mohan et al., 2013). These studies indicated that this injection strategy creates the transition between part and full load operations to improve the combustion. During the transition period, the air-fuel mixture is controlled and suppresses the peak pressure rise rates and high in-cylinder temperatures, which are responsible for higher NO_x emissions.

2.3.2.2. Multi-stage injection. A quantity of fuel is sent to the combustion chamber in multiple stages (early/pilot-main-/post/late) for better combustion. Different pulses are injected into the chamber at different mass rates but with the same overall fuel

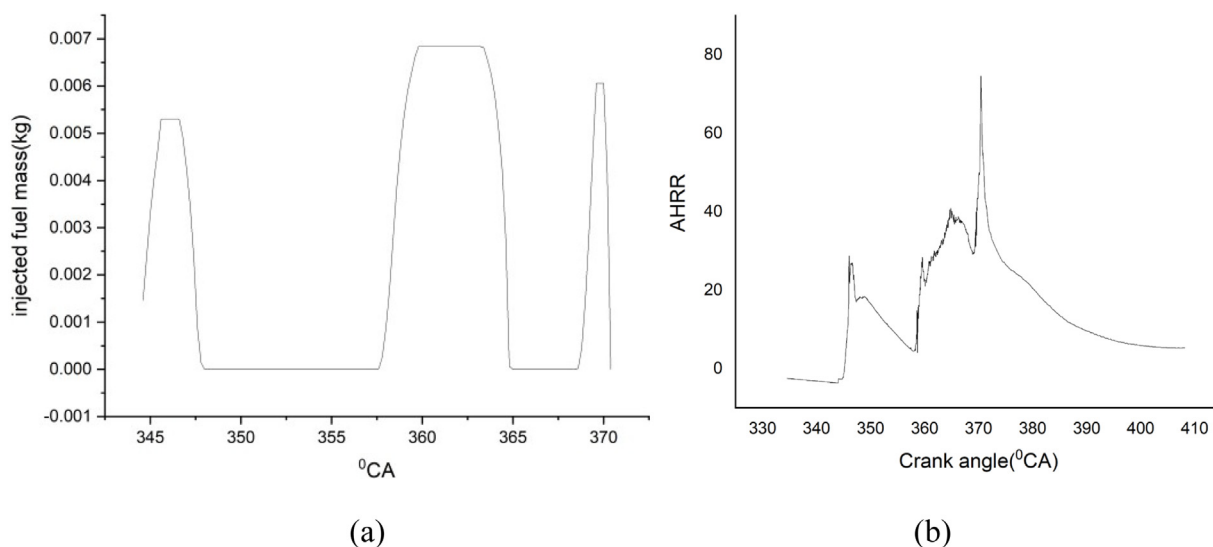


Fig. 8. Fuel mass effects on heat release rate: (a) mass of the fuel vs. crank angle; and (b) apparent heat release rate (AHRR) vs. crank angle.

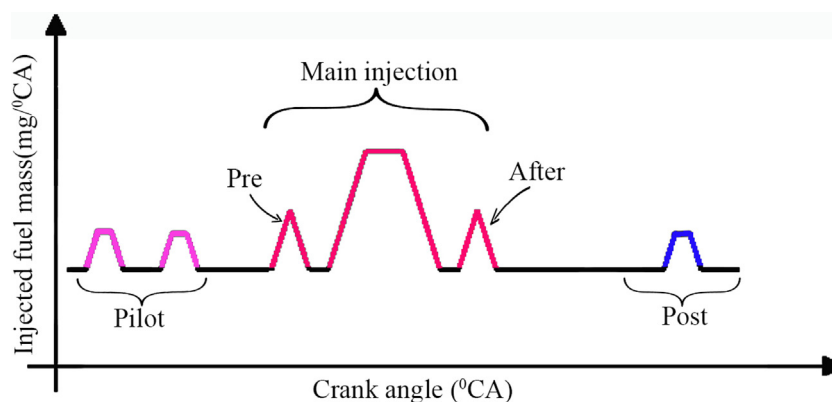


Fig. 9. Types of injection pulses during the multistage injection.

quantity. Fig. 9 represents the different types of fuel pulses associated with the multistage injection. The number of injections will vary depending on the fuels and operating conditions because the pulse number depends on injection rate and combustion duration. Modern injection systems allow up to 8 pulses per cycle (Payri et al., 2020). Depending on the stages and applications, various multistage injections are classified as double injection and triple injection strategies. Injection modules such as pilot-main, early-main, and main-post are classified as double injection strategies. Also, injections like early-pilot-main and pilot main-post are derived under triple injection strategies (Denny et al., 2019; Mathivanan et al., 2016).

Multiple pulse strategies showed a simultaneous reduction of NO_x and PM emissions in CI engines (Reitz and Rutland, 1995). NO_x emissions depend on both the pilot and the main injection. With advancing of the pilot injection and retardation of the main injection, the emissions showed a decreased trend (Kim et al., 2007). For instance, the secondary spray turbulence intensity is affected by the first injection flow, and as discussed, the second injection (main) is also responsible for the overall combustion. Here the spray turbulence depends on the dwell time and injection parameters. During the first injection, air entrainment at the spray boundary is generated; thus, the flammable mixture is formed by vaporizing the fuel. However, the ignition occurs at high temperatures and in high fuel mixture regions. During the second injection, the outer boundary region of the mixture in

front of the second injection is favorable for ignition under a short dwell time compared to the frontal region (Cung et al., 2015; Zhou et al., 2021). For instance, Huang et al. (2015) observed a decrease in heat release rate with the advanced pilot injection timing, but the HRR rate eventually increased with the main fuel injection. The study also revealed that the heavy pilot injected mass effects on the main injection mass, leading to peak pressure rates along with decreased NO_x and soot emissions. Further, Farhan et al. (2020) conducted an experimental analysis with different post-injection strategies and observed a decrease in NO_x , HC, and soot emissions at medium and low load conditions. The study found that unregulated emissions are increased by increasing the mass quantities of post-injected fuel. In some cases, more mass pulses create flexible operations and provide control over the heat release rate and air-fuel mixture formations.

Yang et al. (2015) performed a double injection with LTC strategies by using gasoline and observed that, with the increase of the split percentage, NO_x emissions were reduced with increased soot emissions. Wang et al. (2016) performed a combustion simulation with micro-pilot induced injection for a natural gas engine using modified combustion chamber geometries. The study noted a decrease in HC and CO by 56.47% and 33.55% respectively. Fang et al. (2012) observed reductions in NO_x emissions with the increase in pilot quantity to a certain level and noted the rise in BSFC. The effect of turbulent fluctuation with multiple injections also affects the ID. Thus, NO_x and soot can be

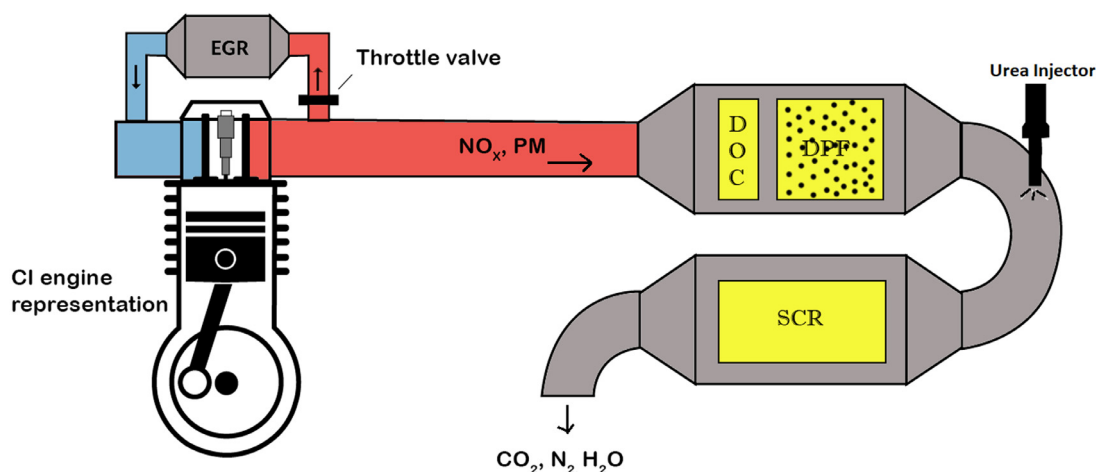


Fig. 10. Exhaust retreatment systems representation.

controlled by optimizing the multiple injection strategies (Zhou et al., 2021).

2.4. Modification of exhaust after-treatment (EGA) strategies

EGA systems are the ideal way to reduce engine emissions (Yoon et al., 2014). NO_x and PM emissions from biodiesel combustion are challenging aspects to control. The NO_x emissions include nitrogen oxide, nitrogen dioxide and nitrous oxide which form inside the cylinder at elevated temperatures. The oxygen content in the fuel and air reacts with nitrogen atoms at higher temperatures and forms NO_x emissions. In addition PM emissions including dry soot, smoke, the soluble organic fraction, metal-organic and inorganic matter, sulphates, water sediments, unburned fuel, and ash will be formed due to fuel accumulation, improper soot oxidation and poor fuel quality (Tan et al., 2019; Zhang et al., 2020c).

Fig. 10 illustrates the exhaust gas retreatment setup for the internal combustion engines. Firstly, the EGR flue gases are passed into the chamber through the intake manifold in controlled proportions by using the throttle valve. Further, the exhaust gases pass through the DOC and DPF to isolate the PM emissions and trap the other HC and CO emissions. Finally, the flue gasses enter the SCR catalyst, and NO_x emissions are converted here to N_2 , CO_2 and water (Apicella et al., 2020a).

2.4.1. Exhaust gas recirculation

EGR effectively reduces the oxygen concentrations inside the chamber because the exhaust tailpipe is connected to the inlet manifold through a throttle valve. Different percentages of the combusted gases will be recirculated into the combustion chamber from the intake manifold along with the fresh air. The terms hot EGR and cooled EGR are classified based on the recirculated gas temperature. If the exhaust gas is directed to the combustion chamber, then the EGR setup is termed hot EGR, and if the recirculated gas is pre-cooled, then the setup is cooled EGR (Zheng et al., 2004). The hot EGR system improves the combustion process and thermal efficiency. At the same time, cooled EGR enhances the volumetric efficiency of the system (Dubey et al., 2019). Another novel concept is reformed-EGR, where the recirculated exhaust gases are added with the reformed reactive gases. For instance, Du et al. (2017) observed a decrease in HC, CO and NO_x emissions by adding hydrogen to the EGR. Moreover, several studies revealed improved performance using the reformed EGR technique (Li et al., 2013; Long et al., 2021; Nguyen et al., 2019).

The combustion temperature gets minimized with the addition of EGR and decreases the NO_x emissions. The EGR contains CO_2 , water vapor particles, and inert nitrogen which reduce the oxygen levels in the intake air mixture (Jacobs et al., 2003; Rakopoulos et al., 2018). Thus, the EGR concentration levels and injection rates play a vital role in controlling diesel engine emissions. Saleh (2008) observed the effect of the EGR on the NO_x and CO_x emissions with LPG compositions and observed that, with the increase of EGR to 5%, NO_x emissions increased and CO emissions decreased. By further increasing EGR to 10%, the study noted a decrease in NO_x emissions but a significant increase in CO emissions. Ogawa et al. (2006a) performed combustion analysis with high ratios of cold EGR operations and noted the ultra-high reduction in NO_x , and noted changes in soot emissions with respect to the oxygen content. In the early days, EGR was introduced to reduce NO_x emissions; this technique was then applied to control combustion rates. For instance, the addition of EGR causes a longer ignition delay and combustion duration (d'Ambrosio et al., 2016). Combustion rates in LTC strategies, at high turbulent kinetic energy rates and with high reactive fluids, create peak pressure rates, high knocking, unregulated emissions and many more. The addition of EGR will create combustion phasing for better fuel mixings and avoids rapid combustion rates.

2.4.2. Diesel oxidation catalyst (DOC) and Diesel particulate filter (DPF)

DOC is the oxidation catalyst used for the oxidation of exhaust gases, and it is placed next to the exhaust tail pipe. The DOC helps oxidize the HC and CO emissions to H_2O and CO_2 and processes the NO emissions to NO_2 which will help with passive regeneration of soot oxidation in the DPF and also helps carry an efficient reaction mechanism with the SCR catalyst (Fayad et al., 2015; Herreros et al., 2014). The reaction mechanisms are shown in Fig. 11. DOC operating conditions require high temperatures for better oxidation and to consider the requirements; metallic catalysts are used to lift the temperatures. Platinum (Pt), Palladium (Pd) and Rhodium (Rh) are the commonly used DOC catalysts with strong base supports of Al_2O_3 , ZrO_2 (Hoang et al., 2019). The higher the reactivity surface area, the higher the rate of oxidation. Caliskan and Mori (2017) observed a 94% reduction in HC and a 99% reduction in CO emissions using DOC (Caliskan and Mori, 2017).

Another post-exhaust treatment uses DPFs to reduce the soot from the exhaust gases. These filters are termed as ceramic wall-flow filters with porous blocked cells at the end. Exhaust gases are directed into this porous medium to trap the PM emissions

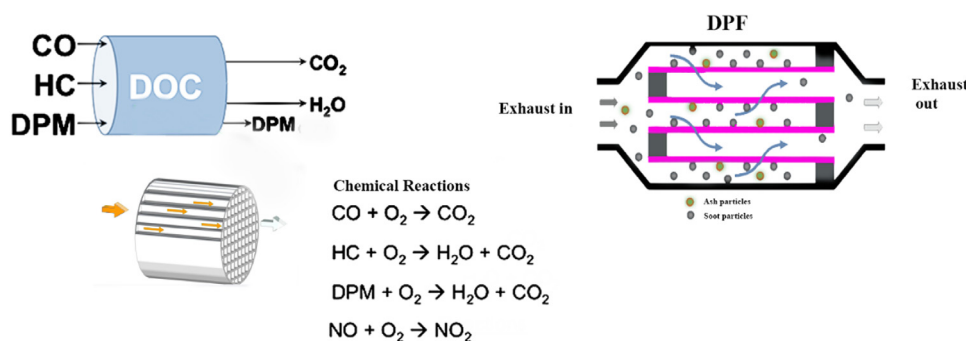
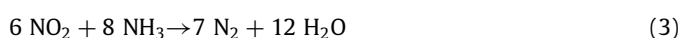
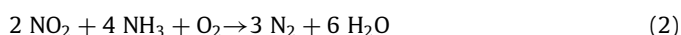
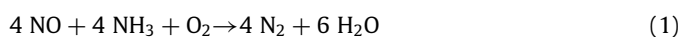


Fig. 11. DOC and DPF exhaust systems mechanism.

in the filter pore matrix. Dry soot accounts for 50% to 80% of PM emissions in diesel engines. Honeycomb and wall-flow guided monolith structures are the standard plugins for the particulate filters. In the DPF, soot accumulation creates exhaust pressure drop rates in the exhaust pipelines. The pressure drop rates are due to the accumulation of matter on the filter, which reduces the expansion work of the piston. If the exhaust back pressure drop rates are too high, the ramifications affect the combustion process, engine performance, filter/engine failure, additional $p - V$ work and high BSFC (Houston and Clyne, 2020; Millo et al., 2015; Piqueras et al., 2019; Yamamoto et al., 2009). Continuous regeneration and soot oxidation reduce the pressure build-up in the systems (Corro et al., 2019b). DPF soot regeneration involves composite, passive, and active regeneration (E et al., 2020). Amalgamated soot gets burned off at temperatures above 600 °C, but the exhaust gas temperatures for the diesel engine range from 200 °C to 500 °C (Fino et al., 2016). Continuous regeneration associated with the oxidation catalyst is categorized as the most effective approach for soot oxidation (Corro et al., 2019a). DOC acts as the continuous heating source for the DPFs and will reduce the CO and UBHC emissions (Zhu et al., 2013). The NO₂ traps formed during the DOC oxidation help the soot to oxidize at 300 °C (as regeneration for the DPF). DOC provides additional support to the DPF, to reduce the PM size and structure by oxidizing SOF (Tan et al., 2019). These post-combustion strategies are well suited to regulate the emissions; however, improvements are still needed in system setup and filter regeneration techniques for effective applications (Mohankumar and Senthilkumar, 2017).

2.4.3. Selective catalytic reduction

NO_x forms such as nitrogen oxide (NO), nitrogen di-oxide (NO₂), and nitrous oxide (N₂O) are generated during the combustion process and are responsible for air pollution, acid rains and photochemical smog and so on, which are damaging our environment and health (Zhu et al., 2020). In diesel engines, alternative fuel is creating a major drawback in the NO_x emissions, and numerous inventions are being progressed with different chemical catalytic processes to reduce the NO_x emissions. Generally, ammonia is used as the reactive agent for the conversion of NO_x emissions to nitrogen and water. The overall reactions associated with the NO_x emission conversions are shown in the following equations.



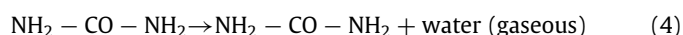
A three-way catalyst (TWC) was used in the early stages to reduce the CO, HC and NO_x emissions (Harold, 2012). Due to the drawbacks of TWCs at lean bursts, a lean NO_x trap (LNT) was introduced to limit NO_x emissions under lean conditions (Maurer

et al., 2017). Further, selective catalytic reduction (SCR) was developed for better reductions of NO_x emissions. Several integrated approaches are modeled with the TWC, LNT and SCR techniques to present the trade-offs between the systems. The potential benefits of these approaches are presented in Fig. 12.

The selective catalytic reduction (SCR) mechanism is followed upstream of the DPF/DOC and is subjected to high temperatures associated with the filter regenerations. SCR catalysts are aligned with the base substrates, and a respective catalyst material is washcoated onto these bases. Ceramic and metallic materials are commonly used substrates and materials such as Aluminum oxide (Al₂O₃), Zirconium dioxide (ZrO₂), Cerium oxide (CeO₂), Pt, vanadium oxide (V₂O₅), zeolites, lanthanum oxide, and titanium oxides are the washcoat materials used for catalytic reactions (Costa and Efstathiou, 2007; Zheng et al., 2005). There are several active catalysts for SCR reactions, and among them, noble metals, transition metal oxides, and vanadium–titanium catalysts are the most popularly used components (Worch et al., 2011; Zhao et al., 2020). These catalyst components are associated with the support carried for high deNO_x activity. TiO₂, Al₂O₃, zeolite, TiO₂, Al₂O₃, SiO₂ and carbons are some of the catalytic carriers used in SCR catalysts (Peng et al., 2015; Vittadini and Selloni, 2004).

Numerous studies have been conducted on different SCR catalysts that are suitable for high temperature conditions. Aqueous urea is widely used as the reductant due to the available ammonia content. Direct ammonia is unstable to store in that state, and it is stabilized with water (Apicella et al., 2020b). About 32.5% aqueous urea solution is categorized as the optimum level concentration. 1 mole of urea is proportionate to two moles of NH₃, and a clean-up catalyst is added to the system to reduce the excess ammonia. The decomposition of the aqueous urea to NH₃ takes place in three stages (Praveena and Martin, 2018). After spraying the urea solution over exhaust gases, evaporation, thermolysis and hydrolysis reactions take place instantly to isolate NH₃. Additional conversion of NO_x emissions to N₂ and water is processed with respect to Eqs. (1), (2) and (3).

1. Evaporation: the water particles in the aqueous urea solution evaporate to form the molten urea



2. Thermal decomposition:



3. Hydrolysis:



Despite the progress, some concerns are arising about using SCR for mobile applications. The currently developed SCR catalysts are of a powder type rather than monolithic solids. The catalytic activities show some differences from the laboratory test results to the industrial pilot tests (Wang et al., 2019a).

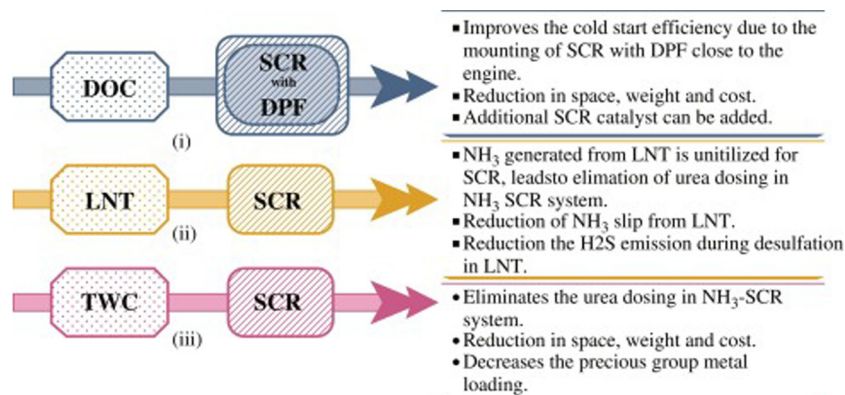


Fig. 12. Integrated approach for introducing the SCR with other EGA systems (Vignesh and Ashok, 2022).

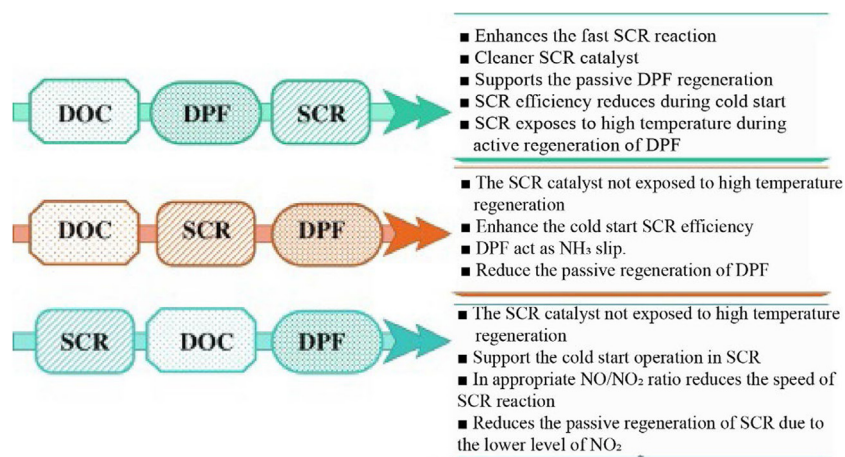


Fig. 13. Different configurations for EGA systems (Vignesh and Ashok, 2022).

Further, the cost of the catalyst implies the need for minimizing or removing some effective catalysts from use in the system (Shan et al., 2014).

2.4.4. Overview on EGA systems

To limit the pollutants emitting from the engines, the major response of the industries is to adopt the EGA systems. The widespread use of a catalytic (SCR) converter at the tail pipe connected with the DOC and DPF systems oxidizes and filters the incomplete combusted products. Several studies have been conducted on the positions of DOC, DPF and SCR systems on the exhaust line and those correlations are presented in Fig. 13. Automobile companies such as Mercedes, Volkswagen, Audi and many more have revealed that the upgraded EGA systems have been the major contributors in reducing emissions under Euro 6 limits. Also, research activities are currently going on with the dual SCR systems, electrically heated catalysts, catalyzed DPF, and close coupled DOC loops, to further limit the secondary emitting pollutants. The upgraded EGA systems with better regeneration techniques can face major challenging conditions (expected Euro 7 standards) (Mendoza Villafuerte et al., 2022). A closed DOC-DPF-urea-SCR catalyst EGA system has shown great potential in limiting the pollutants under Euro limits. Irrespective of the operating conditions, fuel and chamber-type EGA systems act as a passive technique to control harmful emissions.

2.4.5. Pros and cons of biodiesel combustion strategies

Table 2 presents the pros and cons of biodiesel combustion strategies, focusing on engine performance, combustion, and

emission characteristics. The table also considers engine characteristics with the combination of these advanced technologies.

3. Operational factors affecting the diesel engine combustion process

External operating factors play a vital role in performing the combustion process. The operating parameters affecting the combustion rates are described in this section.

3.1. Compression ratio (CR)

CR is the significant parameter that affects the combustion process through auto-ignition, the start of combustion and ignition delay (Muralidharan and Vasudevan, 2011). Many authors observed that the increase in CR revealed improved thermal efficiency with reduced emissions (Çelik et al., 2011; Gingrich et al., 2009; Sayin and Gumus, 2011). Fuels with a high octane number require higher CR to perform better combustion (Awad et al., 2018). Laguitton et al. (2007) observed the reduction in NO_x emissions with the decreasing CR at high loads. In addition, Jindal et al. (2010) carried out an experimental analysis with jatropa methyl ester biodiesel with different CR and derived that, with CR-18, BSFC is decreased along with increased BTE. For dual-fuel operating conditions, an increase in CR helps in enhancing the BTE of the system (Bora et al., 2014; Yoon and Lee, 2011). Furthermore, Bora et al. (2014) conducted an experimental analysis with a dual fuel (diesel and raw biogas) engine by increasing CR

Table 2
Comparison of the results with the blending of advanced technologies.

| Advanced techniques | Pros | | | Cons | | | |
|---------------------------------|---|--|--|---|---|---|---|
| | Performance | Combustion | Emissions | Performance | Combustion | Emission | |
| LTC | Higher engine efficiency (Benajes et al., 2014; Fridriksson et al., 2017) and Lower ISFC (Gao et al., 2013) | Better combustion control at low and medium speeds | 85% reduction in NO _x & 95% reduction in soot (Kocher et al., 2014), Reduced CO (controlled conditions) (Pandian and Anand, 2018; Salahi et al., 2017) | Increased BSFC, knocking behavior at high loads | Peak pressure rises, high HRR at high loads, longer ID | Unburned HC emissions, higher CO emissions (Kim and Bae, 2017; Liu et al., 2014) | |
| Chamber modifications | Lower BSFC (Li et al., 2017a), increased BTE and higher BSFC (Nayak and Mishra, 2019) | Better HRR, longer ID (Nayak and Mishra, 2019) | Decrease in soot (Li et al., 2017b), lower NO _x and soot at lower spray angle (Li et al., 2017a), decrease in HC, CO and smoke emissions (Annamalai et al., 2018; Singh et al., 2017) | Thermal load of cylinder head increases with poor fuel distribution | No change at low loads, uneven combustion design leads to unregulated HRR (Doppalapudi et al., 2021) | Increased soot and CO emissions (Sener et al., 2020) | |
| Injection pressure (increasing) | Similar BTE (Puhan et al., 2009), increase in BTE (Karthikayan et al., 2015) | Increase In ICP, HRR | Reductions in smoke, HC and CO emissions (Kim et al., 2016) | Increase in exhaust gas temperatures (Kannan and Anand, 2011) | Higher injection pressure causes poor combustion due to fuel wall wetting | Increase in NO _x | |
| Injection timing (retarded) | Improved BTE (Barik and Murugan, 2016; Purushothaman and Nagarajan, 2009), decreased BSFC (Barik and Murugan, 2016) | Increased ICP, high HRR | Reduced NO _x , HC, CO and smoke (Purushothaman and Nagarajan, 2009) | BSFC increased with advancing IT (Ganapathy et al., 2011; Sayin et al., 2009) | Delayed combustion, slow burning rate, shorter ID (Shameer and Ramesh, 2018; Sharma and Murugan, 2015) | Increased with advancing IT (Ganapathy et al., 2011) | |
| EGR | Lower EGR levels give improved BTE | Combustion control can be achieved | NO _x emissions decreases | Increased BSFC (Maiboom et al., 2008; Pradeep and Sharma, 2007) | Low HRR and peak cylinder pressures | Soot emissions increases due to the reactions with CO ₂ and H ₂ O (Ladommatos et al., 2000) | |
| LTC | Combustion chamber modifications | Higher BTE, Lower BSFC (Ganji et al., 2018) | Higher cylinder temperatures, shorter CD | Reduction in CO, HC, and NO _x (Salahi et al., 2017) | Higher BSFC due to delayed combustion rates (Benajes et al., 2015a) | Rapid HRR and pressure rise (Salahi et al., 2017) | Increased NO _x (Liu et al., 2018). Increased HC and CO at high loads (Halewadimath et al., 2022) |
| | Injection pressure | Enhanced BTE (Gowthaman and Sathiyagnanam, 2016) | High HRR, peak pressure rises | Reduced, HC NO _x and PM (Jacobs et al., 2005; Kiplimo et al., 2012) | Higher BSFC, higher EGT | Combustion control is difficult | Higher CO emissions (Kiplimo et al., 2012) |
| | Injection timing | Increased in IMEP noted with earlier injection (Park and Yoon, 2016) | Increased ID with retarded IT (Kimura et al., 1999) | NO _x (13%), THC (12%) and CO (27%) (Zehni et al., 2017), ~95% reduction in smoke (Zhang et al., 2012). | Increased BSFC with advanced IT (Benajes et al., 2015b) | Early injection causes wall impingements and incomplete combustion (Drews et al., 2011) | Higher PM emissions are noted at advancing IT (Misztal et al., 2009) |
| | EGR | High EGR levels reduce the engine performance | Combustion can be controlled at high EGR levels (Zhao et al., 2014) | Near zero NO _x emissions | High levels of EGR is required to control the ignition delay and cylinder temperature (Drews et al., 2011; Ying et al., 2010) | Double peak HRR observed at zero levels of EGR (Zheng et al., 2018) | Increased CO and HC emissions with increasing EGR levels (Bhiogade and Suryawanshi, 2016) |

(continued on next page)

Table 2 (continued).

| Advanced techniques | | Pros | | | Cons | | |
|-----------------------------|------------------------------|--|---|--|--|---|--|
| | | Performance | Combustion | Emissions | Performance | Combustion | Emission |
| Chamber modifications | Injection pressure | Increase in thermal efficiency | Increased ICP | Higher NO _x , decreased soot (Li et al., 2017a) | Decreased in BTE (increasing spray angle) (Sener et al., 2020) | Shorter ID and causes sudden heat release rates | High NO _x emissions are noted |
| | Injection timing (retarding) | Lower BSFC for biodiesel, Improved BTE (Jaichandar and Annamalai, 2012; Jaichandar et al., 2012) | Shorter ignition delay, maximum heat release rate (Jaichandar et al., 2012) | Reduced UBHC, CO emissions with advancing IT, decrease in NO _x at retarded IT (Jaichandar et al., 2012) | Higher BSFC than diesel (Jaichandar et al., 2012) | Reduce in peak pressures (Jaichandar et al., 2012) | Increased CO, HC emissions (Jaichandar et al., 2012) |
| | EGR | Increase BTE at controlled EGR | Shorter ID at low EGR levels | Reduced CO, HC and NO _x emissions at all loads (Srikanth et al., 2020), reduced NO _x at low EGR levels (Praveena et al., 2020) | BTE decreases at high EGR levels | Decrease in cylinder pressure, HRR with EGR (Praveena et al., 2020) | Increased NO (Praveena et al., 2020) |
| Injection pressure | Injection timing | Increased IP and late IT has showed increased BTE and IMEP (Kiplimo et al., 2012) | Better heat release rate is observed with optimized IP and IT | Reduced NO, smoke obtained with adjusting IP and IT | Higher BSFC | ID is sensitive parameter while working with | Several emissions trade off was happened at varying IP and IT |
| | EGR | Increased BTE noted with increased IP at low EGR levels (Kiplimo et al., 2012) | Better combustion at low EGR levels | Reduced NO _x . At low EGR levels reduced CO emissions are noted | Higher BSFC | High EGR, dominates the effect of IP on combustion rates | Increased soot |
| Injection timing (retarded) | EGR | Reduce in BTE (Qi et al., 2011) | Peak HRR at high EGR levels, more homogeneous mixture formation (Qi et al., 2011) | Reduced NO _x (Abd-Alla, 2002; Saleh, 2009), reduced soot | Reduced BTE (Bari et al., 2004), Increased BSFC | Lower cylinder temperatures, increased ID | Increased smoke (Abd-Alla, 2002), soot increased at high EGR (Qi et al., 2011) |
| EGR | DOC-DPF-SCR | BTE depends on the EGR rate | EGR levels effects on combustion rate | Reduced CO, HC, NO _x , PM and smoke emissions | Higher BSFC, regeneration effects engine back pressure | Higher EGR levels has showed delayed combustion | Less emissions at high cost |

from 16 to 18 which resulted in an increase in BTE and a decrease in CO and HC by 26.22% and 41.97% respectively. Also, NO_x and CO₂ emissions were increased by 66.65% and 27.18% respectively with the same change of compression ratio. Gnanamoorthi and Devaradjane (2015) observed an increase in combustion performance with an increase in CR. Those authors reported an increase in BTE and reduction in HC and CO emissions with the increase in CR when a direct injection engine was operated with diesel-ethanol blends. Kassaby and Medhat (2013) suggested that the increase in CR for biodiesels revealed excellent results compared to diesel fuel. The author conducted experiments with waste cooking oil and observed an increase in BT by 27.5% for B20 and a decrease in CO and HC by 37.5% and 52% with the increase in CR from 14 to 18. Moreover, the ignition delay period was decreased by 13.95% from CR 14 to 18. A rated amount of smoke opacity reduction was observed for biodiesel with the increase in compression ratios. This was due to the oxygenated contents of the biodiesel blends (Hirkude and Padalkar, 2014; Murayama et al., 2000; Yadav et al., 2017). Dempsey and Reitz (2011) observed increased BT and reduced soot emissions for RCCI combustion at 16:1 CR and also derived the optimized CR of 11:7 for the RCCI engine through computational modeling. The CR must be in the limits of the auto-ignition range, because lower and upper limits of CR lead to the futile combustion and engine knocks.

3.2. Auto-ignition temperature

Auto-ignition is the process at which the fuel gets ignited simultaneously with successive temperatures. The combustion process begins with the auto-ignition regime, and then the flame propagation regime increases corresponding to the surrounded fluid particles (Borghesi et al., 2018; Fukushima et al., 2015). Auto-ignition is associated with the sudden heat release rates inside the cylinder that causes knocking (John, 2018). Biofuels have different physio-chemical properties. Gasoline/ethanol fuels have a lower cetane number that resists the auto-ignition with direct injection (Haas et al., 2009; Zhang et al., 2013).

On the contrary, due to high CR, gaseous fuels (hydrogen, biogas and many more) have revealed pre-ignition characteristics at the end of the compression stroke (Bora and Saha, 2016). The partially premixed combustion has a set of challenges for controlling the combustion phase and exhibits longer ignition delay with lower cetane-numbered fuels (López et al., 2014). Moreover, it is difficult to understand the auto-ignition behavior due to the stratification of fuels that are injected through multiple injector holes (Du et al., 2020). Wang et al. (2019b) studied the auto-ignition behavior of methanol and diesel methanol fuel (DMF). The authors pointed out that the auto-ignition temperature of methanol is higher than the conventional CI operating

conditions, but SOC occurred with the DFM after the injection of diesel spray into the combustion chamber. Finally, the authors noted that EGR significantly affects the auto-ignition process; with the increase of EGR, incomplete combustion of methanol was observed. [Dong et al. \(2018\)](#) conducted a combustion experiment on an ethanol/diesel dual-fuel engine and observed auto-ignition retardation. The authors stated that this retardation is due to the dilution of diesel fuel with ethanol concentrations, which in turn reduces the global reactivity of the blends. Also, the charge cooling effect on the ethanol vaporization affects the cylinder pressure at near TDC that leads to a rise in in-cylinder temperature near TDC.

3.3. Charge or air-fuel (A/F) ratio

The fuel-air mixture substantiates the combustion process and solely impacts the engine working behavior. The air-fuel mixture quality controls the ignition and it is difficult to manage the premixed/diffusion combustion phases as these directly influence the efficiency and emissions ([Taghavifar et al., 2021](#)). The combustion with the lean fuel-air mixture at premixed rates resulted in near-zero NO_x and PM emissions ([Murugesu Pandian and Anand, 2018](#)). A lean mixture causes an incomplete combustion process and tends to emit HC, CO, and soot emissions. In contrast, the fuel-rich mixture creates a shorter combustion duration, elevating the higher temperatures and oxidizing the carbon and oxygen inside the chamber. Charge in the combustion is processed through stratified lean injection and homogenous charge stratified injection. In stratified lean injection, fuel is injected directly into the combustion chamber with high pressures. With homogeneous combustion, the air/fuel mixture charge is enveloped during the intake stroke. Excess air inside the chamber results in lean mixture formation and tends to exhibit less work output. Insufficient oxygen inside the chamber performs unstable combustion that causes longer combustion durations and leads to misfiring. Charge dilution limits the throttling, pump and heat transfer losses compared to the stoichiometric mixtures ([Birtok Baneasa et al., 2017](#)). Moreover, lean fuel mixture engines work at higher compression ratios with fewer tendencies to knock ([Wang et al., 2017](#)).

Higher fuel injection quantities tend to release more heat energy and high-rated power. A lower equivalence ratio mixture reduces the UBHC and CO emissions, and a higher equivalence ratio leads to higher HC and CO emission ([Flowers et al., 2006](#)). In addition, the air-fuel ratio also depends on the type of fuel. Gaseous hydrogen and fossil fuels require more air for better combustion than alcohol blends because the stoichiometric air-fuel ratios to these fuels are 30:1, 15:1, and 10:1. With the EGR ratio, the charge combustion depends on the oxygen content rather than on the air-fuel ratio ([Ogawa et al., 2006b](#)).

4. Comprehensive response of state-of-the-art combustion strategies

The effectiveness of the engine performance, combustion and emission characteristics is the response to the in-cylinder combustion phenomenon. Injection strategies, combustion chamber modifications and LTC techniques have indicated that enhanced combustion achieves better engine characteristics. However, the air-fuel mixture formation and combustion control are quite critical to understand with these approaches. The turbulence from the combusted fumes is proven dominant inside the cylinder and causes uneven combustion and unregulated emissions. Various injection strategies and chamber modifications seem to be an advantageous pathway to conduct further research on using proper

fuel distribution and combustion. Also, these techniques need significant progress in methodological reinforcements aligned with engine operating conditions, engine chamber design, and fuel support systems. With appropriate consideration of these issues, there can be a way to perform the combustion at lean mixtures in creating low-temperature combustion at varying engine operating parameters. In addition, new models such as EGA systems have been studied as additional supports to the diesel engine in reducing emissions. Enhanced developments from the combustion perspective should guide the opportunities and applications of all these cases.

5. Conclusion

As discussed above, higher physical properties of density and viscosity and increased NO_x emissions are the major drawbacks to biodiesel combustion engines. Making slight changes to the biodiesel operating phenomenon has revealed better performance and reduced emissions. The conclusions derived from the study are as follows.

- (1) Better combustion with heavier biodiesel will be achieved by adjusting the IP, piston bowl geometries and injection sprays. At the same time, higher NO_x is noted under these conditions, which can be controlled with EGA systems.
- (2) Homogeneous combustion is noted with the LTC strategies, manifold modifications and retardation of IT, which also helps reduce NO_x emissions at leaner mixtures. However, these strategies require higher CR for performing the combustion. The oxidation rates of HC and CO emissions will be low at lean mixture conditions and using DOC systems can avoid these emissions.
- (3) Injection strategies provide better control of the combustion but lack better combustion distribution. On the other hand, combustion chamber modifications will give better support for the combustion even under lean mixture conditions. The turbulent motions generated inside the chamber avoid the wall wetting and distributes the air-fuel mixture uniformly around the squish and crevice regions and offers better combustion.

Several passive techniques are discussed in this study. The study concludes that the leaner mixture with high turbulence rates inside the chamber can achieve better combustion with biodiesel fuel.

6. Scope of future work

LTC strategies showed significant reductions in NO_x, but these engines require higher CR, and also combustion rates are hard to control at different operating ranges. But the idea of leaner mixture combustion in achieving the low-temperature atmospheres inside the chamber has to be noted for on-road applications with the additional support of injection strategies, chamber modifications and EGA systems. The study recommends that greater focus is needed while adjusting the equivalence ratio along with optimizing the injection rates and chamber modifications. Many gaps can be addressed with these parameters with different biodiesel fuels.

Along with the above discussion, several active techniques infer further potential improvements in reducing biodiesel emissions, such as fuel blends with alcohols and other fatty acid esters, fuel emulsion and fumigation techniques. These techniques also showed reductions in NO_x emissions with biodiesel.

Declaration of competing interest

The authors declare that they have no known competing financial interests or personal relationships that could have appeared to influence the work reported in this paper.

Data availability

No data was used for the research described in the article

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