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PCCI combustion of low-carbon alternative fuels: a review

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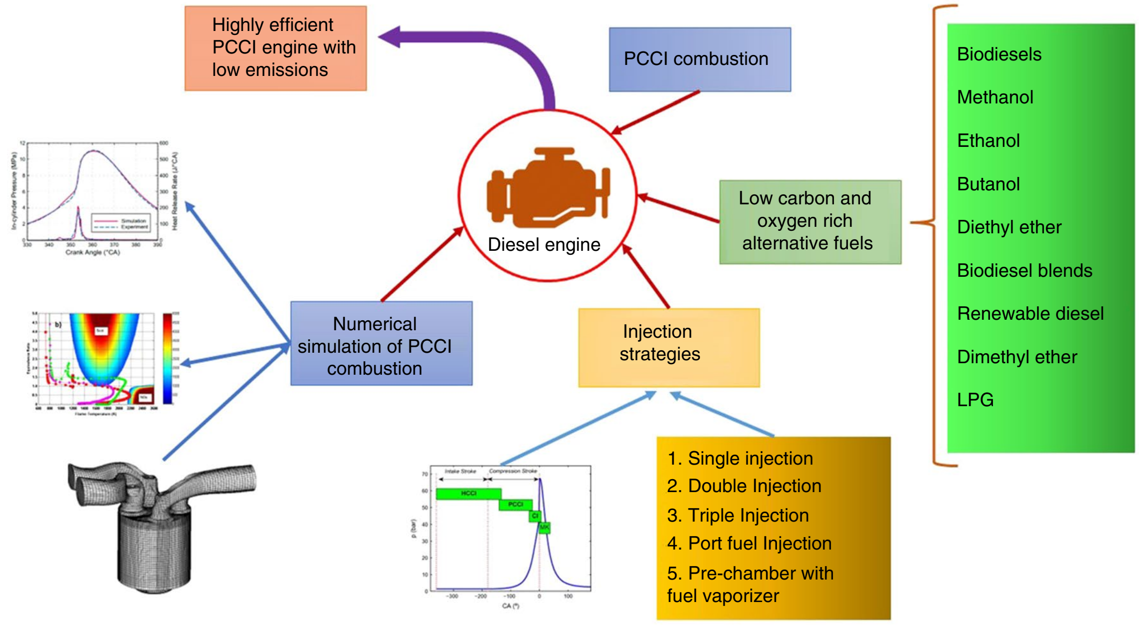
Abstract

Low-temperature combustion in diesel engines gained prominence because of their ability to meet the current emission standards without NOx and PM trade-off. Among the low-temperature combustion concepts, premixed charge compression ignition (PCCI) offers an in-cylinder emission reduction with minimal to zero engine modifications. This work reviews the role of premixed charge compression ignition (PCCI) of low-carbon and oxygen-rich fuels on diesel powertrains' performance. This review covers the fundamentals and significance of PCCI combustion with low-carbon oxygen-rich fuels of both renewable and synthetic origins. Various strategies employed for achieving PCCI combustion, in-cylinder, and external charge preparation are discussed in this review. The effect of a single injection, multiple split injections, injection pressure, and injection duration on PCCI combustion in diesel engines is discussed at length. Low-temperature combustion depends on the chemical kinetics of combustion. The present review discusses the numerical works carried out with detailed chemical kinetics of various conventional and alternative fuels. Challenges in PCCI combustion, such as wall-wetting in early direct injections, combustion phasing, narrow load range, and engine knock for conventional and unconventional fuels, are presented. Bottlenecks in the present PCCI technology, advantages of using alternative fuels for PCCI combustion, and the scope of future work are presented at the end of this review.

Graphic abstract



Keywords PCCI combustion · Emissions · Combustion · Alternative fuels · Low-carbon fuels

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Introduction

The significant issues in the current automobile sector are emissions, soaring fuel prices, and stringent emission norms. These issues are challenging to both industry and researchers around the world. The present transportation sector requires fuel-efficient engine technologies that would reduce emissions and satisfy the present and future energy security requirements. The diesel engines are fuel efficient compared to gasoline engines but would require costly after-treatment systems to make them compliant with emission standards. Incentives and subsidies are provided to electric vehicles to achieve the goal of clean transportation. However, due to their limited range, these measures have not resulted in the expected transformation of vehicle technology from internal combustion engines (ICE) to electric vehicles (EVs). [1]. The assessment of well-to-wheel emissions of BEVs found that greenhouse gas GHG emissions from battery electric vehicles depend on the source of electricity used for charging the EVs [2–4]. In this scenario, advanced combustion concepts, such as homogeneous charge compression ignition (HCCI), premixed charge compression ignition (PCCI), and reactive charge compression ignition (RCCI) are becoming significant. These combustion concepts can simultaneously reduce emissions and improve the performance of the engines for conventional and alternative fuels [5–8]. Among these combustion concepts, PCCI has the advantage of minimum to zero engine modifications. PCCI combustion of low-carbon fuels in diesel engines has much potential to reduce emissions and improve the PCCI range of operation. So, the present work reviews significant research on the principles of PCCI Combustion, low-carbon fuels, the effect of fueling and exhaust gas recirculation (EGR) strategies on PCCI combustion, challenges, and the scope for PCCI technology in the commercial vehicle fleet.

Fundamentals, methodology and challenges of PCCI technology

Fundamentals of PCCI technology

In conventional diesel combustion systems, soot formation occurs at the end of the delay period due to rich combustion. Thermal NOx generates at high-temperature zones during the mixing-controlled combustion phase [9–11]. In diesel engines, both scenarios frequently result in high NOx and soot emissions. In-cylinder emission reduction technologies are employed by many researchers to limit NOx and soot emissions, such as retarded injection timings, high injection pressures, and EGR. Nevertheless, these methods require costly after-treatment devices such as selective catalytic reduction (SCR), lean NOx traps, and particulate filters to meet the current emission standards [12]. To further reduce the NOx and soot emissions to the required levels without NOx-soot trade-off, overall combustion temperatures are reduced using low-temperature combustion (LTC) concepts such as HCCI, PCCI, and RCCI. All these concepts follow typical combustion paths which avoid NOx and soot islands on equivalence ratio and temperature (*Φ-T*) maps [13–15] as shown in Fig. 1. It is possible to reduce NOx and PM emissions to ultra-low levels using HCCI combustion. However, higher heat release rate because of premixed combustion, decrease in ignition delay with the increase in load, complex combustion control, and narrow load range are some significant issues in HCCI combustion [16–19]. Hence, PCCI combustion is adopted as a more reliable and practicable way of reducing NOx and PM emissions.

PCCI combustion is a midway between HCCI and conventional combustion. Unlike HCCI combustion, which often necessities external mixing and very early injections, PCCI combustion is achieved by employing early injections, late injections, high fuel injection pressures, and EGR [20–23]. In early PCCI, combustion occurs before TDC; in late PCCI, combustion happens after TDC. Even though both PCCI concepts rely on ignition delay, early PCCI is more advantageous than late PCCI in controlling emissions and maintaining high thermal efficiency [24]. Despite its advantage, PCCI combustion may also increase emissions due to early injections and a high percentage of EGR at higher loads. Temperature increases with the increase in load, and high amounts of EGR (~>30%) are required to increase the ignition delay at higher loads for sustaining

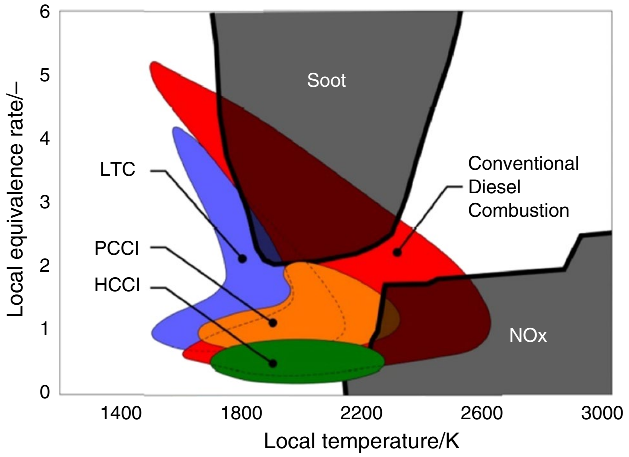

Fig. 1 *Φ*-T maps illustrating low-temperature combustion pathways [13]

PCCI combustion. This will penalize the engine with a rise in specific fuel consumption and a drop in thermal efficiency. Hence, critical control of engine parameters is required to take full advantage of PCCI combustion [7, 8, 16]. Discussion on various injection strategies, EGR, and combustion phasing strategies used for sustaining and controlling PCCI combustion are presented in the below section.

Methodology of PCCI combustion

As discussed in the earlier sections, early single or multiple split injections coupled with EGR can achieve PCCI combustion. In this section, various strategies utilized for early and late PCCI combustion are discussed. In the works of shim et al.[16], late PCCI combustion is achieved in a 2.0 L Euro V engine using pilot and main double injection strategy. The pilot injection timings are swept between 11 and 31° before top dead center (BTDC), and main injection timings are maintained between − 2 and 7° BTDC. To sustain late PCCI combustion, dwell time between main and pilot injections is longer with 0–50% EGR. The CO and HC emissions mainly arise due to very early injection pilot-main and pilot-main-post-injection strategies. A sweep of pilot injection timings from TDC to ~70° BTDC revealed a rise in CO and HC emissions earlier than 50° BTDC. To limit CO and HC emissions, pilot injection timings are maintained between 50 and 60° BTDC. Pilot injection quantity also plays a critical role in forming CO and HC emissions, so the pilot injection quantity is limited to 20% of the total fuel injected [10]. In addition to in-cylinder charge preparation, the premixed charge is prepared by advancing the injection timing till suction and injecting the fuel into a chamber with a fuel vaporizer. The premixed charge is fed into the engine during suction stroke [6]. The efficacy of single and multiple split injection for premixed combustion is studied using an optical engine and a metal engine. The injection timing of − 25° BTDC is maintained for a single injection with 45% EGR and − 13.6° BTDC with 0% EGR. In the case of the double injection strategy, the dwell time between main and pilot injection varies from 5 to 18°, depending on the EGR ratio. Pilot-pilot-main injection strategy is also utilized for premixed combustion with injection starting from 60° BTDC, and the highest efficiency of 48.5% is observed [25]. The single injection strategy for achieving PCCI combustion is simple and less complex when compared with double or triple injection strategies. However, the peak in unburned species and high-pressure rise rates are of significant concern in the single injection strategy. In this regard, Horibe et al. [26] tested both single and double injection strategies on a single-cylinder engine. In a single injection strategy, the injection timings are varied from − 20° after top dead center (ATDC) to 0° ATDC with different injection quantities and %EGR. In the double injection strategy, first injection varies from − 20° ATDC to − 30° ATDC and second injection varies from − 5 to 5° ATDC along with EGR. The experiments show that retarding the injection timing beyond 5° ATDC affects thermal efficiency adversely. Hence, trade-off is required between the second injection and peak pressure rise rates. The advancing first injection is needed for lower smoke emissions, but unburned species concentration may peak. Therefore, judicial selection of operating points such as injection timings, quantities, and EGR rates are crucial for premixed combustion [5, 10, 26]. Much research was conducted to determine the effect of EGR on PCCI and advanced combustion concepts [5, 25–28]. PCCI combustion relays much on EGR for increasing ignition delay, combustion phasing, and for decreasing the heat release rate. Experiments are carried out with and without EGR for various injection timings to assess its effect on PCCI combustion. Results indicated that late injections without EGR performance are better than with EGR. As the injection timing advanced, % EGR played a crucial role in improving indicated mean effective pressure (IMEP) and indicated thermal efficiency [29]. On top of injection timings, pressures, and duration, spray included angle plays a crucial role in PCCI IMEP. As previously described, PCCI combustion requires early injections, resulting in decreased IMEP if the spray hits the cylinder liner (wall wetting). The collision of spray in the piston bowl would concentrate all the mixture and increase IMEP and knocking. Both phenomena are undesirable, and making the spray collide with the upper piston bowl zone is essential for improved IMEP. Numerical and experimental studies conducted on the effect of spray included angle on IMEP by Kim et al.[30] show that 100° spray included angle has improved performance. The present section discusses various injection and EGR strategies employed to realize PCCI combustion. The set of engine operating points required for successful PCCI combustion collected from literature is given in Table 1. The various injection strategies utilized for premixed charge compression ignition and corresponding ranges of injection timing are shown pictorially in Fig. 2.

The strategies used for PCCI combustion of convention and low-carbon fuels such as biodiesels, ethanol, methanol, and their blends are reviewed. It is identified from the research survey that the same strategies which are applied for conventional fuels can also be used for low-carbon fuels. However, adding low-carbon fuels such as ethanol to biodiesel and diesel increases the ignition delay [28]. The reports say that for PCCI combustion of biodiesels such as rapeseed oil methyl ester, higher ignition delays are required owing to their high reactivity [33]. The tests on PCCI combustion of fuels such as biodiesel, dimethyl ether (DME), alcohols, ethers, and their blends show that injection pressures, EGR, injection timings, injection quantities, and compression ratios are vital for effective control of PCCI

Table 1 Methodologies used for PCCI combustion

References	Engine specifications		Injection strategy(SI: Single injection; DI: dual injection; TI: triple injection; MI: multiple injections)							Injection points (CA oBTDC) PI: pilot injection; MI: main injection; Po. I: Post-injection)			EGR(%)
	Compression ratio	Type	Spray Included Angle	Injection pressure	Pre Chamber	SI	DI	TI	MI	PI	MI	Po. I	
Natarajan et al. [6]	17.5	CC:661.5 RPM: 1500			√					18			
Fang et al.[28]	16.8	CC: 4.75 L RPM: 1450				√				7.5~5			30
Yin et al. [25]	18	CC: 12.74 L				√	√	√			25	17.8	45.4
										33	28		
										61	26		
Horibe et al. [29]	15.5	CC: 857		120 MPa		√	√			− 20 to 30	20 ~ 0	–	55 to 25
											5 ~ + 5 ATDC	–	
Jain et al. [31]	17.5	CC: 510.7 RPM:4200		400 to 1000 bar		√					12 ~ 24		15
Jung et al. [27]	17.4	CC: 980 RPM: 1200		160 MPa							25		60
Kanda et al. [23]	16.7, 15.5	CC: 550 RPM: 1500		70 MPa		√					12 ~ 56		0–54
Kim et al. [30]	17.5	CC:624 RPM: 1400	150° to 70°	100 MPa		√	√	√			65	55	–
										70	60		
										75	65		
Kim et al. [32]	17.3	CC:1991 RPM:1400		1600 bar			√			70 to 45	5 ~ 10 ATDC	–	–
Kiplimo et al. [29]	13	CC:781.7 RPM: 1000	140°	80 MPa,140 MPa		√					2 ~ 40		0–40
Kook et al. [9]	18.9	CC: 498 mm^3 RPM: 800	150°	30 to 120 MPa		√	√				250 ~ 50	20 ~ 0	
			100°								250 ~ 50		
Pandey et al. [21]	17.5	CC:1329 RPM:	150°, 90°	180 MPa					√				60
Mancaruso et al. [33]	16.5	CC: 522 RPM: 1500	DI:148° PFI: 14°	MI:80 MPa, PFI: 3.5 bar		√					6		
Mohammadi et al. [34]	17.8	CC:857.54 RPM: 1800	150°	120 MPa		√	√				60	7.5 ATDC	0–25
											60		
Pandian et al. [35]	17.5	CC: 662 RPM: 1500		0.3 MPa		√					40 ~ 50		10–40
Neely et al. [12]	18	CC: 499 RPM: 4000					√			22 to 38	4 ~ 6		

Table 1 (continued)

References	Engine specifications		Injection strategy(SI: Single injection; DI: dual injection; TI: triple injection; MI: multiple injections)							Injection points (CA oBTDC) PI: pilot injection; MI: main injection; Po. I: Post-injection)			EGR(%)
	Compression ratio	Type	Spray Included Angle	Injection pressure	Pre Chamber	SI	DI	TI	MI	PI	MI	Po. I	
Park et al. [36]	16	CC: 497 RPM: 1200	150°	75 & 120 MPa		√					0~80		0–55
Park et al. [37]	16	CC: 498.5 RPM: 1200	150°	55 MPa				√			28	10 ~ 40 (°ASOI)	0–60
											28	10 ~ 30 (°ASOI)	
											28	5 ~ 30 (°ASOI)	
Naoki et al. [38]	16.5	CC: 1298	125°	120 MPa, 90 MPa		√					25~6		0–60
Simescu et al. [39]	16	CC: 12 L RPM: 600 ~ 1668		200 MPa	PFI 020%	√					− 3~11°ATDC		4 ~ 50
Singh et al. [40]	17.5	CC:540.9 RPM: 4200		0.4 ~ 100 MPa			√			30 ~ 40	12~24		15
Soloiu et al. [41]	16	CC:1.132L RPM:1400			PFI								20
Srihari et al. [7]	18	CC: 395 RPM: 3600		0.18 MPa	PFI @3.5 bar		√						
Sun et al. [42]	15	CC:425 RPM: 1500		70 MPa		√					18~0		25 ~ 40
Torregrosa et al. [43]	18	CC:389.9 RPM: 1500	150°	80 MPa		√					38~18		10 ~ 14.5 $[O_2]_{IN}$
Valentino et al. [44]	17.5	CC:1910 RPM: 2500	148°	100 ~ 160 MPa		√					20~0		19.0,19.5$[O_2]$
Venegas et al. [45]	18.3	CC: 1900 RPM: 2000	100° ~ 148°			√					50 ~ 5		15%,30%,45%
Ying et al. [46]	10.7	CC: 1092 RPM: 2200		15 MPa			√			DME22.8, 26.8,30.4 mg/cyl	12		
Xu et al. [47]	17	CC:2124 RPM: 1200	120°	80 MPa		√					30 ~ 15		
Singh et al.[48]	17	CC: 510.7 RPM: 4200		50 MPa			√			35	12		

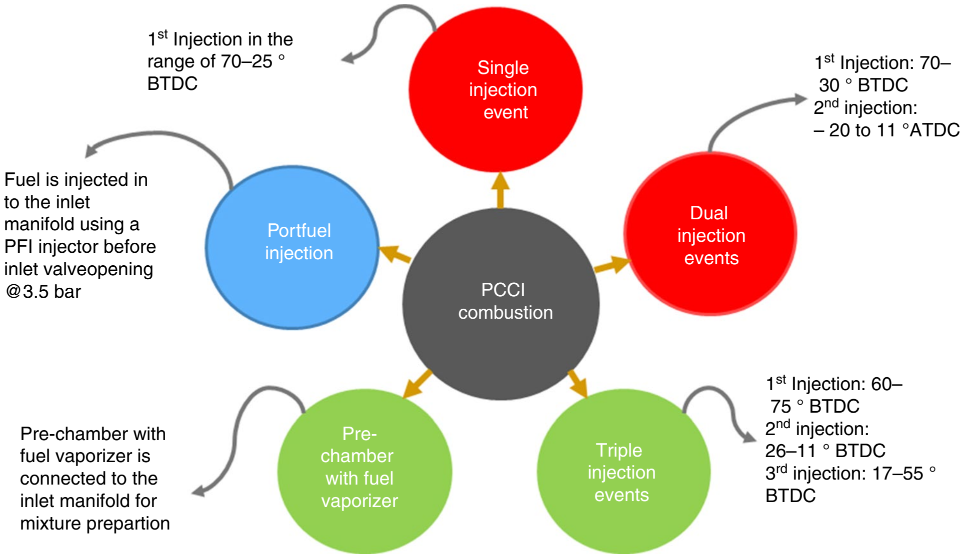

Fig. 2 Pictorial representation of various PCCI combustion strategies

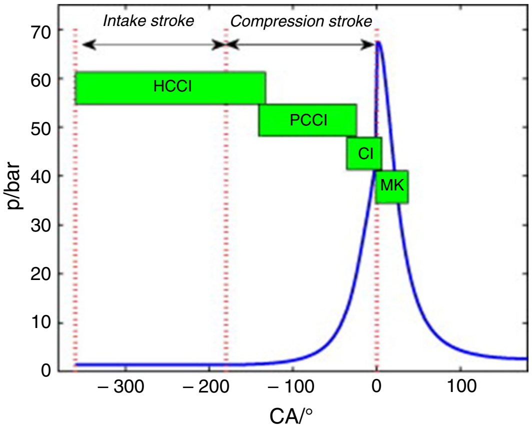

Fig. 3 Representation of different combustion concepts on pressure trace [49]

combustion. Some literature also cited injection nozzle geometries as crucial parameter for premixed combustion [9, 21, 45]. Specific injection timings for different combustion concepts are represented on a standard pressure trace and are shown in Fig. 3.

In summary, PCCI combustion is successfully achieved by many authors with the strategies exhibited in Fig. 2. Although PCCI combustion effectively reduces NOx and soot emissions, it is still tricky to engineer premixed combustion at different loads and speeds. The various challenges in PCCI combustion are discussed in the subsequent section.

Challenges in PCCI combustion

Despite its advantages, PCCI combustion can pose severe technical challenges regarding combustion control, performance loss, and emissions. Various challenges encountered during PCCI combustion are discussed in the present section. As a consequence of early injections, PCCI combustion is prone to higher knocking and combustion noise when compared with conventional baseline combustion [48, 50]. Lower compression ratios are preferred for PCCI combustion to reduce this combustion noise and avoid very advanced ignition [23, 42, 51]. This reduction in compression ratio results in lower efficiencies. Premixed combustion engines require high EGR rates to prolong the ignition delay. These high EGR rates are usually 0–60% and are beneficial for reducing NOx emissions. Nevertheless, they have determinantal effects on thermal efficiency, fuel consumption, HC emissions, and soot [16, 48]. Ambrosio et al.[22] reported a rise in brake-specific fuel consumption when EGR rates are greater than 20%, especially at low loads. In early injection PCCI, due to premixed combustion, it is not easy to control the combustion phasing (CA50). To maintain the combustion phasing, thermal efficiency PCCI must depend upon high EGR rates (>40%) and optimized multiple injection strategies. PCCI combustion with 30% EGR resulted in high HC and CO emissions. High injection pressures are used in PCCI combustion mode to achieve better fuel premixing. Higher injection pressures (>1000 bar), along with advanced injection, lead to inferior combustion and knocking [16, 31]. As the fuel is injected into low pressure and temperature atmosphere, wall-wetting, higher spray penetration, and low vaporization are encountered in early injections. The increase in ignition delay in the case of PCCI combustion generates over lean mixtures and lower cylinder bulk temperatures. The above factors lead to lower efficiencies and increased CO and HC emissions [52–55]. Narrow cone angle injectors are suggested by Chen et al.[56] to reduce wall wetting hence HC emissions. The experimental test shows that the injection advance timing limit is increased from 30° BTDC to 70° BTDC by using narrow cone injectors with 60° spray cone angle than conventional injectors with 145° spray cone angle. One more method suggested to reduce the wall wetting is to inject the fuel multiple times. These multiple injections would result in lesser spray penetrations, hence low HC emissions [57].

PCCI combustion has comparable efficiencies with conventional combustion and lower NOx and soot emissions. Despite these advantages, the operation of PCCI at higher loads is still challenging. The gain of lower soot and NOx emission at low loads is lost at higher loads due to increased temperatures. The rise in equivalence ratios, peak pressure, and pressure rise rates is the bottlenecks in PCCI combustion. Small quantities of post-injection can reduce the increase in soot emissions at higher loads (> 10 bar BMEP). However, post-injections have resulted in a rise in specific fuel consumption [10, 24]. PCCI combustion is inferior to conventional combustion in a few accept such as load range, HC and CO emissions [58]. However, it is still a clean and highly efficient combustion system with low NOx, soot, and comparable thermal efficiencies. Fuel properties play a significant role in extending the load range of PCCI combustion. Fuels with optimum cetane number (~40) and low distillation temperatures increase the load range to medium loads for PCCI combustion. Optimized injection strategies, fuels, narrow-angle injectors, compression ratios, and supercharging can improve the load range and other problems associated with PCCI combustion [28, 59–64]. Various challenges encountered in PCCI combustion and technical solutions suggested in the literature are shown in Fig. 4. Fuel characteristics play a vital role in the performance of a PCCI combustion. The discussion on various low-carbon and alternative fuels utilized for PCCI combustion and advantages of PCCI combustion are presented in the below sections.

Advantages of PCCI combustion

PCCI combustion offers some key advantages over conventional combustion. The primacy of PCCI combustion over other emission treatment methods used with conventional combustion are shown in Table 2.

Alternative fuels for PCCI combustion

Fuel characteristics are essential for any combustion phenomena, and their role becomes increasingly important if the combustion is kinetics dependent, like PCCI combustion. As discussed in the earlier sections, PCCI combustion necessities longer ignition delays for homogenization of charge, which requires low cetane fuels. The various fuel properties necessary for effective PCCI combustion and the performance of different low-carbon fuels and their blends on PCCI combustion are discussed in the present section. The reactivity of the mixture reduces by blending waste cooked oil biodiesel with diesel. The blends increased PCCI combustion's ignition delay and efficiency [41, 72]. Ethanol

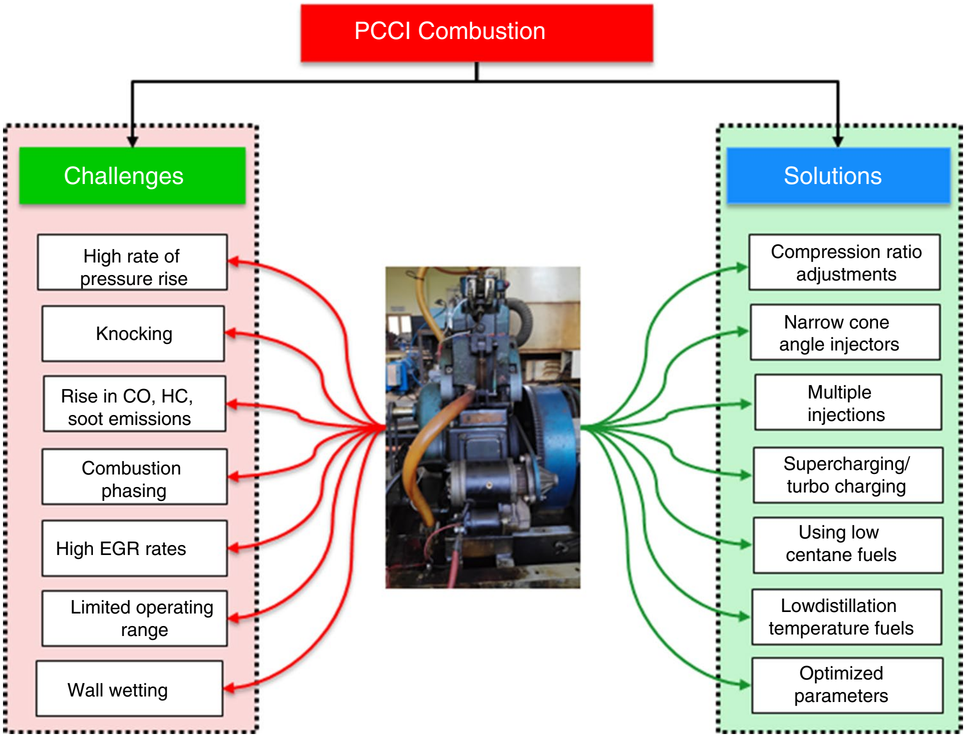

Fig. 4 Various challenges and technical solutions to PCCI combustion

Table 2 Advantages of PCCI combustion

Emissions treatment method	Objective	Limitations with conventional combustion	PCCI advantage
Exhaust gas recirculation [65]	Decrease NOx emissions	Risk of increase in fuel penalty and soot emissions	Can decrease soot emission with minimal fuel penalty
Retarded injections [66]	Decrease NOx emissions	Increase in soot generating precursors	Decrease NOx and soot simultaneously
Intake boosting [67, 68]	Decrease emissions	At high loads increases NOx emissions	Decrease CO, and UHC emissions along with NOx
Alternative fuels [69]	Improve performance and decrease emissions	Fuels like biodiesel increase NOx emissions	Decrease NOx emissions to ultra-low levels
After-treatment technologies [70, 71]	Reduce emissions	Costly and increases fuel penalty in some cases	Decrease the need for after-treatment methods

is added to diesel and biodiesel to test the blends' adaptability to PCCI combustion. Adding ethanol has improved oxygen content in the blends and ignition delay times. High latent heat of ethanol aids in reducing the temperatures, hence NOx emissions and rich oxygen content of the blends helped in reducing soot [6, 8, 28]. Diethyl ether blends are blended with diesel and biodiesel to improve PCCI combustion. The tests indicated a decrease in emissions and improvement in performance in particular test conditions [7]. Biodiesel and diesel blends B20 and B40 are tested with PCCI combustion to explore the adequacy of biodiesel-diesel blends for premixed combustion. Experiments conducted on biodiesel-diesel blends exhibited better emission characteristics than diesel [27, 40]. Methanol and iso-octane are tested with HCCI and partially premixed combustion to assess their performance. Experiments reported ignition in fuel-lean zones and lower charge stratification for methanol. Ignitions in fuel-lean zones improved CO emissions, and lower charge stratification improved the SOI window resulting in low NOx emissions. Numerical simulations carried out at different conditions indicated better emissions for methanol in PCCI combustion [47]. Naphtha gasoline, ethanol, and methanol are tested under partially premixed combustion to study engine-out soot emissions. From the test results, Shamun et al. [73] identified that the maximum limit of soot mass concentrations for methanol is 1.6 mg/m^3, whereas gasoline soot mass concentration never decreased below 1.6 mg/m^3. Oxygenated fuel blends and split injections strategy have good emissions characteristics in diesel engines. The study by Choi et al. [74] with esters and ether blends of diesel noticed a drop in soot emissions, which is significant in rich conditions and split injections. Biodiesels have the inherent ability to reduce soot emissions because of their soot-inhibiting properties. Low-temperature combustion requires longer ignition delays for proper mixing, lower equivalence ratios, and low local combustion temperatures. The high oxygen content of biodiesel aids in improved smoke-less combustion at lower EGR rates [74, 75]. Spray characterization of diesel ethanol blends during early injections is carried out by park et al.[76]. The authors identified a decrease in droplet size, high evaporation, superior atomization, and longer ignition delay for diesel ethanol blend sprays than diesel sprays. On top of alcohol fuel, different diesel fuels were tested for PCCI combustion such as ultra-low sulfur diesel (ULSD), diesel fuel produced from low-temperature Fischer–Tropsch processes (LTFT), and renewable diesel (RD) (hydrotreated camelina oil). Results indicated a drop in gaseous emission for LTFT and RD than ULSD [42]. Gasoline diesel blends are employed in PCCI combustion to mitigate the high burning velocities which result in high combustion noise. Experimental reports indicated a reduction in liquid impingement on cylinder walls and enhanced air-fuel mixing, hence combustion phasing for gasoline/ diesel blends [43, 77]. Dimethyl ether (DME) is tested in PCCI combustion mode and found lower NOx emissions than conventional combustion. Results reported lower HC and CO emissions for PCCI combustion than HCCI combustion [78]

Due to its high cetane number and better combustion characteristics, DME leads to knocking and combustion noise in PCCI Combustion. To mitigate this, LPG is added to DME as an ignition inhibitor to change the premixed fuel property [79]. The effect of biodiesel on PCCI combustion using experimental and simulation techniques is investigated by Hwang et al. [80]. The presence of oxygen species, low sulfur, aromatic content, and low soot precursors in biodiesel is the reasons for low soot emissions for biodiesels. Numerical simulation conducted for biodiesel and their diesel blends for PCCI combustion resulted in lower emissions for B100 than B20 blend. Numerical results further indicated that the rise in specific fuel consumption for biodiesel is not mitigated with PCCI combustion. Methyl decanoate and methyl-9-decanoate are used as surrogate fuels for the numerical simulation of B100 and B20 fuels for PCCI combustion [81]. Jatropha oil biodiesel diesel blends with petrol and diesel as secondary fuels are investigated by Sendilvelan et al. [82] for PCCI combustion. Methanol and polyoxymethylene dimethyl ether blends are numerically tested for their effectiveness in PCCI combustion. The increase in the methanol ratio in the blend resulted in lower NOx and a rise in CO and HC emissions [83]. The visualization studies conducted by Hikichi et al.[84] on a PCCI engine showed a decrease in flame luminosity with methanol injection. The authors also noticed a delay in flam luminosity and effective reaction suppression in the case of earlier methanol injections. In addition to liquid fuels, gaseous fuels are also utilized in PCCI combustion to reduce CO_2 and NOx emissions from dual-fuel engines. Promising results in emissions and brake thermal efficiency are obtained from natural gas diesel premixed combustion [85]. Fuel reactivity and volatility and critical parameters for PCCI combustion processes. Experiments conducted with fuels of different volatility and cetane numbers show that a decrease in cetane number results in high CO and HC emissions. However, lower cetane number fuels inhibited improper combustion of early injected fuel. The rise in fuel volatility resulted in reduced particle concentration and fuel consumption [86]. Nine different fuels with different octane range are tested for premixed combustion. Fuels with octane number less than 70 are found to be running in the entire load range of the engine including idle conditions. Emissions such as NOx and soot are found to be less than EU VI and US 10 regulations [87].

The present section presents various fuels used for PCCI combustion in the literature. The literature survey identified that successful PCCI combustion fuel properties and their composition are vital. Many conventional and alternative fuels are tested for PCCI combustion to identify the typical fuel properties that enhance the PCCI type of combustion. Critical analysis of PCCI combustion of alternative fuels is carried out, and the advantages and disadvantages of different fuels tested with PCCI combustion are showcased in Table 3.

PCCI combustion characteristics

Combustion

The previous section discussed various alternative fuels used with PCCI combustion for possible enhancement in performance and emission. The present section reviews the combustion aspects of different conventional and alternative fuels with PCCI combustion. In this regard, Simescu et al. [88] carried out premixed combustion with PFI and main injections. In their work, the authors identified the HCCI type for the premixed fuel and the diffusion type of combustion for the direct-injected fuel. The increases in the PFI injection quantity resulted in a higher HCCI (premixed) type of combustion than diffusion combustion. Two-stage heat release can be observed in Fig. 5, which is typical of premixed combustion. Single-stage or two-stage heat releases are found to be depending on the injection duration. Andre et al. [88] study on injection duration effects revealed a single heat release peak for short early injections vis-a-vis. Increasing injection duration also raised the equivalence ratios and fuel wall impinging. Ignition delay is one of the prime factors responsible for PCCI combustions. In the works of park et al. [36], advancing the injection timing beyond 30° BTDC made SOC independent of injection timing. At this stage, SOC is purely dependent on the auto-ignition chemistry of the mixture. Advancing fuel injection beyond 30° BTDC, such as 60° BTDC, reported a drop in heat release rates and retarded combustion phase.

Numerical studies are carried out by Yanzhao et al.[90] to understand the HCCI and premixed combustion modes. Their results show that CA50 is independent of SOI timing in the HCCI mode, and CA50 strongly depends on SOI timing in the PCCI combustion mode. Less than 2500 K temperatures are found at advanced injection timing of – 40° BTDC with equivalence ratios less than 1. A comparative analysis is carried out between different LTC strategies, and it is identified that max. Pressure rise for PCCI is in between HCCI and RCCI mode of combustion. At lower loads, the PCCI combustion mode showed nominal heat release rates. In contrast, PCCI resulted in a significant pressure rise at higher loading conditions than RCCI and conventional combustion [35].

As discussed earlier, lower cetane number fuels are most suitable for the PCCI type of combustion. Dijkstra et al. [91] conducted experiments with blends of cyclohexanone with diesel fuels to evaluate the effect of cetane number and oxygen content. In their works, authors identified that the rate of heat release is severely affected by cetane number. Due to the low cetane number, there is an increase in combustion duration and a decrease in pressure rise rate. The experiments with PCCI combustion of diesel-biodiesel blends B20 and B40 reported an advance in the start of combustion (SOC) for B40 and retarded SOC for B20. Authors attributed the advanced SOC for the B40 blend to the domination of biodiesel fuel properties in the blend. Adding biodiesel to diesel has shown a positive effect in decreasing CO and HC emissions in PCCI combustion. Nevertheless, the combustion is found to be slightly inferior to diesel combustion [40]. B20 and B40 blends are also utilized by Elkelawy et al. for PCCI DI combustion. In their works, authors noticed a significant rise in cylinder pressure and advancement in location P_{max}. The increase in premixed ratio resulted in a drastic rise in pressure, combustion noise and knock [72]. Miller cycle is applied for PCCI combustion using the late inlet valve closing technique (LIVC) by Kawano et al.[91]. In their works, authors identified a rise in ignition delay, leaning of the air-fuel mixture, and combustion temperatures below 2200 K.

Table 3 Advantages and disadvantages of various low-carbon fuels

Fuel used	Type of injection	Advantages	Disadvantages	References
Biodiesel–diesel blends B20, B40	Pilot and main injections	B40: better combustion and engine performance B20: better emissions	High particulate emissions for biodiesel–diesel blends	[40]
Waste cooking oil biodiesel–diesel blends	Manifold and direct injection	↑ BTHE, ↓ CO, ↓NOx, ↓UHC, ↓Smoke	Difficulty in operating at higher loads	[72]
Cotton seed biodiesel with *n*-butanol binary mixtures	DI: Cotton seed biodiesel PFI: *n*-butanol	↓NOx, ↓ soot, ↓RI, ↓BSEC	Dramatic increase in CO & HC emissions	[41]
Diesel and fumigated ethanol fuels	Fumigated ethanol into intake air at 40 °C in the intake manifold	↓NOx @ low loads ↓Smoke	↑HC, ↑CO	[8]
Bioethanol–diesel blends	Fuel injection into pre-chamber	↓CO, ↓NOx, ↓ Smoke, ↑Exergy efficiency	Rise in and CO HC emissions at higher load operation	[6]
Ethanol–diesel—biodiesel blends	Direct injection	↓NOx, ↓ soot, ↑ BTHE, prolonged ignition delay with ethanol, High EGR tolerance, extended load range	↑CO, ↑HC	[28]
Diethyl ether in biodiesel blends	Pilot injection into Fuel vaporizer and main injection	↑higher cylinder pressure, improvement in fuel properties, ↓NOx, ↓HC, ↓CO, ↓Smoke	↑Knocking tendency	[7]
Waste cooking oil biodiesel	CRDI system with injection control	↓ PM for biodiesels than diesel, ↓Smoke, ↓NOx	↑CO, ↑HC @ advanced injection timings	[27]
Methanol, Iso-octane	Varied SOI timings	Methanol results in lower charge stratification, extended SOI window, lower NOx levels, ↑ BTHE, ↓ CO	↑ UHC	[47]
Naphtha gasoline, ethanol, methanol	Single injection and double injection	Lower soot mass concentrations for methanol, higher EGR window without NOx-soot trade-off	Higher intake temperatures and rise in intake O_2 increase PN concentrations	[73]
Soybean Biodiesel	Retarding fuel injection timing	↓Smoke, ↓THC, ↓CO, high EGR operation window, high EGR tolerance	↑ NOx, higher cetane number, ↓ ignition delay	[74]
Bioethanol-diesel blends	Varying injection timings	High volatility, superior atomization, ↑ ignition delay, ↓NOx, ↓CO	↑ HC	[76]
LTFT (low-temperature Fischer –Tropsch process), hydrotreated camelina oil (Renewable diesel)	Single injection strategy	LTFT has the best performance, ↓CO, ↓HC, ↓and NOx emissions compared to diesel, and RD has lower carbon footprints	↑CO, ↑HC due to over lean combustion	[42]
Gasoline–diesel blends	Advanced injection timings	↓ liquid fuel impingement, improved air–fuel mixing, better combustion phasing, prolonged ignition delays	↓ BMEP @ advanced injections, ↑ Combustion noise @ advanced injections	[43]
DME diesel, LPG	PI: DME/LPG DI: diesel	↓ NOx, ↓ mass-averaged temperatures, ↓maximum cylinder pressures	Early start of combustion, detonation, ↓ BTHE	[79]
Biodiesel	Single injection strategy	↓ PM suppressed CO and HC emissions due to biodiesel	↑ BSFC, ↑ CO, ↑ HC, ↑ NOx than diesel	[80]

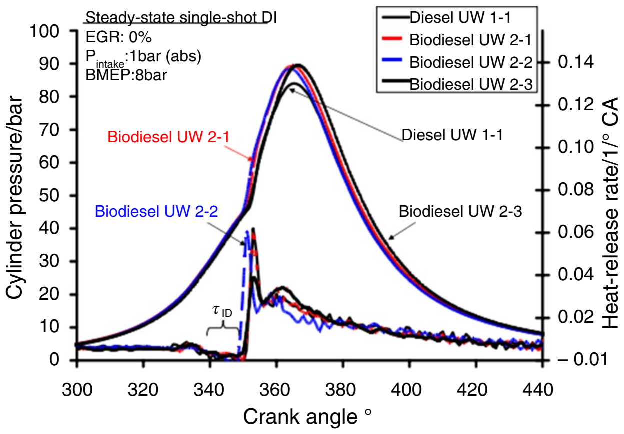

Fig. 5 Premixed combustion phases [89]

A critical review of PCCI combustion characteristics for conventional and alternative fuels is shown in Table 4.

Performance and emission

Low-temperature combustion techniques' fundamental objective is to simultaneously reduce harmful NOx and soot emissions. The LTC techniques, such as PCCI, successfully minimize the trade-off relation between NOx and soot. However, this technique suffers from high CO and HC emissions, a drop in thermal efficiency, and a limited operating range [13, 107–110]. Many researchers tried to address these issues in PCCI combustion using multiple injection strategies, high injection pressures, EGR, boost pressure, and alternative fuels. The techniques applied to reduce the bottlenecks in PCCI combustion technology are discussed below. The experiments by Jacobs et al. [111] on premixed combustion showcased the possibility of ultra-low NOx and soot emissions with PCCI technology. The authors explored the relationship between combustion temperatures, equivalence ratios, and soot oxidation and found that soot formations are independent of equivalence ratios below 1500 K temperature. The authors demonstrated low soot and NOx levels of ~ 0.03 FSN and 3 ppm under rich conditions. Despite lower NOx and soot emissions, high HC and CO emissions are recorded for premixed combustion. To reduce CO, HC emissions and destruction of thermal efficiency, increased boost pressure, diesel oxidation catalyst, low compression ratios, and advanced injection timings are employed. DOC with premixed combustion effectively reduce CO emissions next to paraffins, olefins, and aromatics.

Nevertheless, DOC's conversion efficiency is higher than premixed lean combustion than rich [112–114]. In premixed combustion, the overall combustion takes place in lean mode; irrespective of the equivalence ratio, the temperatures are lower. For facilitating exhaust after-treatment strategies (like lean NOx trap), premixed combustion must occur in a rich mode. However, rich premixed combustion was found to have deactivated the platinum-based DOC [115]. Detailed studies are conducted to determine the root cause of HC emissions in premixed combustion. Numerical and optical techniques such as homogenous reactor simulations, visual access engines, and ultraviolet planar laser-induced fluorescein (UV PLIF) are utilized for analyzing the HC emission formation. The studies identified that forming liquid films, bulk quenching over lean regions, increased charge dilution, and low loads are the causes of higher HC emissions in premixed combustion [116–119]. Diesel fuel vaporizers are developed for external charge preparation in a PCCI engine and used simultaneously with a toroidal combustion chamber. The cut section view of the diesel fuel vaporizer is shown in Fig. 6. The use of diesel fuel vaporizer in conjunction with toroidal combustion chamber resulted in a decrease in CO and HC emissions with a marginal increment of NOx emissions and a 5.31% rise in BTE. The authors concluded that using a toroidal combustion chamber with fuel vapor induction would reduce the drop in BTE [120].

Due to increased HC and CO emissions for conventional mineral diesel, researchers have started identifying more EGR-tolerant and ignition-resistant fuels. In this scenario, many alternatives are tested for their efficacy in PCCI combustion. The effect of fuel properties on premixed combustion is investigated with fifteen different fuels, such as diesel, biodiesels, n-heptane-iso-octane mixtures, and n-cetane-HMN mixtures. Biodiesel blends such as soybean methyl esters and palm oil methyl esters have reduced UHC and CO emissions in premixed combustion mode. The drop in UHC and CO emissions is attributed to the shorter burn durations of biodiesels. The tests conducted with other fuels to determine the effect of ignition quality and volatility on CO and HC emissions proved that fuel volatility has little to no impact on UHC and CO emissions [121]. Fuels such as diesel are mixed with high octane fuels to avoid the adverse effects of using high cetane fuels in PCCI combustion. The research on diesel-gasoline blends shows that they are efficient in increasing ignition delay, operating range, and reducing emissions. A reduction in pressure rise rate is also observed with the increase in gasoline blends. This decrease in pressure rise rate at higher loads would eventually reduce combustion noise and knock. Further studies are conducted to determine the effect of fuel octane on premixed combustion. The use of gasoline-diesel blends in the RON range of 75~85 indicated lower smoke and NOx emissions than diesel. However, high-octane fuels have difficulties at partial loads and high EGR conditions [122–124]. Calophyllum inophyllum B20 blend is tested in diesel engines with low-temperature combustion for a possible reduction in emissions. Results indicated a drop in CO and HC emission compared with diesel and a rise in brake thermal efficiency.

Table 4 Summary of PCCI combustion characteristics for alternative and conventional fuels

Fuel	Combustion Parameters							
	Injection strategy	Ignition delay	Cylinder Pressures	Heat release rates	Combustion Phasing (CA 50)	Combustion noise/Knock	IMEP	References
Diesel	Early direct injection				CA50 advanced with hot EGR%		20% ↑	[92]
Diesel	Sweep of injection from 15 to 40° BTDC	↑ with advancing the SOI, ↑ considerably beyond 55% EGR		↓ with an increase in the EGR rate		↑ with the increase in SOI advance		[93]
Diesel	Single injection sweeps 30° BTDC to 5° ATDC	↑ with injection advancing and retarding, the delay time is highest for late injection	↑ with advancing the injection. Early injections resulted in high cylinder pressures than late injections	↑ for advanced injections and decreased for late injections	advanced with early injections. CA50 near TDC resulted in better performance		Highest at 15° BTDC, and combustion efficiency is found to be higher for early injections than for late injections	[94]
Diesel		↑ with EGR	↓ with higher exhaust gas recirculation	↓ and retarded for higher EGR				[95]
Diesel	Single injection 40° BTDC; 40% EGR	↑ with injection advancement	Highest for injections close to TDC and ↓ with injection advancement	↓ with the rise in EGR and higher at advanced timings			COV_IMEP rose to 12% during PCCI combustion	[96]
B20, Jatropha oil methyl esters	PFI, 0.25 premix ratio	↑ ignition delay with EGR and JOME						[97]
Diesel	Single and dual injection	↑ with injection advancement before 30° BTDC	P_{max} is advanced as injection timing is advanced	The highest heat release rates are observed @ 60PI and 40 MI				[98]
Diesel	Multiple injection strategies (2-pilot and Main)		A sharp rise in cylinder pressures is not observed for the double injection strategy	↑ in HRR and premixed reactions are observed for double injection	CA50 is maintained const. @ 8°ATDC	Combustion noise is found to be less for triple injections	COV_IMEP < 3% for triple injections	[99]
Biodiesels-Soy, Canola, Yellow grease	Fuel injection in the intake manifold	↑ with EGR at a steady state, Canola biodiesel has shown slight SOC advancement	Highest for biodiesels	Highest for biodiesel and single shot DI mode of combustion is observed				[89]

Table 4 (continued)

Fuel	Combustion Parameters							
	Injection strategy	Ignition delay	Cylinder Pressures	Heat release rates	Combustion Phasing (CA 50)	Combustion noise/ Knock	IMEP	References
Blends of diesel and gasoline	Late single injection with EGR	↑ with a decrease in inlet O_2 concentrations and ↑ with gasoline % in the blend					COV_IMEP < 5% is observed for all gasoline-diesel blends	[100]
Ethanol-biodiesel-diesel	Single and double late injection strategy		↓ for ethanol-biodiesel -diesel blends		Retarded CA50 for ethanol-biodiesel-diesel blends	Ethanol-biodiesel-diesel blends show low combustion noise	High instabilities are observed for blends of ethanol–diesel-biodiesel due to their low cetane number	[101]
Gasoline, Diesel	Single injection	Higher for gasoline than diesel		HRR is advanced for gasoline and takes place after injection, unlike diesel	CA50 rises sharply with SOI beyond -30° BTDC for gasoline		High IMEP is observed for gasoline than diesel ~ 14.86 bar	[102]
Soybean Biodiesel-B0, B20, B50, B100	Single injection with early and late injections	Longer ignition delay is observed for late injections than early injections; Biodiesel ↑ ignition delay		↓ with an increase in biodiesel content				[103]
Soybean Biodiesel-B0, B100	Dual injection with pilot and main	Early first injections lead to rapid pressure rise rate; low pressures are noticed for retarded main injection		↓ HRR with advanced injection				[104]
Diesel, biodiesel, biodiesel -ethanol blends	Pilot and main injections		High cylinder pressures are noticed for biodiesel ethanol blends	HRR is highest for biodiesel than diesel and biodiesel ethanol blends	Maintained at 18° ATDC			[105]
Diesel, Biodiesel	Single injection strategy	↑ with late injections and EGR; ↓ with the use of biodiesel and boost pressure	Lower cylinder pressures are recorded MK type of combustion	HRR is higher for MK type of combustion	Biodiesel is found to be more tolerant to high EGR and hence better CA50 control			[74]

Table 4 (continued)

Fuel	Combustion Parameters							
	Injection strategy	Ignition delay	Cylinder Pressures	Heat release rates	Combustion Phasing (CA 50)	Combustion noise/ Knock	IMEP	References
B20D80, B40D60	Port fuel injection		Cylinder pressures are found to be increasing with an increase in pre-mixed ratios; P_{max} is advanced	With an increase in premix ratio, two-stage HRR is recorded				[72]
Ethanol–diesel- bio-diesel	Single fuel injection strategy with EGR	Longer ignition delays are observed with the increase in methanol content in the blend	Drop in-cylinder pressure is recorded for biodiesel ethanol blends	↑ with ethanol content in the blend and retarded HRR is observed	↑ of ethanol content retarded CA50			[28]
DME/LPG with diesel	PFI of DME/LPG mixture; DI of diesel	The rise in LPG content retarded the SOC	The cylinder pressures declined slightly with the increase in LPG content	Two-stage HRR is recorded	CA50 delays with the increase in LPG content; Combustion duration is found to be shortened with LPG addition			[79]
DME	PI of DME and DI of DME	Decrease in pilot DME quantity showed retarded HRR	Cylinder pressures tend to increase with the increase in DME pilot quantity	↑ HRR is observed ↑ in pilot DME contents	Short combustion durations are recorded with the rise of pilot DME content	@ Higher DME pilot quantity knock is observed		[46]
Gasoline -diesel blends	Late single injection	↑ with an increase in gasoline content in the blend					@ 7.5 bar IMEP, COV_IMEP < 5%	[106]

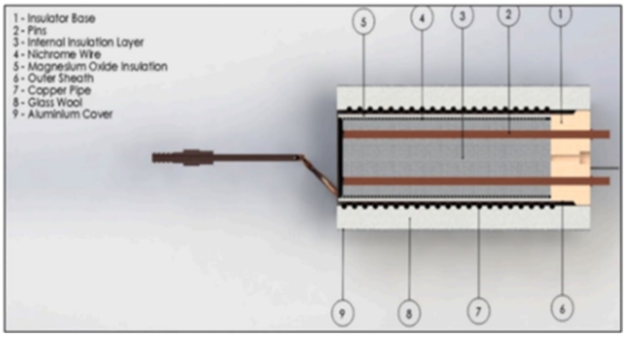

Fig. 6 Cut-section view of diesel fuel vaporizer [120]

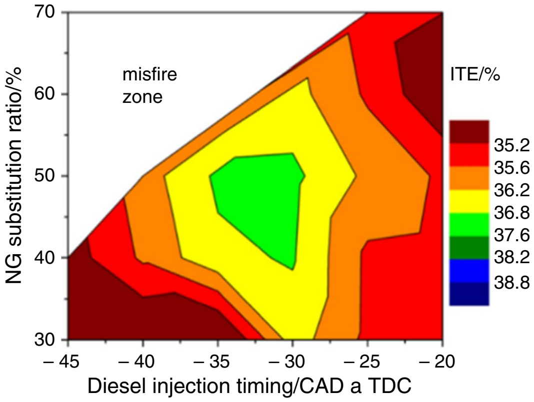

Fig. 7 Effect of natural gas substation and injection timing on indicated thermal efficiency [136]

Authors identified that 15% pilot injection quantity and 10% EGR are best for lower engine out emissions and better performance [125]. The works conducted by Tormos et al.[126] quantifying the biodiesel combustion behavior also revealed faster combustion durations, lower exhaust gas temperatures, and CO and HC emissions for biodiesels. Soy-based biodiesel B50 is studied under two low-temperature combustion modes, early and later injection strategies. Biodiesel for LTC combustion reported decreased elemental carbon (EC) and HC emissions. However, higher particulate emissions are noticed for biodiesel LTC combustion [127]. Premixed combustion used various soy-based biodiesel blends such as B100, B20, and B50. The HC emissions are lower for B100 when compared with diesel, and smoke emissions (FSN) values are decreasing with an increase in biodiesel concentrations in the blend. Nevertheless, the particle size distribution increased with biodiesel content [128]. Fuel reactivity plays a significant role in PCCI combustion. Optimum combustion delay yields good results at a particular load and compression ratio. Low reactive fuels with lower emissions, even in conventional combustion, are required for PCCI combustion. For PCCI combustion, some alcohols and cyclic oxygenates are promising alternatives [129]. Fuel oxygen mass fraction varies from 0 to 15%, and their emission behavior is studied in a diesel engine. Oxygen content and prolonged ignition delays with EGR resulted in low PM emissions [130].

High oxygen content fuels such as polyoxymethylene dimethyl ether (PODE) and methanol blends are investigated with PCCI combustions. The high methanol content in the blends increased premixed combustion and decreased NOx and soot trade-off. PCCI combustion with optimum methanol ratio and pilot injection strategy resulted in the highest brake thermal efficiency of 46.58% [131]. Methanol content and pilot injection timings are studied in a dual-fuel engine. With the increase in methanol content in the blend, there is an improvement in HC emissions, brake thermal efficiency, and COV [132]. In addition to methanol diesel blends, ethanol diesel blends are investigated with PCCI combustion. Results indicated improved brake thermal efficiency and decreased NOx emission compared to diesel PCCI. However, the rise in CO and HC emissions is reported for the increase in ethanol content in the blend [133]. *N*-butanol, often considered the next generation biofuel, is tested in the premixed combustion mode. Results indicated comparable efficiencies with diesel and a considerable reduction in NOx and smoke emissions. The high CO and HC emissions of *n*-butanol are expected to decrease at higher loads with higher flame temperatures. The split injection strategy reduces the high-pressure rise and combustion noise with the rapid burning of *n*-butanol[134]. Different gasoline-like fuels are investigated in premixed combustion mode, and their performance is compared with diesel. At low load conditions, NOx and smoke are reduced without applying EGR. At higher loads and rpm, gasoline fuels resulted in better NOx and smoke than diesel by using sufficient EGR. However, the high CO and HC emissions from gasoline-like fuels require after-treatment and multiple injection strategies [135]. The effect of natural gas and diesel in premixed combustion is investigated by Park et al.[136]. The results indicate that the natural gas substitution ratio increase indicated thermal efficiency, THC, and CO emissions, as shown in Fig. 7. However, the natural gas substitution ratio increased and decreased NOx and smoke emissions below Euro VI limits. Experimental results further indicated that using 50% EGR CO and HC emissions reduced and increased indicated thermal efficiency. The effects of PCCI combustion on after-treatment devices such as lean NOx traps (LNT), diesel oxidation catalyst (DOC), and diesel particulate filter (DPF) are experimentally studied. LNT is benefited from PCCI combustion as NOx emissions are lower. At the low load operating range of PCCI, low light-off temperatures prevail, and DOC efficiencies are lower. Because of the lower PM emissions from the PCCI engines, low desoot events are required, which reduces the fuel penalty [137]. The literature reviewed concludes that using biodiesels, alcohols, and their blends with diesel efficiently mitigate the adverse effects of PCCI combustion. The critical analysis of literature showcasing the effects of various fuels with premixed combustion is presented in Table 5.

Numerical simulations of PCCI combustion

The above sections discussed the effects of various conventional and alternative fuels' premixed combustion on diesel engines' performance and emissions. As discussed in the previous sections, PCCI combustion produces ultra-low NOx and smoke emissions, whereas CO and HC emissions increased dramatically. The critical restrictions for PCCI combustion are the lack of control over parameters such as CO, HC emissions, combustion phasing, and combustion noise. Numerical studies are conducted to study the root causes and develop possible solutions to the issues in premixed combustion. The cardinal works regarding numerical simulations of PCCI combustion are reviewed and presented in this section. Complete cycle CFD simulations are carried out to determine a clean combustion window for PCCI operation carried out by Ming et al. [171]. From the CFD analysis, contours of various engine performance and emissions parameters are obtained as SOI and IVC timing functions. As shown in Fig. 8, retarded IVC timing created a wider start of injection (SOI) range for clean PCCI combustion. The computational mesh used for the simulation is shown in Fig. 9. The rise in CO and HC emissions is mitigated in PCCI combustion owing to biodiesels' 11% oxygen percent. The effect of biodiesel with PCCI combustion is numerically investigated using KIVA-3 V coupled with CHEMKIN. Methyl 9-decanoate is used as surrogate fuel for representing the behavior of biodiesel. The proposed multi-component biodiesel mechanism consists of 71 species and 192 reactions. Three-dimensional numerical analysis is carried out for optimizing PCCI operating parameters using the mechanism. SOI of 30° BTDC is the optimum setting for reducing NOx, THC, and CO emissions with a slight penalty of thermal efficiency. High spray tip penetration is observed for B100 compared to B0. Despite higher spray tip penetration leading to high THC for B100, a compensating rise of oxidation in expansion stroke is observed in the simulations [172]. To accurately capture biodiesel's effect in low-temperature combustion, effective reaction mechanisms with a minimum number of species and reactions are required to reduce the computation cost. In this aspect, tri-component biodiesel surrogates consisting of methyl decanoate, methyl 9-decanoate, *n*-heptane and methyl decanoate, methyl 5-decanoate, and *n*-decane are used. In the above fuel, surrogate methyl decanoates represent saturated and unsaturated biodiesel, whereas *n*-decane is used to match the biodiesel's energy content. The developed biodiesel mechanism contains ~60–70 species and ~170 reactions. The skeletal mechanism thus developed is validated with low temperature combustion mode.

A good agreement with experimental results is observed for neat biodiesel and blends [173, 174]. The computational studies conducted for mixtures of *n*-heptane, iso-octane, toluene, and ethanol resulted in improved operating range. Low octane fuels exhibited improved performance characteristic [175]. Sector simulations are carried out by many researchers instead of full cycle simulations to reduce computational costs. Sector simulations investigate soy-based biodiesel and its diesel blend (B20) for low-temperature combustion. The numerical parametric studies indicated injection timings of 34–28° BTDC are optimum for biodiesel diesel blends. The results further proved that using biodiesel surrogate blend models predicted biodiesel fuel behavior precisely in multi-dimensional engine simulations [176, 177]. The effect of swirl ratio and split injection strategies on biodiesel PCCI combustion is investigated by Zehni et al. [178]. Results indicated a good agreement between experimental and simulated cylinder pressures and heat release rates. Numerical results indicated that increased swirl ratio with split injection had adverse effects on PCCI combustion. Advanced injection timings provided better performance and emission results for both single and multiple injections, and an SOI of 35° is considered a sweet spot value for both injection strategies. A single injection strategy is better at high swirl ratios than a split injection strategy. The fuel droplet distribution at a swirl ratio of 1.1 for both single and main injection at SOI of 40° BTDC. A detailed chemical kinetic investigation on the effects of oxygenated fuel blends is carried out by Curran et al. [179]. In their works, authors studied various fuels such as methanol, ethanol, dimethyl ether, dimethoxymethane, and methyl butanoate as oxygenated fuel additives. From the chemical kinetic model, analysis authors noticed that adding oxygen additives to diesel fuel would increase the carbon atoms bonded to oxygen. These oxygen-bonded carbon atoms are found to provide zero soot precursors. The results further indicated that with the increase in fuel oxygen content to 30–40%, soot precursors would reduce to almost zero. Diesel PCCI combustion numerical simulations are carried out to determine the split injection strategy's effect and optimize the engine parameters for efficient PCCI combustion. Results indicated that second injection timing and the fuel velocity at the injector affect the combustion process. In other work, the effect of boost pressure, EGR rate, premixed fuel fraction, and late injection timings on engine emissions and performance is numerically investigated. Sweeps of the above parameters are carried out using multi-dimensional CFD analysis, and a multi-objective genetic algorithm is used for optimizing the above-said parameters. The numerical models successfully

Table 5 Summary of PCCI combustion effect on performance and emissions with alternative and conventional fuels

Fuel	Injection strategy	Testing condition	Efficiency	Fuel consumption	NOx	Soot	CO	HC	References
Gasoline with RON 99,89 and 69	Single and double injection	@ 16–18 bar gross IMEP	Gross indicated efficiency-54%		↓ below 0.25 g/kWh	1.50 FSN	Low @ high EGR 62%	Low @ high EGR62%	[138]
Gasoline-like fuels and ethanol with 69–99 octane range		@5–18 bar IMEP; CR 14.3:1;1300 RPM; EGR 50%	Gross indicated efficiency -54–56% with a high boost; brake efficiency is higher than 48.5%		↓ below 0.30 g/kwh	1–2 FSN for gasoline fuels and 0.06 FSN for ethanol	↑	↑	[139]
Ethanol	double injection	@ 12 & 5 bar IMEP; 1200 RPM	↓ combustion efficiency@ advanced first injection; ↑ thermodynamic efficiency @ advanced first injection		↑ @ retarded first injection; ↓@ retarded second injection	↓	↓@ retarded first injection; ↑ @ retarded second injection	↓@ retarded first injection; ↑ @retarded second injection	[140]
80% gasoline and 20% n-heptane	Double injection strategy	@ 10 bar IMEP; 1200 rpm; 25% EGR	↑ thermodynamic efficiency		↓ with an increase in the pilot ratio	↑ with an increase in the pilot ratio		↑ with rise in pilot ratio	[141]
Gasoline	Single injection	@ 5 bar IMEP; 1200 rpm; 25% EGR; 2-stroke	Combustion efficiency is highest @ 400 bar rail pressure; indicated efficiency is highest @ 600 bra rail pressure		↓ with increase in rail pressure	↓ with increase in rail pressure	↑ with increase in rail pressure	↑ with increase in rail pressure	[142]
N-heptane, iso-octane, and toluene blends with RON in the range of 60	Single injection	@5.1 bar Fuel MEP; 1200 RPM				Soot ↓ and PN increased with early injections			[143]
Naphtha	Single injection	@ 5.1 bar fuel MEP; 1200 RPM			↓	↓			[144]
Diesel	Dual injection	@3–10.8 bar IMEP; CR:18–16; Speed: 1500–200		↑ with injection close to TDC	↓ for lower compression @ part loads	↓ for lower compression @ part loads	↑ with decrease in compression ratio	↑ with decrease in compression ratio	[145]

Table 5 (continued)

Fuel	Injection strategy	Testing condition	Efficiency	Fuel consumption	NOx	Soot	CO	HC	References
DME / Diesel	PI -DME; DI -diesel	1700 RPM; BMEP:0.25–0.5 MPa	↑ with increase in DME premix ratio	↓ with increase in DME premix ratio	↓ with increase in DME premix ratio	↓ with increase in DME premix ratio			[146]
Diesel / Ethanol	Dual injection strategy	@ 0.615 Mpa IMEP; 1200 rpm	3.2% gain in indicated efficiency		33% reduction compared to diesel	33% reduction compared to diesel	< 10 g/kWh	< 10 g/kWh	[147]
Natural gas + ozone	Pilot injection	800 RPM; intake pressure of 2.7 bar			↑ with ozone addition		↓ with ozone addition	↓ with ozone addition	[148]
Lemongrass oil	PFI	@ 0–75% of full load; 1500 RPM	deteriorated when compared with diesel		↓	↓			[149]
Biodiesel/*n*-butanol	PI: *n*-butanol; DI: biodiesel	@1–3 bar IMEP; 800 RPM		↑ for *n*-butanol	↓77% @3 bar IMEP	↓ 98% compared to mineral diesel @ 3 bar IMEP	↑ by 10–20 times	↑	[150]
Gasoline/ Ethanol/ diesel	Double injection	@14.8 bar IMEP; 25% EGRs	Thermal efficiency: Ethanol > Gasoline > diesel	↓ retarded pilot; Ethanol < Gasoline < diesel	Ethanol < Gasoline < diesel	Ethanol < Gasoline < diesel	Ethanol > Gasoline > diesel	Ethanol > Gasoline>diesel	[151]
B20/ B50 diesel blends of coconut oil biodiesel	Multiple injection strategies				< 100 ppm is obtained	↓			[152]
Diesel/ palm oil B100	Single injection strategy	@ 2 bar BMEP; 1400 RPM; EGR 46–50%		Remained constant with rise in EGR levels	↓ B100 than diesel	↓ B100 than diesel	Diesel↑ B100↓	Diesel↑ B100↓	[153]
DME	Dual injection	@1500 rpm; $\Phi = 0.2–0.3$			90% drop for the second injection after 10^o ATDC				[154]
DME	PI: DME DI: DME	@0.05–0.55 bar IMEP;1000 RPM		↑ with port-injected DME	↑ with max NOx < 30 ppm		↓	↓	[155]
Diesel, synthetic fuel produced from HTFT and LTFT processes	Dual injection	@ ~ 3.70 bar IMEP; EGR 23–38%; 1500 RPM	↑ 1.5% compared to diesel	↓ for HTFT& LTFT; Diesel > HTFT > LTFT	↓ for LTFT; ↓ by 17% compared to diesel PCCI	↓ for LTFT; ↓ 63% compared to diesel PCCI	↓ for LTFT; ↓ 75% compared to diesel PCCI	↓ for LTFT;↓ 80% compared to diesel PCCI	[156]

Table 5 (continued)

Fuel	Injection strategy	Testing condition	Efficiency	Fuel consumption	NOx	Soot	CO	HC	References
Waste cooking oil biodiesel	Single injection			↑	Little success in simultaneous reduction in NOx and PM				[157]
DME/ methanol	PI: DME DI: Methanol	@2000 RPMs						Methanol and formaldehyde constituted 70% of HC emissions	[158]
Ethanol	Dual injection	@16.28 & 14.39 bar Fuel MEP; EGR 0–50%	Sligh increase in thermal efficiency with EGR		↓	↓	↓ with EGR 40–47%; and with 95% catalyst efficiency	↓ with EGR 40–47%; and with 97.5% catalyst efficiency	[159]
Diesel, Ethanol, gasoline	Single injection	EGR 48 and 38%; Fuel MEP:20.4 and 28.3 bar	↓ for stoichiometric PPC; ethanol has high efficiency than other test fuels			Almost no soot for ethanol with λ~1			[160]
Diesel/ ethanol	PI: Ethanol DI: Diesel	@ IMEP 10–12 bar; EGR 0–60%; boost 2–1.75 bar; RPM 1500	Fuel efficiency dropped by 10%		↓ and less than 0.2 g/kWh	↓ and less than 0.01 g/kWh	↑ with ethanol injection	↑ with ethanol injection	[161]
Diesel/ Ethanol	Single diesel injection and PI: Ethanol; DI: Diesel	@ IMEP 10 bar &17.6 bar; 1500 RPM	↓		↓ @ high ethanol fractions	↓ @ high ethanol fractions	↑ with ethanol content	↑ with ethanol content	[162]
Wheat gram oil/ diesel	PI: Ethanol; DI: Diesel	@ ~ 1–4.4 kW BP; 1500 rpm		↑ with bioethanol share; ↑ from 27.64%-29.14%	↓ from 813–756 ppm	Reduced from 67–30%	↓ with bioethanol addition	↓ with bioethanol addition	[163]
Methanol, n-pentanol, and safflower biodiesel blends	PI: *n*-pentanol & methanol DI: biodiesel	from 25% of full load to full load; 1500 RPM	For 10% mass basis efficiency ↑ with n-pentanol than methanol		↑ with alcohols	Smoke ↓ with alcohols	↑ with alcohol injection	↑ with alcohol injection	[164]
Hydrogenated Vegetable oil	Multi-pulse injection strategy	@ 4.8 bar IMEP; 1500 rpm	↓ efficiency @ high EGR regimes		Near zero NOx emissions	Non-monotonic characteristics with sharp cut-off	↑	↑	[165]

Table 5 (continued)

Fuel	Injection strategy	Testing condition	Efficiency	Fuel consumption	NOx	Soot	CO	HC	References
Water -diesel emulsions blend with diesel	Port injection and direct injection	@0–100% of full load; 1500 rpm	↑ with water-diesel emulsion content in the blend		↓	↓	↓	↓	[166]
Diesel, Citronella -biodiesel blend (B20), cedar wood oil diesel blend (B20)	Port fuel injection with preheating	@0–100% of full load; 1500 rpm	↑ @ part load for B20 blends		↑		↓ decreased due to preheating	↓	[167]
Diesel-methanol	Fumigated methanol in inlet manifold; two injections	1500 rpm; Pin = 1.5 bar; Tin = 35 °C; $\Phi = 0.38$; 5 bar BMEP	↑ ~ 50%				↓ by 40%	↓ by 50%	[168]
Diesel-Methane	PI: methane DI: diesel	1500 rpm; 4.1 bar IMEP,$P_{in} = 1.5$ bar, & 12.1 bar IMEP, Pin = 1.8 bar				↓ < 0.05 FSN	↑	↑	[169]
Pentanol, dimethyl ether blends with diesel blends	Injection close to TDC	0.2–0.8 MPa BMEP; EGR: 15&30%; 1600 RPM		↑ due to low energy content	↓by 56% @ 30% EGR for D60DM20P20 Blend compared to diesel	↑ for oxygenated test blends	↑	↑	[170]

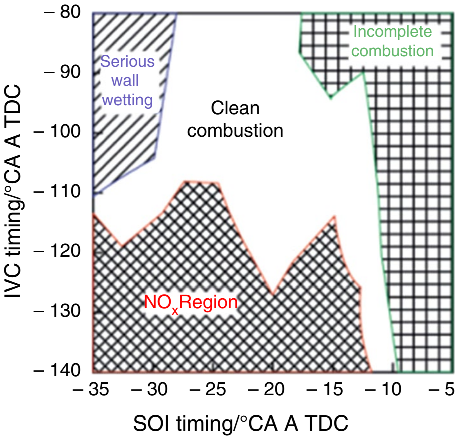

Fig. 8 Effect of SOI and IVC timing on PCCI combustion [171]

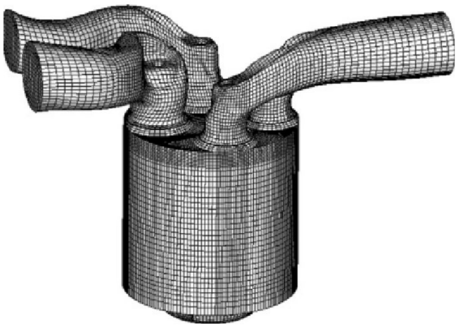

Fig. 9 Computational Mesh [171]

predict the PCCI combustion, and Pareto analysis for various objectives is carried out. Out of this peak pressure rise rate of 4.3 bar/deg., 54% EGR, IVC pressure 1.74 bar, 36% premixing, and late injection timing of 2.9 deg. ATDC are found to be optimum inputs [180, 181]. New spray models are developed to improve the accuracy of the low-temperature combustion simulations.

The proposed model reduced the dependency of the results on the size of the gas phase cell and predicted the relative velocity between the droplet and gas. The simulation's radius of influence is set with a collision radius of 2 mm. Peculiar CO emission behavior is observed while injection timing sweeps at high EGR conditions. This particular injection timing is termed "sweet spots." The phenomenological spray model is implemented in KIVA—3 V code for characterizing the emission behaviors of CO at high dilute conditions. The model successfully predicts the sweet spot behavior in LTC Conditions [182, 183]. CFD studies are conducted to simulate diesel–methanol dual-fuel combustion under low-temperature combustion with high percent energy substitution. The sweeps of injection timings are simulated and found that at high methane lean premixed mixture, methanol cannot be able to sustain combustion resulting in high HC and CO emissions [184]. Numerical simulations are carried out using CONVERGE CFD software for diesel methane dual-fuel combustion to address the issues such as low thermal efficiency and high HC and CO emissions. Simulations are conducted by varying the injection timings and pilot mass amounts (15%, 30% & 45%). Advancing the pilot injection timings decreased methane slip, CO, and HC emissions, whereas advancing the pilot timing beyond 24^{o} BTDC increased CO and HC emissions [185, 186]. The effect of ultra-high exhaust gas recirculation and modulated kinetics on low-temperature combustion is investigated numerically through KIVA-CMC. Results indicated a simultaneous NOx and PM emission reduction in both LTC modes. Temperature is the most critical parameter in reducing NOx emission in high EGR LTC mode. In contrast, the ignition delay is the critical parameter in reducing CO and PM emissions in MK mode [187].

A stochastic reactor model is utilized for numerical analysis of the PCCI engine instead of directly coupling chemical kinetics to CFD. This stochastic approach reduced the computational costs and improved the emission prediction capabilities. The numerical results from the model indicated that fuel-rich pockets in late injections are desirable for autoignition and advanced combustion phasing. The results proved that piston bowl geometry influences in-cylinder mixing and pollutant formation [188]. Dual-fuel low-temperature combustion is analyzed using CFD with various fuels such as diesel–methane and diesel–natural gas. Computational results indicated that higher rail pressure and swirl ratio reduce CO and HC emissions to a larger extent with a penalty of NOx emissions. The computational studies further revealed that the results are susceptible to the component rate in diesel surrogates, injected mass, and velocity. The numerical works further proved that the multi-dimensional modeling of the engine is accurate for predicting dual-fuel combustion [189–192]. A phenomenological two-zone model is computationally faster than the multi-zone model. Results indicated that the proposed model is faster than the multi-zone model and can predict the combustion trajectories and the effect of oxygen concentration on lean combustion [193]. A simple schematic of the two-zone combustion model and 2D Φ-T map with combustion trajectory predicted by the model is shown in Fig. 10a, b.

Numerical studies are further conducted to determine the effect of injection timings, injection angles, and fuel concentration distribution in the cylinder. Optimum injection angles are predicted from the simulations, which reduced wall wetting and improved the homogeneity of the mixture. The results also showcased that retarding the injection timing resulted in a more premixed combustion fraction decreasing NOx and PM emissions [194–196].

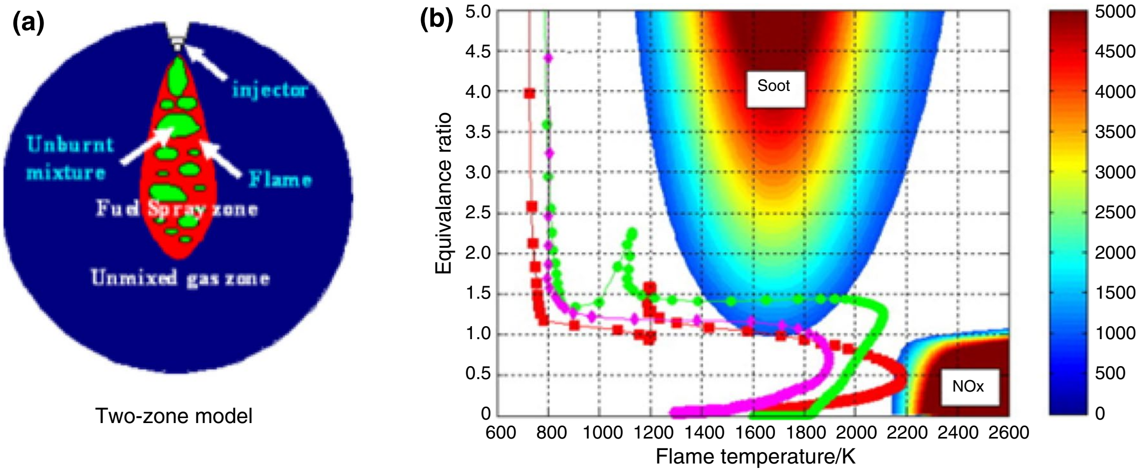

Fig. 10 **a** Two-zone phenomenological model, **b** Model predicted combustion trajectories in 2D Φ-T map; Red: conventional combustion; Green: High dilution combustion; Pink: High energy efficient clean combustion [193]. (Color figure online)

The effects of ethanol premixed combustion with a dual injection strategy are numerically investigated using CONVERGE. Results indicated that the first injection could not be used for combustion control, whereas the second injection provided better control over combustion phasing. The simulation shows that ethanol premixed combustion showed allowable heat release rates due to the thermal stratification acquired due to fuel injections [197]. The direct injection effect of Diesel/ Methanol/ Diesel and Methanol/Diesel/Methanol injection modes are numerically studied. A premixed type of combustion is observed in all three modes of injection. Results showed decreased EISFC, soot, THC, and CO emission for the M/D/M injection mode due to higher amounts of first methanol injection [198]. The effect of late IVC is investigated numerically and found that turbulent kinetic energy and in-cylinder swirl ratio are greatly affected by engine speed. Late IVC decreased effective compression ratio and increased ignition delay at wide operating ranges of the engine. Retarding IVC from 220 to 280 degrees, resulted in a 50% drop in NOx and smoke emissions [199]. Two-dimensional numerical simulations are carried out using *n*-heptane/ air in PCCI combustion mode. The effect of charge and temperature stratification on PCCI combustion with detailed chemical kinetics is investigated. Results indicated that ignition delay is hugely affected by charge, temperature stratifications, equivalence ratio, and temperature gradient. Increased equivalence ratio has little effect on LTHR, whereas an increase in overall temperature gradient advances HTHR and decreases peak [200]. From the above review, it can be concluded that CFD simulations are good at capturing the combustion and emission phenomena in PCCI Combustion.

Conclusions

The authors conduct a detailed literature survey on premixed charge compression ignition engines fueled with conventional and alternative fuels. The present review showcased the fundamentals, development, and practical challenges in PCCI combustion. The critical conclusions from this study are presented below:

1. Premixed charge compression ignition combustion is a potential and reliable option for in-cylinder reduction in engine-out emissions.
2. Various issues such as high CO and HC emissions, the requirement for high EGR, combustion phasing, limited operating range, knocking, combustion noise, and high rate of pressure rise are noticed in PCCI combustion by many researchers.
3. The use of narrow cone angle injectors, split injection strategies, alternative fuels, adjustments in compression ratio, and optimized parameters have improved the PCCI combustion characteristics.
4. Alternative fuels and their blends, such as biodiesels, alcohols, and natural gas, are effective in reducing emissions and improving performance with PCCI combustion in diesel engines.
5. The numerical simulations of PCCI combustion proved that optimized injection strategies, injection angles, and correct fuel mixtures would reduce emissions. Numerical results further indicated a decrease in soot precursors with oxygenated fuel blends.

From this review, it is identified that PCCI combustion is a better option for in-cylinder control of emissions. The shortcomings of PCCI technology can be overcome with suitable fuels and stringent engine parameter control, as seen in the present review. To summarize, PCCI combustion is an efficient low-temperature combustion technology to achieve future emission regulations with fuel flexibility by which energy security can be achieved.

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Declarations

Conflict of interest Authors declare that they do not have any competing interest.

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