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Challenges and Opportunities for Application of Reactivity-Controlled Compression Ignition Combustion in Commercially Viable Transport Engines

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ABSTRACT

Several advanced low-temperature combustion (LTC) strategies have been developed to reduce the harmful emissions from diesel engines. These LTC strategies, such as homogeneous charge compression ignition (HCCI), premixed charge compression ignition (PCCI), and reactivity-controlled compression ignition (RCCI), can reduce engine-out nitrogen oxides (NOx) and soot emissions simultaneously. LTC investigations exhibit several limitations of HCCI and PCCI combustion modes, such as lack of combustion control and other operational issues at higher engine loads, making their application in production-grade engines challenging. RCCI combustion mode exhibited promising results in combustion control, engine performance, and applicability at higher engine loads. The potential of the RCCI concept was demonstrated on different engine platforms, showing engine-out NOx levels below the limits proposed by the emissions regulations, together with ultra-low soot emissions, eliminating the need of after-treatment devices. However, the RCCI combustion mode has several challenges, such as excessive hydrocarbons (HC) and carbon monoxide (CO) emissions at low loads and excessive maximum pressure rise rate (MPRR) at high loads, which limit its effective operating range and practical applications. This review article includes recent advancements in RCCI combustion mode, its potential for using alternative fuels, the effects of different parameters on RCCI combustion mode and its optimization, and the ability of RCCI combustion mode to extend the engine operating limit to reach higher loads, which prevents the application of this concept in commercial applications. The findings of different optical diagnostics have also been included, which have been performed to understand the detailed chemical kinetics of the fuel-air mixtures and the effect of fuel reactivities on the RCCI combustion mode. The first part of this article focuses on these studies, which provide important outcomes that can be used for the practical implementation of RCCI combustion mode in production-grade engines. The second part of this article covers different RCCI combustion mode strategies that can be used to eliminate the restrictions of RCCI combustion mode at high loads. Among the different techniques, dual-mode concepts have been extensively investigated. The dual-mode concept is based on switching between two different combustion modes, typically an LTC mode and conventional compression ignition (CI) combustion mode, to cover the entire operational range of the engine. Many studies showed that the NOx and soot emissions from stationary engines with dual-mode RCCI/CI combustion had substantially improved versus a single-fueled CI combustion mode engine. Results related to the measurements of emissions and performance in transient conditions and driving cycles have also been included, which exhibit promising results for RCCI combustion mode. A comprehensive review on overcoming the challenges and real-world applicability of RCCI combustion mode is not available in the open literature yet. This article includes the results of relevant RCCI combustion mode investigations carried out in single-cylinder and multi-cylinder engines, intending to fill this research gap. Finally, the results from alternative RCCI combustion mode concepts such as the dual-mode, hybrid-RCCI, simulations, and experiments in transient conditions using various driving cycles make this article uniquely relevant for researchers.

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Nomenclature

2-EHN	2-ethylhexyl nitrate
AADI	air-assisted direct injection
AMP	accumulation mode particles
ATAC	active thermo-atmosphere combustion
BC	black carbon
BMEP	brake mean effective pressure
BSFC	brake-specific fuel consumption
CAD	crank-angle degree
CAxx	crank angle position for xx% cumulative heat release
CBG	compressed biogas
CDC	conventional diesel combustion
CHR	cumulative heat release
CI	compression ignition
CL	close-loop
CN	cetane number
CO	carbon monoxide
CO2e	CO_2 equivalent
CPOX	catalytic partial oxidation
DDFS	direct dual-fuel stratification
DDI	dual direct injection
DEF	diesel exhaust fluid
D-EGR	dedicated EGR
DI	direct injection
DICI	direct injection compression ignition
DMCC	diesel methanol compound combustion
DMDF	diesel-methanol dual-fuel
DMDF	dual-mode dual-fuel
DME	dimethyl ether
DOC	diesel oxidation catalyst
DP	dynamic programming
DPF	diesel particulate filter
DPI	diesel pilot injection
DTBP	diterbutyl peroxide
EGR	exhaust gas recirculation
EMC	energy management control
EREV	extended range electric vehicle
EV	electric vehicle
FFV	fuel-flexible vehicles
FHEV	full hybrid electric vehicle
FIP	fuel injection pressure
FoV	field of view
FSN	filter smoke number
FTP-75	federal test procedure
GCI	gasoline compression ignition
GDI	gasoline direct injection
GHG	greenhouse gas
GM	general motors
HC	hydrocarbon
HCCI	homogeneous charge compression ignition
HCHR	homogeneous combustion heat release
HRF	high reactivity fuel
HRR	heat release rate
HTHR	high-temperature heat release
HVO	hydrotreated vegetable oil
HWFET	highway fuel economy test
ICE	internal combustion engine
ICFB	in-cylinder fuel blending
ICT	intake charge temperature
ITE	indicated thermal efficiency
ITEg	gross indicated thermal efficiency
JC08	japanese cycle
LCA	lifecycle analysis
LIVC	late intake valve closing
LPDDI	low-pressure dual-fuel direct injection
LPG	liquefied petroleum gas
LRF	low reactivity fuel
LTC	low-temperature combustion
LTHR	low-temperature heat release
MHEV	mild hybrid electric vehicle
MPC	model-predictive control
MPRR	maximum pressure rise rates
MSR	methanol substitution ratio
NC	next cycle
NFL	natural flame luminosity
NL	natural luminosity
NMP	nucleation mode particles
NOx	nitrogen oxides
NP	nanoparticles
NVH	noise vibration harshness
NVO	negative valve overlap
OC	organic carbon
OL	open-loop
PAHs	polycyclic aromatic hydrocarbons
PCCI	premixed charge compression ignition
PCHR	premixed combustion heat release
PCI	premixed compression ignition
PDFC	piston-split dual-fuel combustion
PHEV	plug-in hybrid electric vehicle
PLIF	planar laser-induced fluorescence
PM	particulate matter
PN	particle number
$PODE_n$	polyoxymethylene dimethyl ether
PPC	partially premixed combustion
PPCCI	partially premixed charge compression ignition
PRF	primary reference fuel
PRR	pressure rise rate
PSD	particle size distribution
RBC	rule-based control
RCCI	reactivity-controlled compression ignition
RDE	real driving emissions
RON	research cetane number
R-RCCI	reverse RCCI
SCR	selective catalytic reduction
SHEV	series hybrid electric vehicle
SI	spark ignition
SMR	steam methane reforming
SoC	star of combustion
SOC	state of charge
SPCCI	spark controlled compression ignition
SR	steam reforming
TC	total carbon
TDC	top-dead-center
TFR	thermochemical fuel reformer
TPM	total particle mass
TPME	thevetiaperuviana methyl ester
UDDS	urban dynamometer driving schedule
v/v	volume over volume
VCR	variable compression ratio
VVT	variable valve timing
WHTC	worldwide harmonized light vehicle test cycle
WLTP	world harmonized light vehicles test procedure

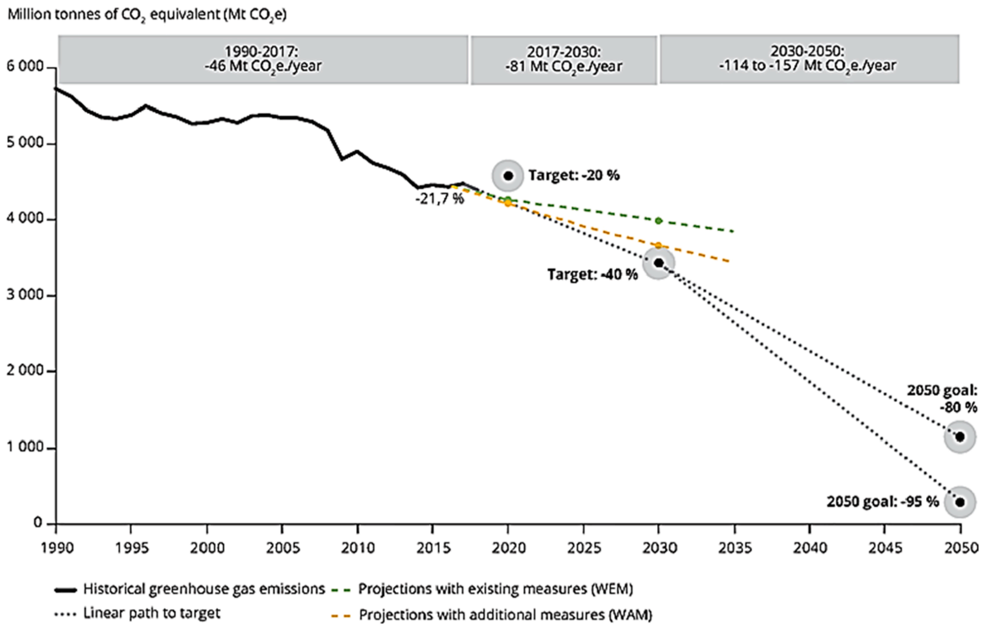


Fig. 1. GHG emission trends, projections, and targets (Reprinted from [6] with permission of European Environment Agency (EEA)).

1. Introduction

The internal combustion engines (ICE) were conceptualized in 1876 [1], more than 145 years ago, and they have played an extremely important role in human development by catering to power generation and transportation. Fossil fuels have traditionally been the main energy source to power the ICEs; however, it is well documented that their production, exploitation, and use have several negative environmental impacts [2,3] and health issues associated with air pollution generated [4]. The energy landscape has exploded in recent years, giving rise to newer options such as synthetic fuels, fuel cells, electric and hybrid vehicles, and renewable fuels. These are aimed to reduce the negative

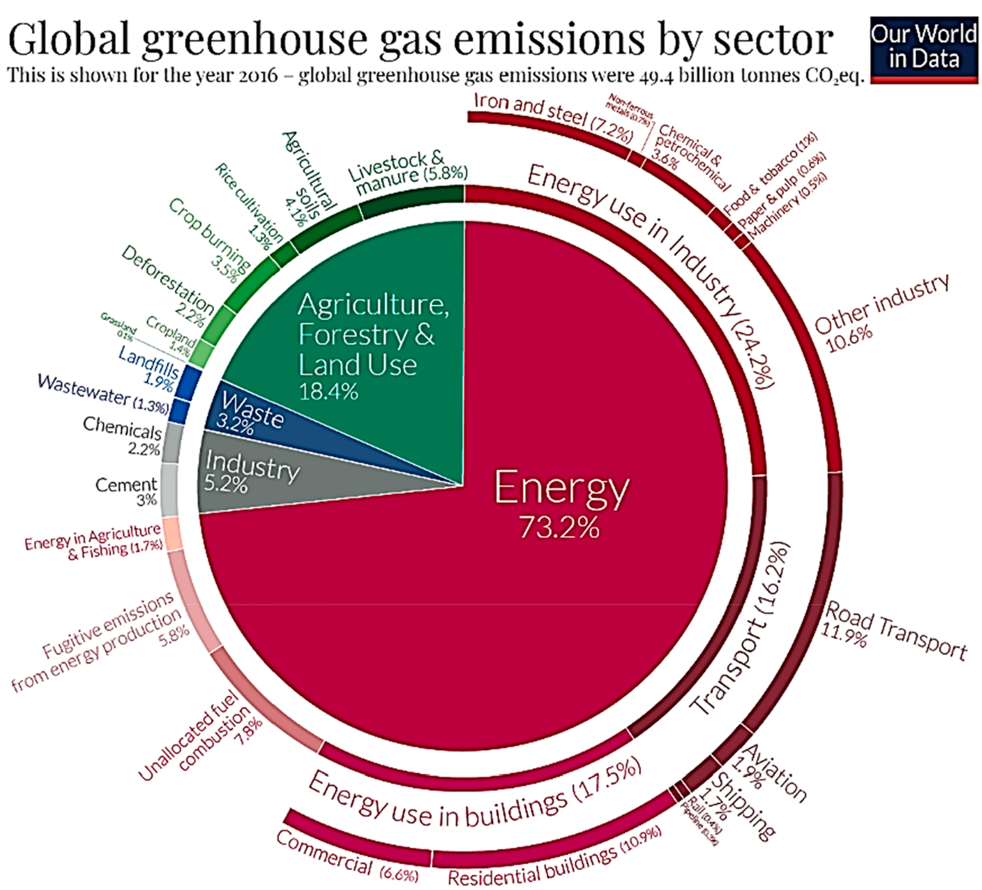


Fig. 2. Global greenhouse gas emissions, sector-wise (Reprinted from [5] with permission of Our World in Data).

impact of pollutants from the transport sector, which currently accounts for ~16.2% of the global greenhouse gas (GHG) emissions [5]. Additionally, to prevent the dire effects of GHG emissions and achieve the goal of a climate-neutral economy by 2050, emissions regulators have imposed stricter emission limits in the transport sector [6–8], thus accelerating the demand from the automotive manufacturers for a cleaner engine-out exhaust from vehicles. Hence, an accelerated optimization of ICEs and the introduction of efficient combustion concepts are vital for reducing emissions in the foreseeable future. This section deals with these aspects related to the automotive sector and presents the current scenario and possible solutions to deal with these burning issues.

1.1. Current Automotive Scenario and Challenges

Knowing that in the short-to-medium term, ICEs would continue to move the world and remain an important power plant for vehicles in the transport industry, active research and development for the improvement in fuel consumption and reduction in pollutants (namely carbon monoxide (CO), hydrocarbon (HC), nitrogen oxides (NOx), and soot/particle matter (PM)) would continue to be important. In particular, the evolution of compression ignition (CI) engines with direct injection (DI) of high cetane fuels would be important since they are highly efficient, widely used, and produce relatively lower CO and HC emissions than their spark ignition (SI) engine counterparts. On the other hand, one of the main challenges for these direct injection compression ignition (DICI) engines would be higher NOx and PM emissions due to their higher combustion temperatures [9,10] and fuel-rich regions [11,12], respectively. A close relationship among these pollutant species is another challenge because of a trade-off. Preventive measures to control one of them promote the formation of the other [13]. Several routes have been discussed to control the emissions of these pollutants, such as the use of alternative fuels [14], exhaust after-treatment systems [15], improving combustion, and using advanced combustion concepts [16]. Among these options, the advanced combustion concepts have significant potential since they provide several benefits over conventional diesel combustion (CDC) without using after-treatment devices for the exhaust gas. After-treatment systems increase vehicle production and operational costs, fuel consumption, and engine complexity and require frequent maintenance [17,18]. The reactivity-controlled compression ignition (RCCI) is a dual-fuel combustion concept aiming to simultaneously reduce the engine-out soot and NOx emissions while maintaining the engine performance and efficiency comparable to the CDC [19].

1.2. Need for Alternative Combustion Concepts

Between 1990 to 2017, global GHG emissions were reduced by ~21.7% (4483 megatons of CO_2 equivalent (CO_2e)) [6]. In 2018, the energy sector reduced CO_2e emissions by 22.2% with respect to 1990; however, the transport sub-sector (the category that provides movement of humans, animals and goods from one location to another) GHG emissions increased ~20% with respect to 1990 [6] (Fig. 1). Among the GHG emissions from the transport sector, road transport contributed 73.45% [5] (Fig. 2). A clear impact of the activities in this sector underscores the need for emission regulations such as EURO VI for light-duty and heavy-duty vehicles and subsequent strategies to develop efficient ICEs that can achieve the target GHG emissions by cutting down on carbon dioxide (CO_2) emissions. Hence, a combination of newly developed combustion strategies such as low-temperature combustion (LTC) and alternative fuels will be hugely beneficial since electrification does not guarantee decarbonization if the main sources of electricity generation are non-renewables [20,21].

CDC has excellent efficiency and engine performance, indicating that its fuel-to-energy conversion is effective [22]. The problem lies in the engine-out emissions. Reducing emissions will then get the ICE to an acceptable emission footprint. The emissions control techniques have traditionally been deployed on two fronts: (i) active control strategies using after-treatment systems and (ii) passive control strategies to control pollutant formation inside the engine's combustion chamber. On the after-treatment side, diesel oxidation catalysts (DOC), diesel particulate filters (DPF), and selective catalytic reduction (SCR) have been deployed to comply with prevailing emission standards. However, they have several disadvantages, which have already been mentioned in Section 1.1. It is worth restating that reducing the emissions during the in-cylinder formation stage would reduce the after-treatment requirement, thus minimizing the overall system complexity and engine cost.

Traditionally, a large number of pollutant formation control strategies – such as optimized fuel injection strategies, including the use of high fuel injection pressures (FIP), fuel injection timing optimization, multiple fuel injections, use of exhaust gas recirculation (EGR), increased in-cylinder turbulence (increased in-cylinder motion and turbocharging), and redesign of the combustion chamber and injection system –have been deployed with varying degrees of success for controlling the NOx and the soot emissions. A combined in-cylinder combustion optimization and after-treatment device approach has also been explored to control the emissions to meet the regulatory requirements [23,24]. These methods struggle primarily because of the intrinsic trade-off between the soot and the NOx in diesel engines. More often than not, managing to reduce one pollutant increases the other. Primarily, NOx is formed through a thermal mechanism (although other mechanisms also exist and can be prevalent during some combustion modes) where nitrogen and oxygen in the cylinder combine in the presence of high temperature and adequate residence time. Hence reducing the combustion temperatures and duration can reduce the NOx formation. Nonetheless, this has a counter effect on soot formation since the temperature and residence time are not high enough to burn off the soot particles. LTC strategies that deal with the soot-NOx trade-off have been developed over time and can be integrated with the use of advanced fuels to mitigate these emissions. In particular, many LTC strategies have been developed, which improve the fuel-to-work conversion efficiency while providing low soot and NOx emissions [25]. It is worth noticing that CO_2 equivalent emissions need to be reduced by more than half of the current levels by 2050 to avoid severe global warming [26]. A single pathway is not enough to reach this emission goal. Hence it is important to address more than one strategy at a time. Another path explored for more than two decades is using oxygenated fuels to achieve virtually soot-free combustion [27,28]. However, the increase of the oxygen content in the fuel composition (as with some alcohols) can increase NOx emissions and peak pressures if suitable strategies to address those issues are not used [29,30]. Since their characteristic properties also differ from commercially available fossil fuels, dedicated calibration and fuel injection strategies need to be developed for alternative fuels. However, research has shown that combining both the LTC and alternative fuels in the same system can benefit both emissions and engine efficiency.

1.3. Evolution of Different LTC Concepts

In the past few decades, LTC modes have gained significant attention from researchers (correlated to the increasing number of publications in the field) due to their excellent capabilities in terms of emission reduction, especially NOx and soot. LTC modes including homogeneous charge compression ignition (HCCI), partially premixed charge compression ignition (PPCCI), and premixed charge compression ignition (PCCI), have the potential to reduce NOx and PM emissions simultaneously without a "soot-NOx dilemma" [31–33]. In all LTC modes, relatively lower in-cylinder combustion temperatures are common, which is the main reason for extremely low NOx emissions. To explore other features of LTC, Akihama et al. [34] applied ultra-high EGR in a conventional diesel engine. The objective of this study was to explore the potential of EGR for the simultaneous reduction of NOx

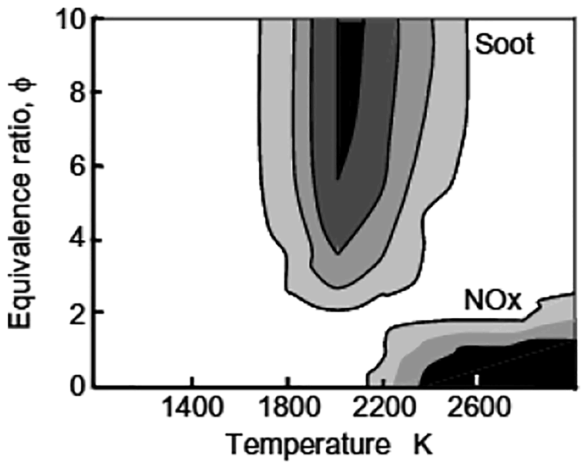


Fig. 3. φ-T diagram for identifying soot and NOx emission zones (Adapted from [34]).

and soot. They reported a 'soot-NOx dilemma' in conventional diesel engines, which could be avoided only by achieving combustion outside the NOx and soot zones in Fig. 3.

The authors indicated that smokeless engine combustion could be achieved for any equivalence ratio at in-cylinder temperatures below 1500 K. However, a more sensitive behaviour of soot emissions was observed at in-cylinder temperatures above 2000 K. They reported that soot formation increased rapidly at excessively richer mixtures. The NOx formation increased significantly at relatively lower loads ($1<\varphi<2$). In this regard, Beatrice et al. [27] reported an important finding that oxygenated fuels exhibit different soot-NOx characteristics than conventional diesel. They reported that using oxygenated fuels and high EGR could drastically reduce both soot and NOx. Various LTC methodologies were extensively explored by researchers worldwide to meet stringent emission norms. LTC strategies, as the name indicates, promote a combustion process at lower temperatures. Therefore, it involves reducing the flame temperature and allowing sufficient air-to-fuel mixing to increase the fuel-air mixture homogeneity [35]. This method theoretically reduces NOx and soot because these pollutants are strongly influenced by the flame temperature and the equivalence ratio [36] while preserving thermal efficiency. This concept virtually combines the benefits of CI and SI engines. The necessary in-cylinder conditions can be achieved by early fuel injections [37], improved fuel atomization [38], or even using fuels with lower CN [39,40], alternative fuels, or a combination of diesel- and gasoline-like fuels [41]. This allows more time for fuel-air mixing before the start of combustion (SoC).

Similarly, EGR-controlled combustion can trigger this combustion mode with injections close to the top-dead-centre (TDC) [42]. LTC addresses the conventionally inferior capacity of diesel-fueled engines to prepare a premixed fuel-air mixture before the SoC, caused by the higher fuel viscosity and lower fuel volatility [25]. Like other combustion strategies, the LTC use various alternative fuels, providing additional freedom for controlling combustion. Alternative fuels reduce the dependence on existing petroleum reserves [43] and open up the use of new fuels having special properties such as higher fuel oxygen or absence of carbon. This can lead to extremely low soot emissions and increase the tolerance to higher EGR for reducing NOx emissions. Two key concepts emerge in this LTC method: (i) reduced in-cylinder temperature and (ii) improved in-cylinder fuel-air mixture formation. The soot-NOx trade-off associated to the conventional CI combustion mode is another important aspect that promotes LTC engine development. Even though the LTC strategies address the soot-NOx trade-off issues and maintain conventional diesel-like efficiencies, they have challenges as well, such as higher CO and HC emissions [44], lack of ignition timing control over the high and low load extremes [45], and increased combustion noise [46].

A broad range of technologies is covered in the ambit of LTC, including HCCI, PCCI, partially premixed combustion (PPC), premixed compression ignition (PCI), and gasoline compression ignition (GCI), and RCCI. Among these, the RCCI combustion mode is a strategy that is extensively reviewed in this article. Besides these combustion-specific methods, integration of LTC in hybrid powertrains has also been explored [47]. Among many LTC strategies, few are described here, which have assisted in the evolution of the RCCI combustion mode concept, focusing on their capabilities, the drawbacks encountered, and how they have helped the evolution of the RCCI combustion mode.

1.3.1. HCCI Combustion Concept

HCCI is the most common technique for achieving LTC by combining the premixing of fuel (similar to gasoline engines) and the compression ignition (similar to diesel engines) [48]. Originally coined by Najt and Foster [49], it consists of homogeneous fuel, air, and EGR mixture. An early injection of fuel to form a well-mixed charge is required to achieve HCCI combustion in gasoline engines. In diesel engines, early direct injection of fuel in the intake stroke can be implemented, which provides enough time for fuel-air mixing in the combustion chamber. The fuel-air mixture is compressed in the compression stroke, increasing its temperature to the autoignition temperature, until a volumetric combustion is initiated at multiple sites in this homogeneous fuel-air mixture. The ignition is controlled by the composition of the fuel-air mixture and the in-cylinder temperature. A lean and homogeneous fuel-air mixture auto-ignites without a spark because of the increased in-cylinder temperature at the end of the compression stroke. Local temperatures are kept at low levels, and there is no high-temperature flame front [50]. It experiences lower throttling losses, favouring higher thermal efficiency [51], fuel versatility [52], and lower CO_2 [53], NOx, and PM [54] emissions. Due to these attributes, the HCCI combustion mode ensures smoother LTC, leading to significantly lower NOx and PM emissions [55]. The gasoline HCCI concept was first demonstrated by Onishi et al. [56] in 1979, who named it 'Active Thermo-Atmosphere Combustion (ATAC).' They successfully achieved stable combustion in a gasoline-fueled 2-stroke engine and reported significant improvement in emission characteristics, engine noise and vibrations. They suggested that a relatively leaner mixture combustion was the main reason for these observations. However, the applicability of this novel combustion technique only up to part-load was a major limitation. After this pioneering work, several researchers, including Thring [57] (1989), Christensen et al. [58] (1997), Stanglmaier et al. [59] (1999), and Maurya et al. [60] (2011), successfully demonstrated this combustion concept in gasoline-fueled engines. In most of these research studies, gasoline-like fuels were used to achieve HCCI combustion, demonstrating superior emission characteristics compared to the conventional SI combustion mode. Researchers also explored HCCI combustion using diesel-like fuels to resolve the soot-NOx trade-off observed in CDC engines. Some of the most influential factors were the fuel injection strategies [61] and other characteristics of the fuel injection system, such as an injector and the injection pressure [62]. The main challenges in achieving diesel-fueled HCCI combustion were achieving a homogeneous fuel-air mixture, avoiding the fuel wall-wetting, controlling the auto-ignition, and excessive pressure rise rates (PRR). In particular, the PRR was crucial to avoid potential damage to the engine components and control the noise and vibration issues [63]. The combustion control brings another set of issues. Combustion can be affected by small variations in the charge composition or system temperature [64]. Singh et al. [65] reported combustion characteristics of diesel-fueled HCCI combustion in which they used an external mixture preparation device for preparing a homogeneous charge. They reported that combustion characteristics of the HCCI combustion mode were dominantly affected by the charge homogeneity. They also explored the performance and emission characteristics of diesel-fueled HCCI combustion engines and reported superior emissions characteristics but slightly degraded engine performance. A detailed comparison of PM emission characteristics of diesel-fueled HCCI combustion engine and CDC engine exhibited relatively lower PM and trace metals from the HCCI combustion mode [66]. In another study carried out by Singh et al. [67], diesel and biodiesel blends were used to achieve the HCCI combustion mode. They reported almost similar HCCI combustion characteristics of diesel-biodiesel blends as that of mineral diesel. However, relatively inferior biodiesel properties affected the engine performance and emissions adversely. Several test fuel blends were also used to explore the effects of fuel properties on the HCCI combustion engine, including mineral diesel-alcohol, mineral diesel-gasoline, mineral diesel-kerosene, and mineral diesel-biodiesel. They reported that fuel properties, especially volatility, affected the HCCI combustion, resulting in superior engine performance and emission with increased fuel volatility. Nonetheless, HCCI is one of the few LTC concepts that has been commercially implemented. In the Mazda Skyactiv X engine, a strategy denominated spark controlled compression ignition (SPCCI) uses a lean burn compression ignition combustion with gasoline, spark-ignited local combustion (to increase in-cylinder temperature and propitiate auto-ignition) and a high compression ratio (CR), leading to HCCI combustion that is stable and improves the fuel economy by ~5% [68].

Although the HCCI combustion exhibited a sharp reduction in both NOx and PM emissions, relatively higher HC and CO emissions due to lower in-cylinder temperatureand the use of high EGR adversely impacts the popularity of HCCI combustion mode for deployment in production-grade engines. This also leads to an overall reduction in the thermal efficiency of the HCCI combustion engines. Lack of direct combustion control and excessive PRR are two critical challenges of HCCI combustion, limiting its application at higher engine loads. This issue assumes a more serious dimension in the mineral diesel-fueled HCCI combustion engine due to its lower volatility and auto-ignition temperature. It has been reported in several studies that mineral diesel-fueled HCCI combustion results in either too advanced or too retarded combustion phasing, leading to lower thermal efficiency. These drawbacks motivated researchers to develop a new LTC strategy, known as PCCI combustion mode.

1.3.2. PCCI Combustion Concept

PCCI combustion mode evolved from the HCCI combustion mode to address the difficulties and challenges of HCCI. Like its predecessor, the PCCI combustion mode reduces soot and NOx, but not as much as the HCCI combustion mode [69]. HC and CO emissions are lower in PCCI combustion mode than in HCCI combustion mode [70]. Unlike the HCCI combustion mode, the PCCI combustion mode is not fully homogeneous. Additionally, PCCI combustion offers relatively better control of the SoC and combustion duration because of the charge dilution by highEGR to delay the ignition and increase the mixing duration [71]. Higher EGR levels in PCCI combustion mode affect combustion stability at high engine loads, which could be addressed by early fuel injections strategies. However, this has a counter-effect of reducing the thermal efficiency [72]. Another negative effect of using high EGR levels is the increased fuel consumption and higher CO, HC [73], and PM emissions due to reduced availability of oxygen [16]. Musculus et al. [74] proposed a conceptual LTC model describing spray formation, mixing, ignition, and pollutant formation. Their analysis stated that both HC and CO highly depend on the mixture distribution. A narrow range of equivalence ratios providing a low CO yield can make CO emission more problematic than HC emissions in early-injection PCCI combustion engines. Solutions such as increased boost pressure [70,75], high FIP, multiple injections to reduce the maximum heat release rate (HRR), and enhanced mixing to reduce combustion noise and soot [76] have been proposed to extend the engine operating range. The effect of EGR and variations in injection parameters on the PCCI combustion mode has been extensively studied by various researchers worldwide [77–80], with a common observation of a simultaneous reduction in NOx and PM by early fuel injection timings. High EGR in PCCI combustion mode also led to a slight increase in brake-specific fuel consumption (BSFC). Torregrosa et al. [79] reported a reduction in the combustion noise by using a pilot injection in the PCCI combustion engine; however, brake mean effective pressure (BMEP) decreased significantly as the pilot injection quantity was increased. Emission reduction by variable valve timing (VVT) and early start of injection (SoI) timing as 46 and 30° bTDC were achieved by Murata et al. [69] and Torregrosa et al. [79], respectively, at the cost of a significant reduction in the engine torque.

PCCI combustion mode can be realized using a variety of fuels [81], including fuel blends with lower cetane numbers (CN) fuels such as gasoline, to enhance fuel-air mixing before the SoC [82] and to extend the engine operating range [83]. Fuels with high CN are favourably combined with a longer ignition delay of the PCCI combustion strategy. Lilik and Boehman [84] tested two synthetic diesel fuels produced by a low- and high-temperature Fischer-Tropsch process, having 81 and 51 CN, respectively. They reported that HC and CO emissions were reduced by 32% and 31%, respectively, for the CN 51 fuel than baseline diesel and 80% and 74% for the CN 81 fuel. CN 81 fuel also maintained PM and NOx emissions at the same level as baseline diesel due to a shorter and less intense premixed combustion that preceded a mixing controlled combustion phase. Alternative fuels such as alcohol blending seem feasible for achieving more efficient and controllable PCCI combustion. Some researchers used ethanol-diesel blends in CI engines to attain low temperatures during combustion. Mohammadi et al. [78] explored the possibility of very clean combustion using ethanol in a partial PCCI combustion engine. They realized that the utilization of cooled EGR with an early pilot injection leads to a significant PM-NOx trade-off. Park et al. [80] investigated a narrow spray angle injection strategy using bioethanol blended diesel in a conventional diesel engine. They reported that the premixed combustion phasing decreased with increased bioethanol content in the fuel. A significant reduction in HC and CO emissions was also reported.

More recent works evaluated the potential of bio-origin fuels such as hydrotreated vegetable oil (HVO) [85] and diesel-biodiesel blends [86] in PCCI combustion mode. The sensitivity to the mixture dilution and conditions were evaluated [85] by varying the boost pressure and EGR levels. The results show that the increasing EGR from 0% to 39% reduces NOx emissions by almost 7 g/kWh with a penalty increase of ~ 0.3 g/kWh PM. An optimized condition balances both pollutants as 0.5 g/kWh of PM and 1.5 g/kWh of NOx, which is lower than mineral diesel. Specifically, they emphasized that HVO responds well to higher EGRs (25%) and lower boost pressures (130 kPa). A similar trends of reduction in both NOx and PM emissions [86] were also found for increasing the percentage of waste cooking oil biodiesel in test fuel. Additionally, they observed some reduction in CO and HC emissions (~0.05% and ~25 ppm, respectively). Biofuels like HVO, free from cycloalkanes and aromatics and with the presence of oxygen in their molecule, play a significant role in reducing soot formation, making it easier to control the NOx emissions.

Many research studies also explored another version of LTC, namely PPC. In PPC, a stratified fuel-air mixture is used. The degree of fuel stratification can be controlled by varying the SoI timing and other fuel injection parameters, such as multiple fuel injections [87]. In PPC, both premixed and diffusion flames are present after auto-ignition in several locations in the combustion chamber [88,89]. PPC can be applied to both gasoline and diesel-fueled engines. Kalghatgi et al. [90,91] applied PPC in light- and heavy-duty engines and simultaneously achieved superior engine efficiency and lower exhaust emissions. It was suggested that gasoline with the research octane number (RON) of around 70 is highly recommended for the optimum PPC. Using this fuel, PPC results in a trade-off between combustion stability at low loads and HRR at high loads [92–94]. Han et al. [95,96] performed PPC experiments using diesel/gasoline blends. PPC exhibited relatively higher engine efficiency and lower exhaust emissions than CDC. However, relatively inferior combustion stability at low loads and high soot emissions at high loads are two major concerns of PPC. Liu et al. [97] also performed PPC investigations to resolve these issues. They used a blend of polyoxymethylene dimethyl ethers ($PODE_n$) and gasoline in a heavy-duty diesel engine for achieving PPC. They reported significantly improved soot-NOx trade-off in the PPC using gasoline/$PODE_n$ blends without any fuel efficiency penalty. Due to the addition of $PODE_n$, the combustion efficiency and combustion stability also exhibited a significant improvement, especially at low loads. Although preliminary LTC approaches demonstrated significant potential in reducing NOx and PM, these combustion strategies have many issues related to engine performance compared to baseline CDC and, more specifically, their applicability in production-grade engines. Several techniques such as EGR [98], VVT [99,100], variable compression ratio (VCR) [101], and intake air temperature variations [60] have been investigated extensively to overcome these challenges. These interventions resolved some of the issues related to the engine performance and emission characteristics; however, these techniques could not resolve their applicability at higher engine loads. In most LTC studies involving mineral diesel as test fuel, it was seen that the high reactivity of mineral diesel was the main hurdle for extending the extreme load limits (low and high loads) of the LTC concept. In LTC, many researchers advocated using low octane fuels (diesel-like fuels) at lower engine loads.

However, high octane fuels (gasoline-like fuels) could be used at high loads to achieve stable combustion, superior engine performance, and lower emissions [102]. Lu et al. [103] explored the effect of the addition of gasoline-like fuels in PCCI combustion mode. They performed experiments using the blends of butanol and gasoline vis-à-vis pure butanol fuelled PCCI combustion mode. They reported that adding a small amount of gasoline in butanol resulted in superior PCCI combustion mode by increasing the maximum pressure and temperature, leading to lower HC and CO emissions from PCCI combustion mode. Following the preliminary research efforts of Bessonete et al. [102], Inagaki et al. [104] demonstrated yet another version of LTC, namely dual-fuel PCI combustion. Two fuels with different reactivity were injected into the combustion chamber using two separate injectors in the PCI combustion. The fuel quantities were varied depending on the engine load to control the overall mixture reactivity in the combustion chamber. The results exhibited excellent control over various combustion parameters, which was impossible with previous LTC strategies. PCI combustion yielded extremely low NOx and PM emissions and showed significant potential for further improvement. Shim et al. [105] carried out a detailed investigation of dual-fuel PCCI combustion mode using a combination of compressed natural gas (CNG) and mineral diesel. They compared the combustion, performance and emission characteristics of dual-fuel PCCI combustion mode to single fuel PCCI combustion mode. They reported that combustion control was easier in dual-fuel PCCI combustion mode, which can be done by adjusting the premixed amount of the second fuel. They also reported that the dual-fuel PCCI combustion mode exhibited superior engine performance and lower CO and HC emissions than the single fuel PCCI combustion mode. Hence, many researchers explored this combustion strategy, which further developed as the RCCI combustion mode. The next sub-section provides insights into the fundamentals of RCCI combustion mode and its evolution with time.

1.4. RCCI Combustion Concept

Previous efforts to develop a commercial LTC engine failed due to certain limitations, especially for high-load applications. The high chemical reactivity of mineral diesel is the major concern for diesel-fueled LTC engine development, resulting in inferior control over combustion. Bessonette et al. [102] suggested that the in-cylinder reactivity of the fuel-air mixture should be varied to achieve stable LTC at different engine loads. Based on these recommendations, Kokjohn et al. [106] proposed the concept of RCCI, wherein gasoline was the premixed low reactivity fuel (LRF), and diesel was the directly injected high reactivity fuel (HRF) for achieving reactivity stratification in the cylinder. In their work, 50% thermal efficiency could be achieved along with virtually-zero NOx and soot emissions. In RCCI combustion mode, a LRF was supplied in the intake manifold, mixed with the intake air to form a fully premixed intake charge. The HRF was directly injected to ignite the premixed intake charge; however, the directly injected fuel did not completely mix and remained stratified before the SoC.

In the RCCI combustion mode, the combustion phasing is determined by the global reactivity of the mixture, given by the LRF to HRF ratio. At the same time, the stratification of the ignition delay controls the combustion timing (duration and phasing). Initiation of combustion takes place at HRF-air mixture locations, and the combustion progresses from high-to-low reactivity regions, which allows a controlled sequential ignition. It provides superior combustion control than other LTC strategies. RCCI combustion mode exhibits low soot and NOx emissions with enhanced engine efficiency than CDC for various speeds and loads. This combustion strategy requires highly premixed air-fuel mixtures, which can be achieved by using two injections of HRF in the combustion chamber. The first injection is used to increase the in-cylinder reactivity. The second injection improves the mixing and acts as an ignition source, resulting in lower soot formation [107]. By altering the global charge reactivity in combination with one or more direct injections, complete control over the combustion phasing and HRR can be accomplished [106,108-113]. This allows RCCI combustion engine operation with combustion efficiencies over 97% across a wide load range (from 4 bar to 20 bar BMEP) [114].

Additionally, gross indicated thermal efficiencies (ITEg) approaching 60% have been demonstrated experimentally. The dual-fuel strategy allows easier control of the combustion phasing, which is regulated by the local concentration and the injection timing of the HRF . The combustion duration is controlled by the mixture reactivity gradient, which can be tailored to reduce the PRR and combustion noise. Cycle-to-cycle control would be relatively easier to implement in production-grade engines with proper feedback.

RCCI engine operation has been expanded by the direct dual-fuel stratification (DDFS) combustion strategy, which utilizes two direct injectors, each of which dedicated to injecting either the LRF or the HRF. These are centrally mounted in the combustion chamber to achieve a clean and efficient RCCI combustion in a heavy-duty engine [115,116]. RCCI combustion mode relies heavily on the fuel reactivity stratification gradient to achieve clean and efficient combustion, and a broad range of fuels may be used. RCCI combustion is essentially an inherently fuel-flexible LTC strategy. Research shows that the RCCI combustion mode offers relatively more flexible control over combustion since fuel concentration and reactivity stratifications regulate the HRR [117]. Due to wide availability, gasoline and diesel have been used as the LRF and HRF, respectively, in most previous RCCI research [108,113,108-121]. In many studies, primary reference fuels (PRF) as iso-octane and n-heptane have also been studied as the LRF and HRF, respectively [104, 122-125]. Researchers also explored the potential of alcohols as LRF [120,122,126-130] and biodiesel as HRF for RCCI combustion mode [131,132]. However, mobile engine applications may preclude the ability to carry large quantities of two separate fuels. As a result, cetane improvers, such as di-tert-butyl peroxide (DTBP) and 2-ethyl-hexyl nitrate (EHN), have also been studied as low quantity additives to condition the LRF to perform as an HRF to approximate single fuel RCCI combustion [120,127,130,131,133]. Splitter et al. [133] carried out RCCI combustion investigations using two fuels having large reactivity differences and optimized the in-cylinder fuel stratification. They observed a simultaneous reduction in NOx and PM emissions and achieved ~60% indicated thermal efficiency (ITE). Liu et al. [134] explored the ignition and flame development in RCCI combustion mode. They used several fuel supply strategies to achieve different fuel stratifications. They reported that the auto-ignition and flame front propagation could be controlled by regulating the degree of fuel stratification. Kokjohn et al. [135,136] carried out detailed investigations of RCCI combustion mode using optical diagnostic and simulation. They reported

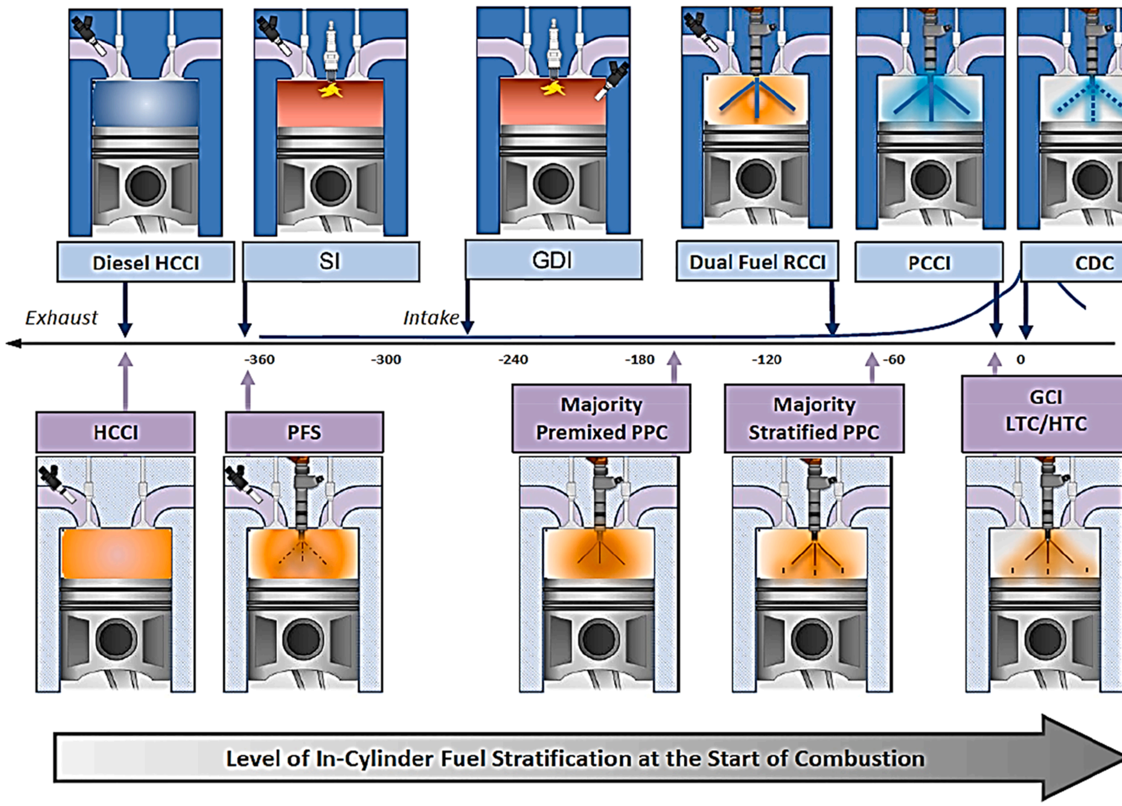


Fig. 4. Comparison of various conventional and advanced combustion strategies with respect to fuel stratification (Adapted from [143]).

that the fuel-air mixture reactivity stratification dominantly controlled the ignition and combustion events. However, the role of mixture concentration stratification and thermal stratification was less influential. That is why the reactivity and concentration stratification focus on RCCI combustion mode to extend the engine load range. The mixture reactivity gradient can be enhanced by enlarging the reactivity difference between the premixed and DI fuels [137–140]. Many studies have reported that the RCCI combustion mode exhibits higher thermal efficiency due to shorter combustion duration, increased specific heat ratio, and decreased heat transfer losses than conventional CI combustion [19, 141].

Olmeda et al. [142] suggested that the primary reason for the relatively higher efficiency of all LTC modes is common in which mixture homogeneity plays an important role because this leads to rapid combustion events. Under optimized combustion phasing, this results in higher fuel-to-work conversion efficiency. The uniform charge inside the combustion chamber also results in similar global and local temperature distribution. This is useful in reducing the localized effect of temperature distribution in heat transfer from cylinder walls. LTC strategies have emerged as pathways for achieving high engine efficiency and relatively cleaner combustion than conventional combustion engines in the last few decades. In most LTC techniques, the thermodynamic cycle efficiency can be increased by idealizing the heat release during combustion towards constant-volume energy conversion. In addition to the constant-volume approach, LTC strategies also result in relatively lower heat loss from the cylinder walls due to significantly lower peak of in-cylinder temperatures. This feature of LTC is also reflected in NOx emissions. Improving mixture formation and spray performance reduces PM formation by reducing peak equivalence ratios during combustion. Numerous strategies have been developed to achieve clean and efficient combustion, as summarized in Fig. 4 [143].

Despite the great potential of the RCCI combustion mode to operate at high engine loads, it required a high degree of fuel stratification to avoid excessive HRR. In the absence of fuel stratification at higher engine loads, the fraction of diffusion phase combustion becomes dominant in RCCI combustion mode, leading to higher soot emissions [125].

2. Evolution of RCCI Combustion Concept

RCCI combustion mode can be considered a special dual-fuel combustion in which an appropriate fuel pair can be used under optimized operating conditions. The dual-fuel combustion strategy is not new. It was introduced in the 1970s as ATAC. The basic difference between the two dual-fuel combustion modes is the methodology of fuel introduction. In ATAC, the LRF is directly injected into the combustion chamber, and a two-stage ignition assists combustion. In the ATAC concept, a hot (thermal) atmosphere with (active) combustion products and radicals is created by igniting a small amount of premixed HRF, which dominantly affects the combustion of LRF.

In the RCCI combustion mode, an LRF is supplied through the intake port, forming a premixed charge. This premixed fuel-air mixture is then ignited using the directly injected HRF [144,145]. The HRF is delivered through the intake port to form a premixed charge [146–149].

In ATAC, the combustion parameters at varying engine loads can be controlled effectively by the fuel injection parameters, such as the fuel injection timing of the LRF. However, the RCCI combustion mode is more popular due to its engine performance improvement and emission reduction potential. Hence, the RCCI combustion mode has been extensively explored experimentally and numerically [150–153]. Recent studies focused on extending the engine load window from low to high load and then to full load [154–156] in RCCI combustion mode. Cooper-Bessamer applied the dual-fuel concept, and they demonstrated a gas-diesel dual-fuel engine in 1927 for the first time [157]. However, a successful dual-fuel concept was first demonstrated commercially in 1940 by the National Gas & Oil Engine Company Limited, England [158]. Their dual-fuel engine model used a dual-fuel combustion strategy in which a direct-injected diesel ignited a premixed natural gas-air

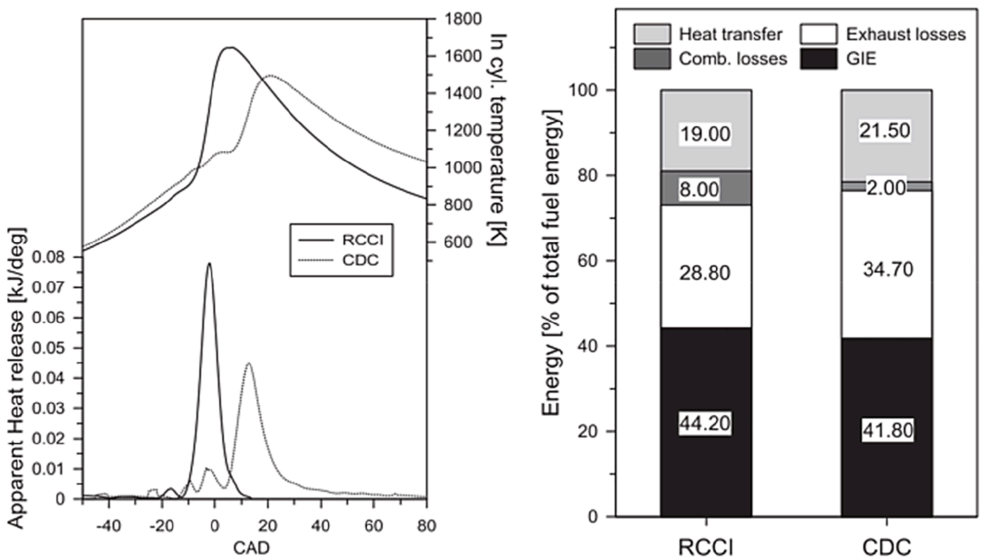


Fig. 5. In-cylinder temperature, apparent heat release (left), and energy distribution (right) for RCCI combustion mode gasoline engine vis-à-vis CDC engine (Reprinted from [142] with permission of Elsevier).

mixture. In this concept, the pilot injection of mineral diesel acted as a "spark plug," allowing sufficient ignition energy at multiple ignition sites in the premixed charge. This resulted in flame propagation, which combined the ignition mechanisms of both conventional SI and CI engines.

Most dual-fuel engines use gasoline as the premixed fuel [159], and, in most of the industry-standard dual-fuel modern engine combustion strategies, a diesel pilot ignition (DPI) is used with gaseous fuels [160–162]. The basic concept of RCCI combustion mode has become popular among researchers because of its potential for achieving superior engine performance and significantly lower emissions, especially NOx and soot. Previous studies have shown that the RCCI combustion mode can effectively utilize many renewable fuels, such as biofuel [163–167], $PODE_n$ [168,169], and methanol [170,171]. Olmeda et al. [142] performed RCCI combustion mode experiments using different LRF (gasoline and E85) and compared the heat transfer characteristics to the baseline CDC combustion engine (Fig. 5). These experiments were performed in a single-cylinder light-duty research engine equipped with several thermocouples in the cylinder head and liner (Fig. 6). They reported that the heat transfer characteristics of both LRFs were almost similar; however, both test fuels exhibited relatively lower heat transfer compared to the baseline CDC combustion. The exhaust losses were slightly higher for the E85 due to the longer combustion duration. This effect was balanced by higher combustion efficiency, proving that both LRFs can be applied in RCCI combustion mode and would deliver similar energy use and efficiency results. The authors also performed an energy analysis of the gasoline-diesel fueled RCCI combustion mode and CDC mode engines focusing on their energy balances. The RCCI combustion mode engine exhibited superior energy capacity from the combustion process. Relatively shorter combustion duration was responsible for this trend, which reduced 13% heat loss with respect to the CDC engine. Relatively lower enthalpy of exhaust gases from the RCCI combustion mode was another important factor, resulting in lower exhaust losses. These lower heat losses also positively affected the combustion efficiency.

Jia and Denbratt [172] investigated the combustion characteristics of mineral diesel-methanol fueled RCCI combustion mode at high engine loads. They reported ultra-low soot and NOx emissions. Singh et al. [173] performed comparative investigations of RCCI and PCCI combustion modes to baseline CI combustion mode using critical parametric analyses. They reported that RCCI combustion mode was comparable to CI combustion mode at low loads. However, the RCCI combustion mode exhibited superior engine performance at higher engine loads and significantly lower NOx and PM emissions than PCCI and baseline CI combustion modes (Fig. 7).

In a similar investigation by Han et al. [174], different LTC modes, namely PCCI, HCCI, and RCCI, were compared to baseline CI combustion mode. They used n-butanol and mineral diesel as test fuels in different combustion modes. They reported that all LTC modes emitted significantly lower NOx and soot than baseline CI combustion mode. RCCI combustion mode exhibited relatively higher efficiency and superior combustion control than other LTC modes.

The potential of using single fuel in RCCI combustion mode to reduce the system complexity has also been explored. In this strategy, an LRF can also be used as HRF with a small amount of cetane improver such as DTBP, 2-EHN, etc. [122]. Splitter et al. [133] used gasoline as LRF and gasoline with DTPB as HRF. They compared single fuel RCCI combustion mode results with dual-fuel (gasoline/mineral diesel) RCCI combustion mode. They reported slightly improved engine performance of the single fuel RCCI combustion mode. This might be due to the higher reactivity gradient generated by DTPB, which led to relatively lower low-temperature heat release (LTHR). A lower LTHR results in lower compression work, leading to higher engine efficiency. In another study by Hanson et al. [175], single fuel gasoline-gasoline +3.5% 2-EHN fueled RCCI combustion mode was compared to gasoline-diesel-fueled RCCI combustion mode. They reported that adding a cetane improver resulted in relatively faster high-temperature heat release (HTHR) of single fuel RCCI combustion mode, leading to a relatively shorter combustion duration than dual-fuel RCCI combustion mode. Mohammadian et al. [176] explored this strategy using isobutanol as LRF and isobutanol+20% DTPB as HRF and reported similar RCCI combustion mode characteristics to dual-fuel RCCI combustion mode. Wang et al. [161] also used isobutanol/isobutanol doped with a DTPB fuel pair in a heavy-duty engine running under RCCI combustion mode. They reported that isobutanol required more DTPB for generating a reactivity gradient. Although the combustion characteristics of both alcohol and gasoline were similar in RCCI combustion mode, alcohols required higher doping with cetane improver due to their lower CN of alcohols, especially butanol [122,177]. A reverse single fuel strategy was also explored in a few studies, in which HRF was used as the LRF [178,179].

Although PPC and RCCI combustion modes have significantly improved combustion, performance, and emission characteristics than baseline CDC and other LTC modes, RCCI combustion mode has several technical challenges. Results of previous studies showed that the RON of

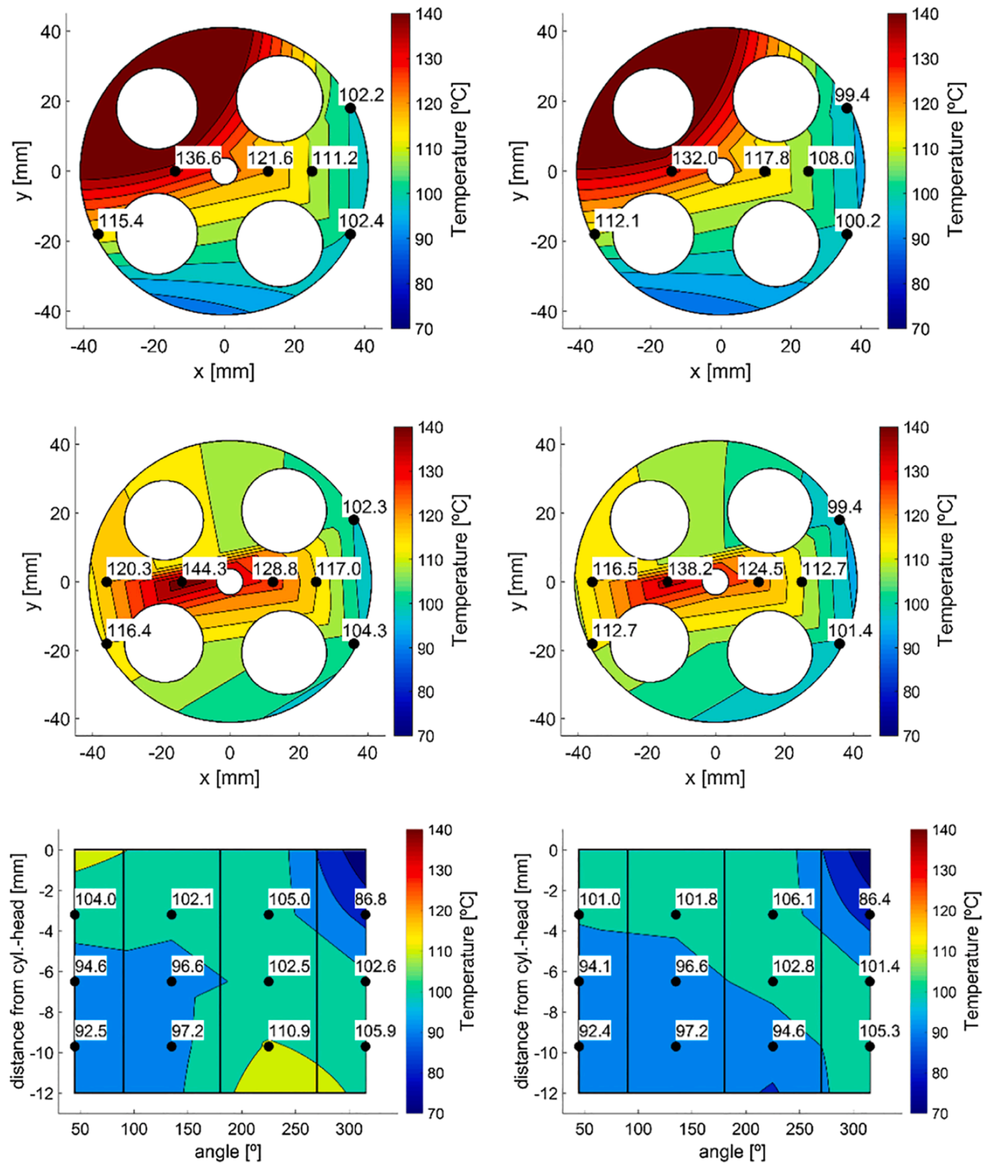


Fig. 6. Measured cylinder head temperature surfaces at 7 mm (upper graphs) and 4 mm (middle graphs) from the fire deck and liner (lower graphs) temperatures for RCCI combustion mode gasoline engine (left) and RCCI combustion mode E85 engine (right) (Reprinted from [142] with permission of Elsevier).

the fuel should be changed depending on the engine load. The RON indicates how much compression the fuel can withstand before auto-igniting; the higher the RON, the higher pressure would be needed for ignition. For effective RCCI combustion mode at low loads, RON should be reduced to enhance combustion stability. Similarly, a higher RON should accelerate the combustion at medium and high loads. In RCCI combustion mode, a fraction of premixed LRF cannot be completely oxidized due to the dominant charge cooling effect near the cylinder walls and crevice regions. This results in relatively higher HC and CO emissions, promoting incomplete combustion. A new combustion strategy, ‘reverse RCCI’ (R-RCCI) combustion, has gained significant attention. In the R-RCCI combustion mode, reverse reactivity stratification is used to achieve superior performance and emissions to the conventional RCCI combustion mode. In R-RCCI, a small amount of HRF is premixed in the intake manifold to ignite the LRF, which is directly injected into the cylinder during the compression stroke (Fig. 8).

In R-RCCI combustion mode, a relatively smaller quantity of HRF than conventional RCCI is premixed, resulting in lesser fuel trapped in the squish and crevice regions. The higher oxidization tendency of the trapped premixed HRF is another important aspect of the R-RCCI combustion mode, enhancing the combustion efficiency. A lower combustion rate of R-RCCI compared to PPC makes it more suitable due to more reactivity stratification. Many researchers have carried out detailed experimental and simulation studies and reported that the R-RCCI combustion mode results in lower heat transfer, leading to higher thermal efficiency. Previous studies showed that R-RCCI balanced the crucial parameters required for soot oxidation, namely the degree of premixed combustion and combustion temperature. This led to very low soot emissions at all engine loads. Huiquanet al. [87] conducted an experimental investigation in RCCI, PPC, and R-RCCI combustion modes. They reported that the R-RCCI combustion mode resulted in lower NOx and soot emissions than PPC (Fig. 9). The authors focused on the RCCI combustion mode exhaust losses caused by its lower combustion rate, which resulted in lower ITE. They suggested that the R-RCCI configuration effectively resolved this issue by using PODE injection in the port and gasoline injection directly in the combustion chamber (Fig. 10).

Liu et al. [131] used PRFs (n-heptane and iso-octane) as test fuels in the R-RCCI combustion mode to investigate the effects of fuel concentration and reactivity stratifications. They reported a significant increase

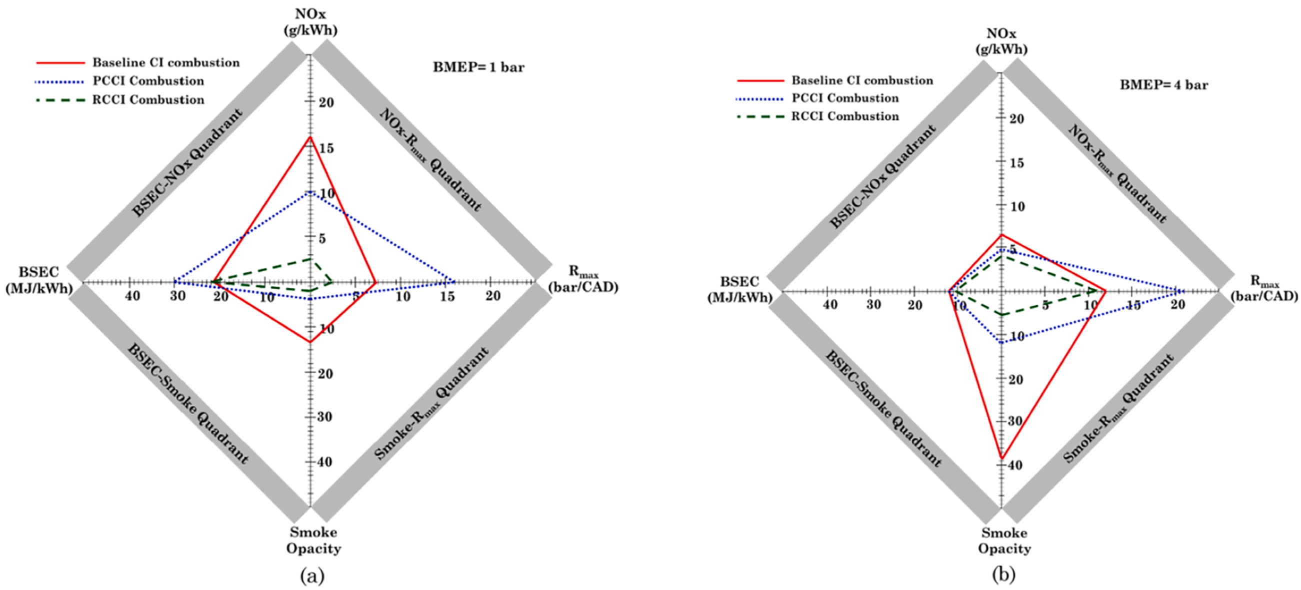


Fig. 7. Comparison of baseline CI, PCCI, and RCCI combustion modes at (a) low engine load (1 bar BMEP) and (b) high engine load (4 bar BMEP) (Reprinted from [173] with permission of Elsevier).

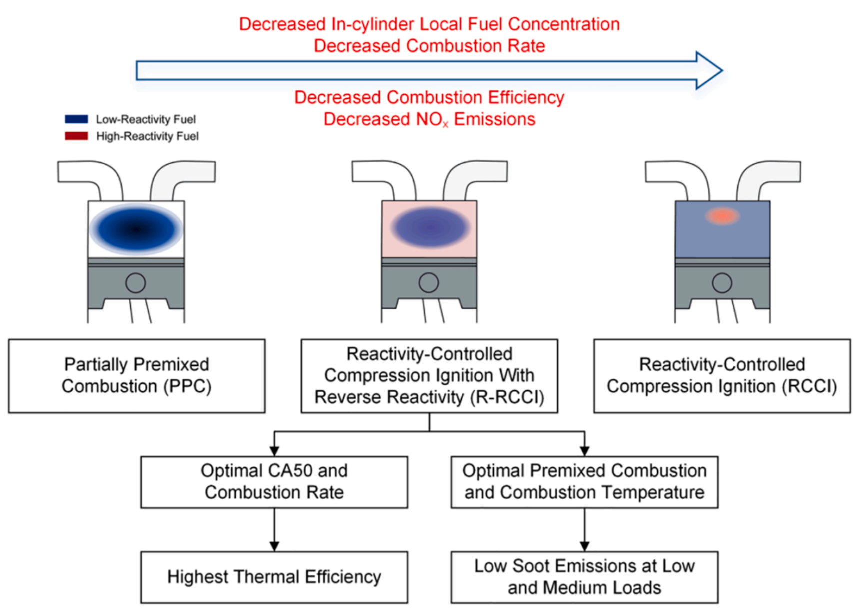


Fig. 8. PPC, RCCI, and R-RCCI combustion modes at low-to-medium loads (Reprinted from [87] with permission of Elsevier).

in the engine's thermal and combustion efficiency. Ji et al. [180] and Lu et al. [146] also used the same test fuels (n-heptane as the premixed HRF and iso-octane as DI fuel) in their investigations. They focused on the fuel injection timing of the LRF and reported a weak effect of the DI timing of the LRF on the R-RCCI combustion mode. The SoI sweep exhibited the best ignition characteristics at the DI timing of − 25°CA aTDC, leading to the highest thermal efficiency.

Yao et al. [149] and Chen et al. [181] investigated the RCCI combustion mode using the reverse fuel reactivity combination. They used a modified single-cylinder diesel engine with premixed dimethyl ether (DME) as HRF and injected methanol as LRF directly into the combustion chamber. They utilized the basic concept of ATAC, in which the heat released by premixed DME was enhanced by the in-cylinder thermal atmosphere. They reported that this method extends the load limit due to higher reactivity stratification achieved by direct methanol injection. They found that early direct injection of fuel results in homogeneous combustion, similar to HCCI; however, the late direct injection can be used for the three-stage combustion. Lu et al. [146,182,183] investigated a similar type of combustion using different LRFs, namely iso-octane, n-butanol, and ethanol, along with premixed n-heptane as HRF. Based on the heat release pattern, they categorized the combustion of two fuels having different reactivities in three ways:(i) a two-stage HCCI-like heat release process dominated by the thermal atmosphere, (ii) a three-stage heat release process dominated by the active atmosphere combustion, and (iii) a heat release process lying in between the above two categories dominated by both the active and thermal atmospheres [146]. Cui et al. [184] performed an optical investigation of the R-RCCI combustion mode to understand the combustion better. They reported that the premixed ratio (r_p) was the most critical parameter in controlling the combustion phasing.

Lu et al. [182] also used premixed n-heptane and directly injected methanol in the R-RCCI combustion mode and reported significantly

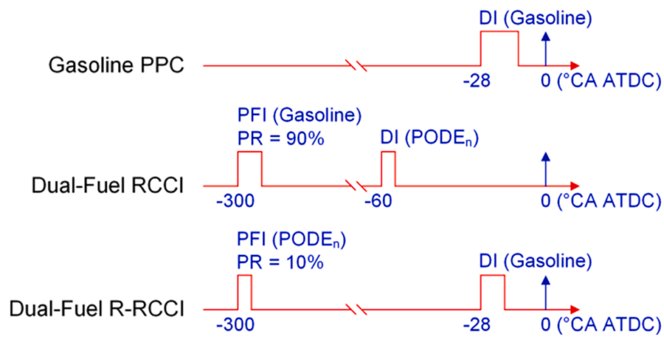


Fig. 9. Fuel induction strategies corresponding to the electrical signal to the injectors for PPC, RCCI, and R-RCCI combustion modes (DI: direct injection; PFI: port fuel injection; PR: premixed ratio) (Reprinted from [87] with permission of Elsevier).

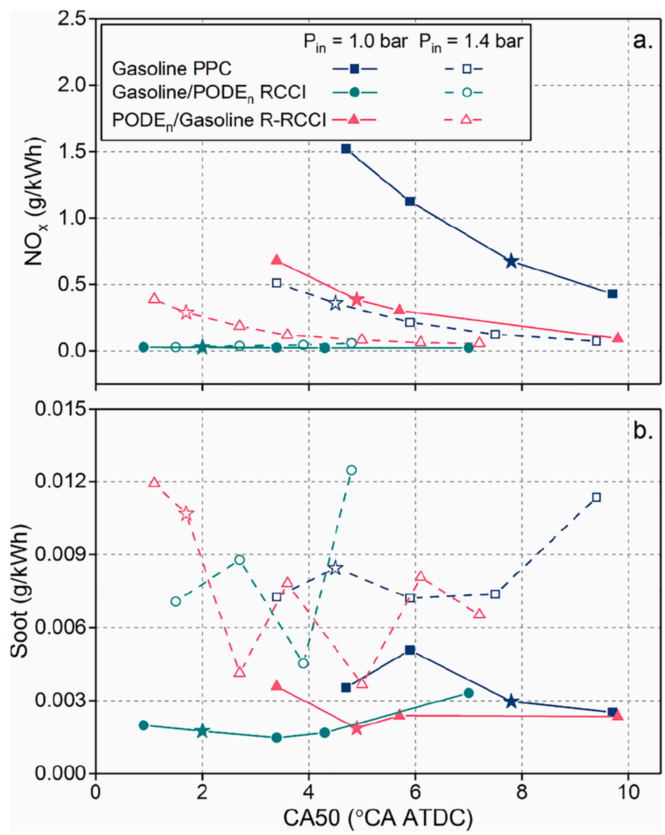


Fig. 10. Comparison of NOx and soot emissions among PPC, RCCI, and R-RCCI combustion modes (Reprinted from [87] with permission of Elsevier).

lower NOx and soot emissions. They suggested that combining optimum combustion characteristics and fuel properties promoted huge reductions in soot and NOx emissions.

Tang et al. [185] also investigated the R-RCCI combustion mode using a combination of n-heptane and iso-octane as test fuels. They analyzed the effect of injection timings of iso-octane on the combustion characteristics of dual-fuel combustion. They divided the combustion into three regimes, namely LTHR, homogeneous combustion heat release (HCHR), and premixed combustion heat release (PCHR), where HCHR and PCHR were the two sub-parts of the HTHR (Fig. 11). They observed that the SoI timings of iso-octane had little effect on the LTHR; however, retarding the SoI timing of iso-octane resulted in relatively dominant HTHR. This was mainly attributed to HCHR and PCHR as the two HTHR regimes, in which HCHR became more dominant at retarded SoI timings. Recently, a new version of the RCCI combustion mode, namely "Intelligent Charge Compression Ignition" (ICCI), has been introduced in which both LRF and HRF can be directly injected to achieve superior combustion control [186,187]. Zhao et al. [187] carried out a detailed investigation using ICCI and RCCI combustion modes and compared their performance and emission characteristics to conventional combustion using blends of HRF and LRF. They reported that the conventional combustion using blends resulted in superior combustion and performance characteristics at low engine loads; however, RCCI and ICCI combustion modes exhibited superior combustion control at higher engine loads, leading to higher engine efficiency. Among both LTC modes, the RCCI combustion mode was found superior, especially up to medium engine loads; however, the ICCI combustion mode showed greater potential to be adopted at higher engine loads. This was mainly due to precise control of injection parameters of LRF and HRF.

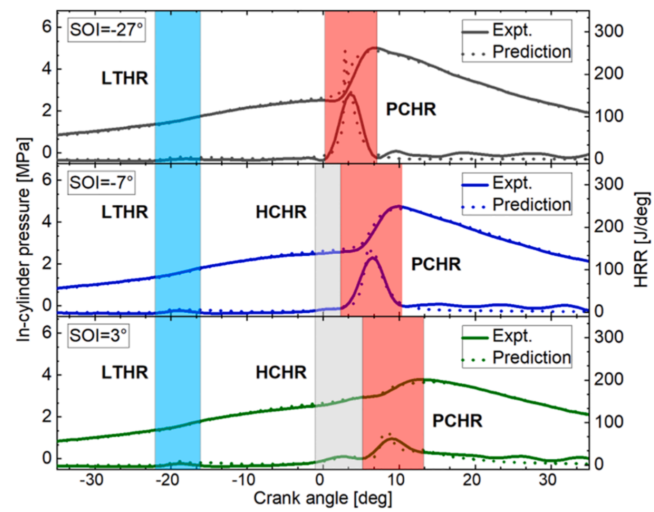


Fig. 11. Experimental and predicted in-cylinder pressure and HRR profiles at three SoI timings (Reprinted from [185] with permission of Elsevier).

This section summarizes the evolution of the RCCI combustion mode as an LTC strategy. A LRF is introduced into the engine through the intake port, forming a premixed charge, which is ignited using a directly injected HRF. The RCCI combustion mode performance is analysed using numerous test fuels, including alternative fuels. Some of these alternative fuels aided in reducing soot and NOx emissions. RCCI combustion mode was compared to other combustion concepts, including the ATAC. The injection mechanisms/strategies were different among these combustion modes, e.g., the ATAC inducted the LRF through direct injection and the HRF through the intake port. This section also underscores some of the issues faced by RCCI, including relatively higher HC and CO emissions. In the RCCI combustion mode evolution, reverse RCCI was explored as a new strategy consisting of reverse reactivity stratification, i.e., injecting a smaller quantity of HRF into the intake manifold to ignite the LRF, directly injected into the cylinder. A smaller quantity of HRF injected in the R-RCCI combustion mode leads to lesser fuel entrapment in the crevices. Finally, some fundamental studies on the effect of boundary conditions of R-RCCI are listed to explore how the combustion is affected.

2.1. Fuel Kinetics of LTC

Most ICEs are operated by gasoline- and diesel-like fuels, which contain primarily straight-chain alkanes, branched alkanes and aromatics. Previous studies using PRFs exhibited that the combustion kinetics of these fuels and their auto-ignition characteristics are significantly affected by the fuel composition and molecular size and structure of fuel constituents. The gasoline and diesel combustion in

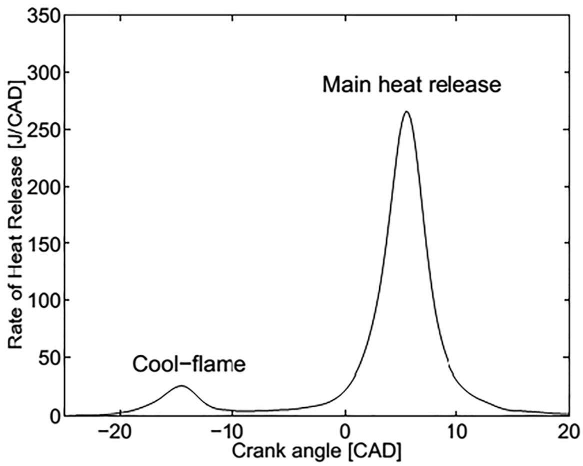


Fig. 12. Typical HRR for two-stage combustion in the LTC mode (Adapted from [192]).

conventional combustion mode is mainly affected by transport-controlled high-temperature premixed and non-premixed flames [188]. However, in LTC modes, autoignition and cool flames play an important role in combustion characteristics due to a higher degree of fuel-air premixing. Ju et al. [188] reviewed several studies for a detailed understanding of cool flames in LTC mode and reported that cool flames are affected by parameters such as EGR, fuel structure, etc. EGR reduces the flame temperature in LTC mode since it has a dominant role in suppressing hot flames. This results in dominant cool flames in LTC mode because a transition between cool and hot flames during combustion affects the engine performance and emission characteristics [189]. Agarwal et al. [190] reviewed the fuel combustion chemistry and reported that PRFs (n-heptane and iso-octane) behave differently during auto-ignition in LTC engines. N-heptane is reactive straight-chain paraffin with a low octane number and lower auto-ignition resistance. Compared to n-heptane, iso-octane is a less reactive branched-chain paraffin with a higher octane number [191]. They explained the combustion characteristics of LTC and reported two-stage combustion in which the first stage is related to the LTHR. The second or the main combustion is associated with the HTHR (Fig. 12). The first stage of ignition is regarded as a 'cool flame' with a negative temperature coefficient (NTC) [192].

The analysis of chemical reactions during the combustion of hydrocarbon fuels in ICEs shows three main reaction routes, depending on temperature. Below 750 K, reactions are dominated by chain-propagating steps, including oxygen molecules and the generation of partially oxidized species [193]. Between 800 and 950 K, the chain propagating steps of the 'O' molecule yielding conjugate alkenes and HO_2 radicals [193]. Above 1000 K, the main fuel radical reactions include thermal decomposition of C-C bond breakage, formation of alkenes and smaller radicals [194]. In previous studies, two different regions of chain reactions were defined as: (i) a low-temperature kinetics region or ignition process below 950 K and (ii) the high-temperature kinetics region, where the bulk chemical energy is released [192]. However, the introduction of alternative fuels, such as alcohols, ethers, biodiesel, etc., has shown slightly different LTC kinetics due to different molecular sizes, molecular structures, bond energies and functional groups [195–199].

The cool flames in combustion have been investigated extensively; however, after introducing LTC modes, they have become more relevant due to a relative dominance of low-temperature reactions (LTR) and high-temperature reactions (HTR) in the LTC mode. Pease [200,201] reported that in the LTC mode, radical formation is sensitively affected by the temperature, decreasing with increasing temperature due to faster dissociation of chain-branching intermediates. In this context, fuel reactively plays an important role because, at low temperatures, the rate of chain branching for straight-chain paraffins (like n-heptane) is much more intense than for branched-chain paraffins (like iso-octane). This is due to the structure of n-heptane radicals, which leads to higher alkyl-peroxy radical (RO_2) isomerization from keto-hydro-peroxide decomposition [191]. A research study by Barusch et al. [202] reported that the cool flame chemistry of LTC was significantly affected by RO_2 and hydroperoxyl alkyl radicals (QOOH). On the other hand, many less reactive methyl groups and tertiary and quaternary C atoms in its structure promoted lower reactivity of branched-chain paraffins (like iso-octane). For compact and highly branched fuels, ignition was inhibited by a large fraction of strongly bonded H atoms . The OH radical pool controlled the iso-octane combustion, mainly derived from the n-heptane low-temperature pathway.

Many previous studies reviewed the fundamentals of cool flames and their applications in different combustion models [203–205]. Most studies reported that the temperature and radical production rate affected the oxidation of fuels having low-temperature reactivity [203, 204]. The chain branching processes also affect the ignition of hydrocarbons, which occurs in two stages. In the first stage, partial oxidation of fuel occurs, resulting in intermediate species like C_2H_4, CH_2O, and CH_3CHO. The first stage of the ignition process is controlled by the low-temperature peroxy chemistry [195,204,206], and it does not contribute to the temperature rise. The first stage ignition process further progresses by the oxidation of intermediate species, which raises the temperature to a critical limit, where the second stage ignition starts. In this stage, complete oxidation of fuel results in the formation of CO_2 and H_2O along with a significant temperature rise. This is a major reason why the second ignition stage is also known as high-temperature ignition (HTI). In conventional combustion modes, significantly higher initial temperatures result in only HTI; however, LTI and HTI both contribute to LTC modes. Their relative dominance depends on temperature and pressure conditions. In most studies related to cool flames, their behaviour depends on the initial temperature and the chain branching chemistry. This is divided into three categories: low-temperature, intermediate-temperature, and high-temperature routes. At lower temperatures (up to 750 K), OH, O, or HO_2 radicals start the fuel oxidation by H-abstraction from the fuel molecule (RH) and generate fuel radicals (R). These fuel radicals react with O_2 and form RO_2, which undergoes internal isomerization, resulting in the formation of QOOH. Decomposition of QOOH leads to many OH radicals, depending on the low-temperature pathway. Due to the formation of these large numbers of radicals, chain branching becomes rapid, which adds more energy and OH radicals to the system [207]. The low-temperature reactions are exothermic, which further increase OH radical formation due to increasing temperature [191].

$$R \rightarrow RO_2 \rightarrow QOOH \rightarrow O_2QOOH \rightarrow OQ'O + 2OH$$

With increasing temperature, the chain-propagating pathways suppress the chain-branching pathways, in which peroxy hydroperoxyl alkyl radicals (O_2QOOH) are replaced by cyclic ether via the following reaction.

$$R \rightarrow RO_2 \rightarrow QOOH \rightarrow OH + \text{cyclicether}$$

However, the NTC affects the above reaction due to reverse reactions and significantly reduces the formation of radicals. Pilling et al. [208] reported another feature of the low-temperature reactions. The chain-branching reactions do not act as actual chain-branching because of the formation of either one or no radials at low temperatures. They suggested that this behaviour of chain branching at low temperatures is common to all test fuels. However, few oxygenated fuels, including diethyl ether, dibutyl ether, etc., exhibit differences in the two NTC regions. The main reason for the first NTC is the dominance between the second O_2 addition and QOOH decomposition. The competition between the first O_2 addition and direct beta scission for the fuel radicals is the main reason for the second NTC [209]. The tendency of the dual NTC effect is mainly controlled by C-O bond energy. Rodriguez et al. [210] explored the low-temperature behaviour of DME and reported that the NTC effect for DME starts at ~550 K; however, it becomes significantly noticeable in the temperature range of 600-750 K. In this temperature region, fuel oxidation becomes slow due to dominant contributions of O_2QOOH, QOOH, and RO_2 decomposition reactions. At intermediate temperature (900-1100 K), high-temperature chemical reactions dominate the low-temperature reactions, resulting in chain branching, where fuel oxidation is thought to be governed by a branching process involving HO_2 [192].

$$RH + HO_2 \rightarrow H_2O_2 + R$$

$$H_2O_2 + M \rightarrow OH + OH + M$$

Here, M is a non-reactive third-body. For initiating these reactions, HO_2 is mainly formed by the following reactions:

$$RO_2 \rightarrow \text{alkene} + HO_2$$

$$QOOH \rightarrow \text{cyclicether} + OH \rightarrow \text{alkene} + HO_2$$

The reaction progresses with the formation of OH radicals by decomposition of RO_2 and QOOH and reaction between aldehyde radicals and O_2. Most of the time, the formation of these lower aldehydes is a possible reason for the faint blue luminescence of cool flames [200,201]. At intermediate temperatures, the chain-branching reactions produce multiple OH radicals; however, these reactions are highly sensitive to heat loss and oxygen concentration.

$$CH_2O \rightarrow HCO \rightarrow HO_2 \rightarrow H_2O_2 \rightarrow 2OH$$

In some cases (at relatively higher temperatures), this intermediate chain-branching reaction is surpassed by a multi-stage warm flame, where both cool and hot flames can exist [211]. Mostly the LTC lies in the H_2O_2 concentration history, which is produced at ~1000 K by low and intermediate temperature reactions. Li et al. [212] suggested a prominent pathway of H_2O_2 formation, as given below:

$$HO_2 + \cdot HO_2 \rightarrow H_2O_2 + O_2$$

H_2O_2 concentration increases steadily, with H_2O_2 decomposition much slower than its production. H_2O_2 decomposes rapidly at temperatures around 1000 K, yielding many OH radicals. An increase in OH radicals consumes any remaining fuel, resulting in the ignition. The important requirement for cool flames to induce in-cylinder combustion is the energy required so that low-temperature reactions can increase the chamber temperature to the high-temperature region. When the reactions become slower during the NTC period, high cylinder temperature can be increased sufficiently by external factors such as external heating or increasing CR, which forces the engine operation towards a high-temperature regime by increasing the compression work during the compression stroke. As shown below, a different chain-branching reaction occurs at extremely high temperatures (above 1100 K).

$$H + O_2 \rightarrow OH + O$$

This reaction dominates the overall reaction, leading to the main ignition, identical to all test fuels. This is the unique chemical kinetics of all test fuels at high temperatures where chain branching reactions become fuel independent. Since molecular oxygen participates in this reaction, lean fuel mixtures are more reactive in this high-temperature regime. In contrast, rich fuel-air mixtures are oxidized quickly at low temperatures due to chain branching, which depends on the radical species formed directly from the parent fuel molecules [213]. The consumption and production pathways of O_2 and OH (radical pool) reveal that the major O_2 consumption reactions related to the LTC chemistry of hydrocarbons directly contribute to the heat release. Heat release reactions are related to O_2, implying the significance of the fuel mixing process for the LTC. These radical-branching reactions are very important in the high-temperature regime because the products contain more radicals than the reactants. Similar to previous experiments, Mancaruso and Vaglieco [214] performed UV–Visible imaging and spectroscopic measurements, and they detected the chemical species involved in the LTC. They injected bio-ethanol in the intake manifold and mineral diesel directly into the cylinder, similar to RCCI combustion mode. They reported that the SoC of the homogeneous fuel-air mixture in LTC was mainly controlled by OH radicals generated during the intermediate temperature chain branching pathways of LTC. The concentration of OH radicals was directly controlled by the temperature and type of reactions, e.g. at low, intermediate or high temperature. Liu et al. [215] explored the fuel kinetics of RCCI combustion mode and reported that CH_2O and HCO (formyl radical) are the two most effective species that play an important role in conducting the reactions in the HTHR stage. Even in diesel engines, it has been revealed by the stabilization mechanism and overall ignition process that the LTHR pattern is similar to the LTC engines, which progresses towards a dominant HTHR. Krisman et al. [216] experimentally demonstrated this behaviour, where thermal and molecular diffusion gradually transitioned from the LTHR to flame stabilized autoignition (HTHR).

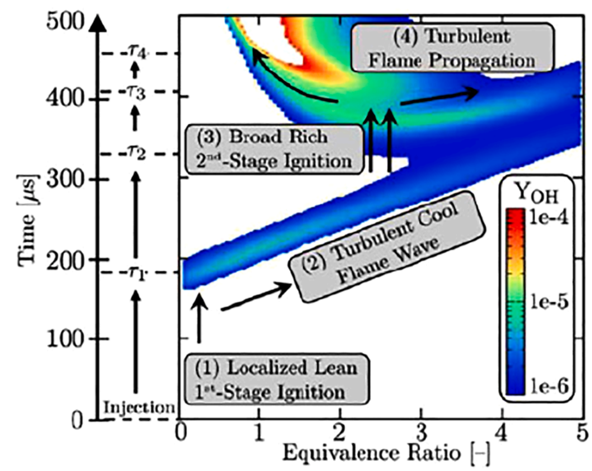


Fig. 13. Evolution of characteristic events and associated time scales for ignition in high-pressure spray flames (Reprinted from [217] with permission of Elsevier).

Dahms et al. [217] related these combustion regimes and suggested that LTC reactions and cool flames cannot be neglected in diesel engines because the ignition process is associated with these reactions (Fig. 13). This sub-section showed the general fuel combustion kinetics followed in LTC and the importance of cool flames and NTC in different LTC modes. The next sub-sections discuss details of these reaction mechanisms, where the effects of different features of LTC kinetics on combustion, performance, and emissions are elaborated.

2.2. Emissions from RCCI Combustion Mode

Researchers have explored several techniques to resolve the issue of higher emissions of CO and HC from RCCI combustion mode engine. This section summarizes the studies wherein the performance and emission characteristics of the RCCI combustion mode engine are the primary focus. Wai et al. [218] investigated a diesel-methanol dual-fuel (DMDF) combustion strategy to control the engine-out emissions. They conducted experiments using a pilot injection to achieve stable combustion at a high methanol substitution ratio (MSR). They examined the effects of pilot injection parameters, such as the start of pilot injection timing, pilot injection quantity, etc. They reported that these parameters could effectively control the DMDF combustion, resulting in superior fuel economy, especially at higher MSR. Zhang et al. [219] performed a similar experiment on a mineral diesel-methanol dual-fuel engine and

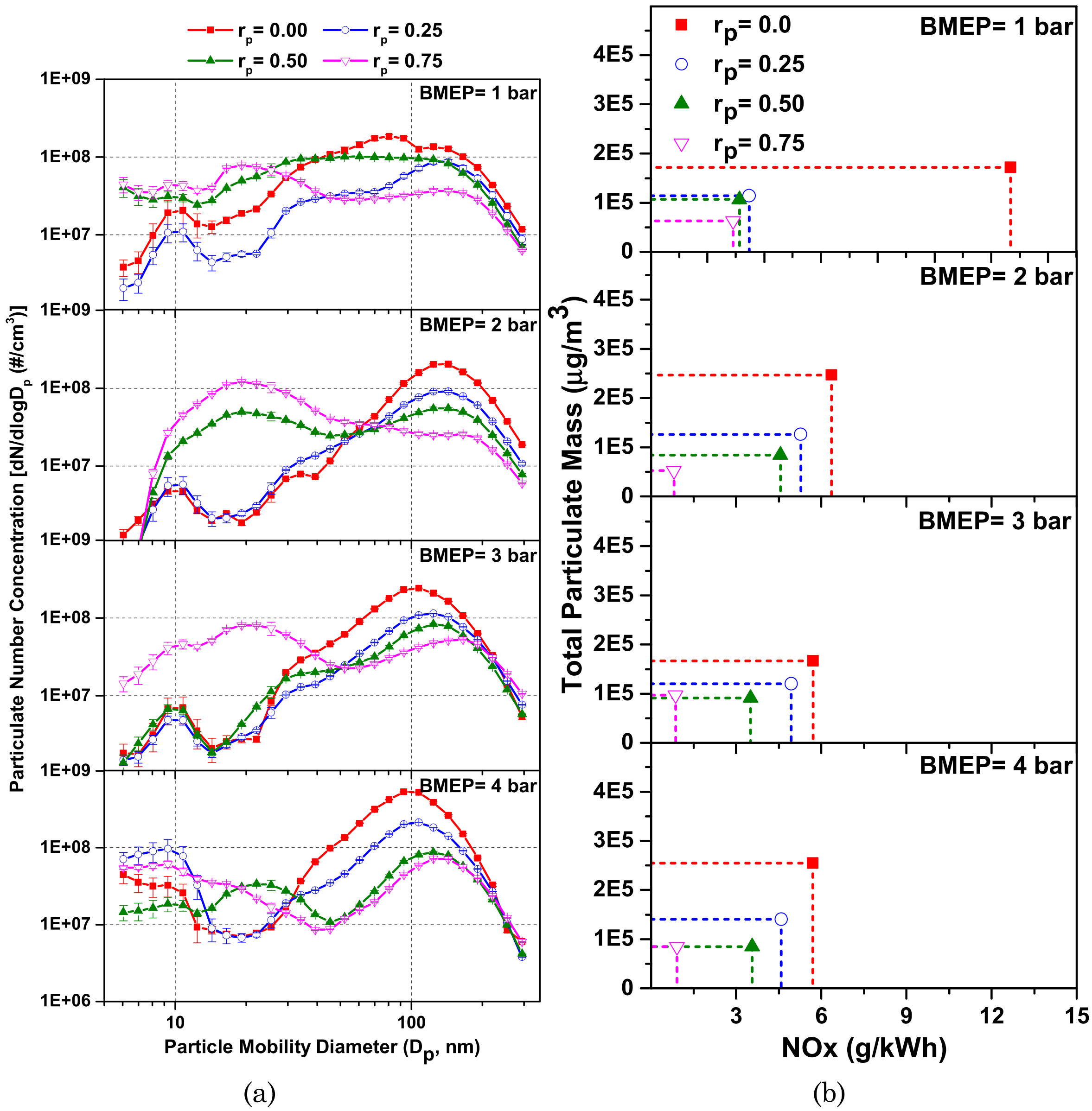


Fig. 14. (a) Number-size distributions of particulates, (b) Correlation between the total particulate mass and NOx emitted in RCCI combustion mode at different engine loads and r_p (Reprinted from [171] with permission of SAGE).

reported that methanol fumigation exhibited a significant reduction of NOx and PM emissions; however, the charge cooling effect of methanol enhanced incomplete combustion, leading to relatively higher HC, CO and formaldehyde emissions.

Although RCCI combustion mode emits significantly lower PM than other conventional combustion modes, however, PM emitted by RCCI combustion mode cannot be neglected. Therefore, PM emissions from RCCI combustion mode engines were routinely investigated. Splitter et al. [220] reported that RCCI combustion mode engines emitted approximately two orders of magnitude lower PM mass emissions than conventional CI engines. Singh et al. [171] performed RCCI combustion mode investigations intending to explore the permissible limits of the r_p of methanol at different engine loads. They reported that the increasing r_p of methanol first improved the engine performance; however, it exhibited relatively inferior performance at higher r_p of methanol (>75%). To investigate the RCCI combustion mode and emission characteristics further at extreme limits (combining engine load and r_p), they measured the particle number (PN) concentrations emitted by the RCCI combustion mode engine vis-à-vis baseline CI combustion mode engine (Fig. 14). They reported that the number of particles emitted by RCCI combustion mode was significantly lower than baseline CI combustion mode. RCCI combustion mode emitted more nucleation mode particles (NMP) size range [171]. The relative dominance of particles in various size ranges, namely nanoparticles (NP), NMP, and accumulation mode particles (AMP) in the two combustion modes, showed an interesting trend. Fig. 14 shows that higher engine loads exhibited higher AMP concentrations in both combustion modes, and NMP concentration decreased when increasing engine load. Variations in the number of NP followed a mixed trend at different engine loads, and maximum NP concentration was found at the maximum engine load (BMEP of 4 bar). Higher peak in-cylinder temperature at maximum engine load might be a possible reason for the emission of a higher number of NPs. A significant fraction of NPs was generated due to the pyrolysis of lubricating oil.

In another study [221], the results showed a very low soot emission (<0.01 g/kW-hr) from a light-duty engine operating in RCCI combustion mode. Prikhodko et al. [222] compared the effectiveness of the DOC on different advanced combustion techniques. They concluded that DOCs

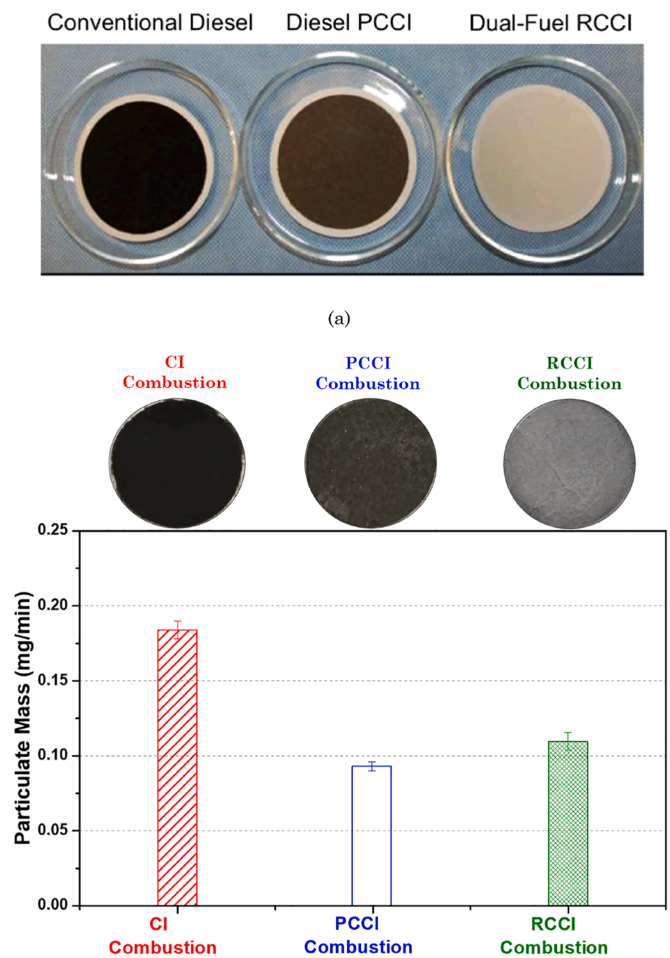


Fig. 15. (a) Comparison of PM samples collected on the filter papers from conventional CI, diesel PCCI, and dual-fuel RCCI combustion modes (Reprinted from [222] with permission of SAE), and (b) Gravimetric analysis of PM mass collected on the filter papers from CI, PCCI, and RCCI combustion modes (Reprinted from [16] with permission of Elsevier).

could reduce the emission of small-sized particles effectively. They collected the PM samples (on filter paper) from the CI, PCCI, and RCCI combustion mode engines operated at identical speed and load conditions. The filter papers exhibited the PM sampling with the lightest colour from the RCCI combustion mode engine; however, the PM mass collected on the filter paper was higher than that collected on the PCCI combustion mode engine filter (Fig. 15).

In a similar study by Agarwal et al. [16], a comparison of the physical appearance of the particulate-laden filter papers and PM mass emitted by PCCI, RCCI, and CI combustion modes was done. They reported that the RCCI combustion mode resulted in a slightly higher PM mass on the filter paper, in which a significant fraction of the PM included volatile substances. The physical appearance of these filters also exhibited the presence of black carbon (BC) to be the maximum in the CI combustion mode, followed by the PCCI combustion mode.

Jiang et al. [223] performed RCCI combustion experiments to investigate the particulate emissions by gravimetric measurements and reported slightly lower total particle mass (TPM) than the baseline CI combustion. They reported that PM mass emitted by the RCCI combustion mode engine was significantly lower than the baseline CI combustion mode engine. At higher engine loads, TPM emitted by the RCCI combustion mode engine was reduced substantially (by ~57%) due to a higher r_p of LRF because of improved fuel-air mixing. Due to emissions of condensable HC species, PM emitted by the RCCI combustion mode engine contained higher soluble organic fraction (SOF) than baseline CI combustion mode. The relatively lower in-cylinder temperature was the main reason for the higher SOF of the PM emitted by the RCCI combustion mode engine, which remained the same even at higher engine loads; however, SOF in the PM emitted by the baseline CI combustion mode reduced significantly at higher engine loads.

Morphological analysis of the particulates using transmission electron microscopy (TEM) was done by Storey et al. [224], which revealed the presence of condensed hydrocarbon droplets in the PM samples from the RCCI combustion mode engine. This analysis also indicated that the PM from the RCCI combustion mode engine had lower carbonaceous material than the soot emitted by the baseline CI combustion mode engine. The absence of graphitic structure in the PM emitted by the RCCI combustion mode engine indicated a different PM formation mechanism than the baseline CI combustion mode. For investigating the effect of fuel pairs on the particulate characteristics, chemical characterization (organic carbon (OC)/total carbon (TC) ratio) of the PM emitted from CI and RCCI combustion mode engines fueled with different fuel pairs was performed. It was concluded that the fuel chemistry did not affect the RCCI combustion mode because all fuel pairs showed similar OC/TC ratios. Smoke meter measurements showed a consistent and significant reduction in BC content of the PM emitted by the RCCI combustion mode engine. However, this does not accurately represent the TPM since the smoke meter only assesses the BC. Kolodziej et al. [225] investigated the RCCI combustion mode when both fuels (LRF and HRF) were directly injected into the combustion chamber. Their results showed that PM emitted by the RCCI combustion mode had a bimodal particle size distribution (PSD), which was sensitive to the SoI timings of gasoline and gasoline-diesel ratio. They also concluded that advancing the SoI timing of gasoline resulted in lower NMP concentration; however, AMP concentration increased. Increasing the gasoline-diesel ratio decreased the TPN emissions. Storey et al. [224] also investigated the PSD sensitivity to the fuels used in the RCCI combustion mode by employing three fuels pairs: diesel-gasoline, diesel-E85 (85% ethanol + 15% gasoline), and B20 (20% biodiesel + 80% diesel)-gasoline. There was no significant difference in the PSD among these fuel pairs, indicating that PSDs from the RCCI combustion mode were largely insensitive to the test fuel properties. Agarwal et al. [16] compared the particulate characteristics of different LTC modes vis-à-vis baseline CI combustion mode. They used mineral diesel to achieve PCCI and CI combustion modes and a combination of methanol and mineral diesel as HRF and LRF to achieve the RCCI combustion mode. They reported significantly lower PN emissions from both LTC modes than baseline CI combustion mode; however, the RCCI combustion mode emitted relatively higher PN than the other two combustion modes (Fig. 16 a). They suggested that fuel-rich zones in the combustion chamber and a dominant diffusion combustion phase in the baseline CI combustion mode were the two prime factors for higher PN emission from the RCCI combustion mode than baseline CI combustion mode. However, homogeneous fuel-air mixing in RCCI combustion mode resulted in fewer fuel-rich zones, leading to dominant premixed combustion.

Statistical analysis of particulate results exhibited that the number concentration of particles was higher in CI combustion mode; however, the RCCI combustion mode emitted relatively higher particulate mass (Fig. 16 b). The particulate number-mass distribution showed that the PCCI combustion mode emitted lower PN and mass than RCCI. Relatively superior in-cylinder conditions in the PCCI combustion mode than in RCCI combustion mode might be a possible reason for this behaviour, promoting homogeneous fuel-air mixing, leading to lesser soot nuclei formation. Their study concluded that all LTC modes could reduce both PM and NOx emissions; however, the RCCI combustion mode was more dominant in NOx and PM reduction due to the combined effects of LTC and the use of oxygenated fuels. Particulate bound trace metal analysis was another important aspect related to particulate toxicity. Agarwal et al. [16] compared the particulate bound trace metal concentrations emitted in the PCCI, RCCI, and baseline CI combustion modes and reported significantly lower trace metals from the PCCI and RCCI

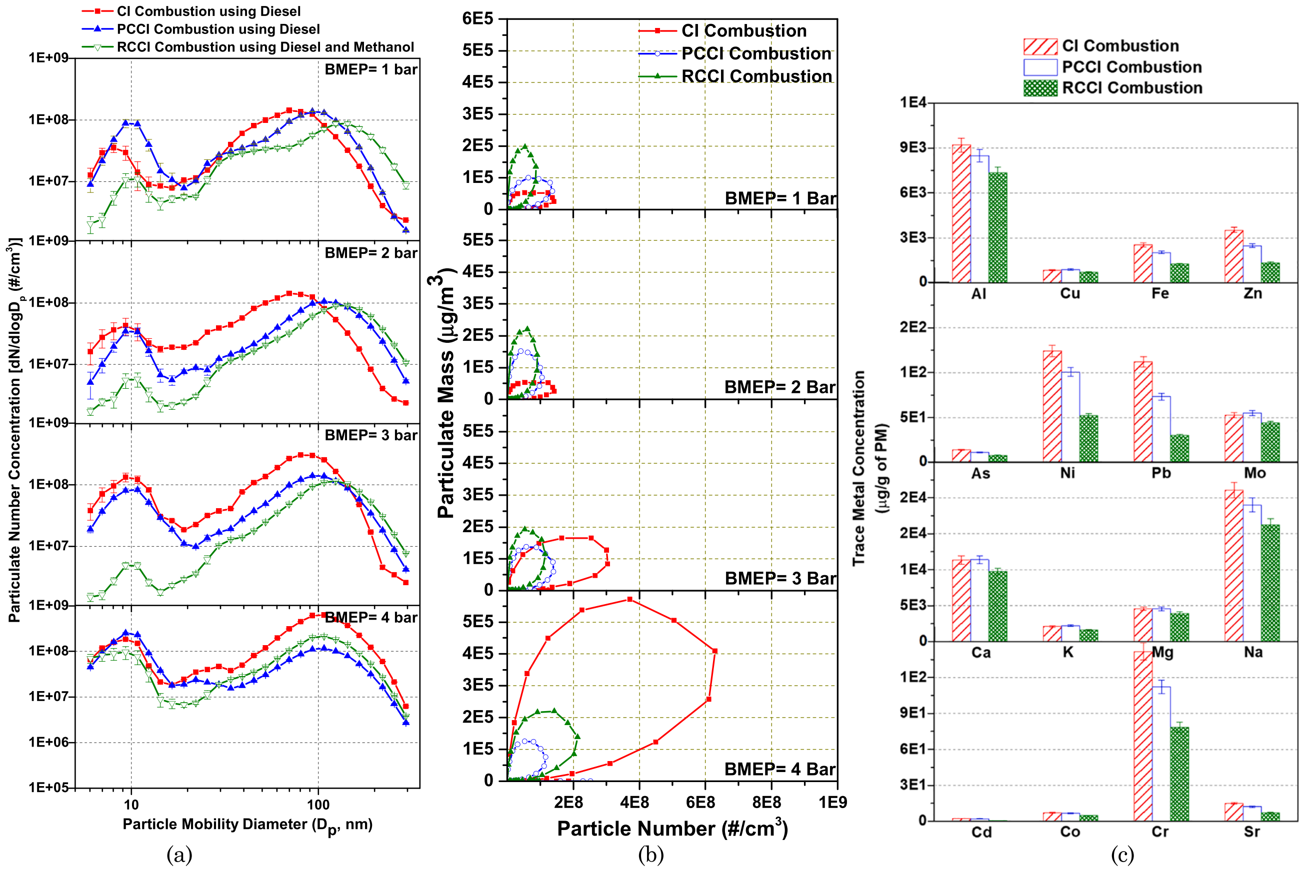


Fig. 16. (a) Number-size distribution of particulates and (b) Qualitative correlation between the number-size and mass-size distribution of particles, and (c) Particulate bound trace metals emitted in CI, PCCI, and RCCI combustion modes at varying engine loads (Reprinted from [16] with permission of Elsevier).

combustion modes than in the baseline CI combustion mode (Fig. 16 c). They suggested that relatively lower peak in-cylinder temperature was a major factor responsible for lesser trace metals such as Ni, Ar, Pb, Cr, etc., in both these LTC modes.

This section covers the findings of the experimental investigations of the RCCI combustion mode, particularly emissions. Attention was given to higher HC and CO emissions than other combustion modes, such as baseline CI combustion mode, even when using fuel combinations such as diesel and methanol (formaldehyde was also detected). Additionally, detailed studies on the emission characterization, particularly PM emission, were summarized, concluding that the RCCI combustion mode emitted lesser BC but might emit a higher mass of non-BC particles. It also emerged that the chemical characterization of particulate matter in terms of OC/TC ratio was not affected by the fuel chemistry. Finally, the RCCI combustion mode exhibited fewer trace metals than the baseline CI combustion mode.

2.3. *Alternative Fuels for RCCI Combustion Mode*

Alternative fuel utilization potential is an important feature of the RCCI combustion mode. RCCI combustion mode can utilize many alternative fuels, ranging from high-cetane to low-cetane. The RCCI combustion mode with alternative fuels exhibits higher efficiency and cleaner combustion than CDC. This feature makes the RCCI combustion mode suitable in the current scenario when the transport sector is battling energy security and emissions issues. In the last few years, a wide range of alternative fuels has been investigated in RCCI combustion mode. These are divided into two main categories: alternative fuel usage as (i) HRFs, and (ii) LRFs.

2.3.1. *Alternative Fuels as HRFs in RCCI Combustion Mode*

Most RCCI combustion mode investigations were carried out using mineral diesel as HRF. To explore the effects of HRF properties on RCCI combustion mode, Ryskamp et al. [226] carried out RCCI combustion mode experiments using nine different compositions of mineral diesel, having different CN, aromatic content, and distillation temperatures. They reported that the CN of HRF was dominant compared to other fuel properties. They concluded that a lower CN of HRF resulted in higher NOx and PRR; however, higher cetane fuels showed greater potential for higher engine load operation. Therefore, in the RCCI combustion mode, higher cetane alternative fuels are preferred [227–233] to compensate for the combustion delay that is typical to this mode due to elevated proportions of LRF. Many alternative fuels, such as biodiesel, DME, etc., have been explored as alternatives to mineral diesel [227–241]. Recent studies have focused on biodiesel types, achieving a superior reactivity stratification due to the higher CN of biodiesel than baseline mineral diesel. The CN of biodiesel depends on the saturation and molecular structure of the constituent hydrocarbons from the feedstock oils or fats, affecting the oxygen its molecules can contain. In biodiesel, the oxygenation of the molecules is generally higher. However, there is high variability in the determination of this metric even with two fuels coming from the same feedstock [242]. Wang et al. [232] used PODE$_n$ as HRF and gasoline as LRF to achieve RCCI combustion mode at higher engine loads. They reported that biodiesel's higher CN created more reactivity stratification than baseline mineral diesel. This helped achieve superior combustion control, especially at higher engine loads. The relatively higher PM reduction potential of PODE$_n$ than mineral diesel is another reason why PODE$_n$ was extensively explored as HRF in the RCCI combustion mode. Tong et al. [154] compared the RCCI combustion mode and performance characteristics of a heavy-duty engine fueled with gasoline-PODE$_n$ and gasoline-diesel. They concluded that gasoline-$PODE_n$ RCCI combustion mode exhibited more stable combustion characteristics. Near-zero PM emissions from the gasoline-$PODE_n$-fueled RCCI combustion mode were another important finding of their study; however, they did not discuss these characteristics elaborately. To further explain the reasons for superior emission characteristics of $PODE_n$ as the directly injected HRF in the RCCI combustion mode, Wang et al. [233] performed detailed simulations of the in-cylinder fuel distribution of gasoline-$PODE_3$ fueled RCCI combustion mode. They reported relatively superior management of the in-cylinder equivalence ratio and reactivity distribution of $PODE_3$ for improving the RCCI combustion mode engine performance more flexibly than baseline mineral diesel. García et al. [234] explored the potential of $PODE_n$ in the RCCI combustion mode, wherein they used diesel-$PODE_n$ blends as HRF and gasoline as the premixed LRF in different r_p. They concluded that the higher blends of $PODE_n$ with mineral diesel complied with the EURO VI emission regulations even at higher engine loads (up to 80% load). In a few other studies, this technique was also explored, where lower blends of $PODE_n$ with mineral diesel resulted in superior combustion efficiency for the same NOx levels. This increase in combustion efficiency led to higher brake thermal efficiency (BTE) [237].

Song et al. [235] evaluated the suitability of $PODE_n$ in the RCCI combustion mode by comparing emissions from natural gas-diesel and natural gas-$PODE_n$-fueled RCCI combustion mode engines. They demonstrated that a combination of natural gas-$PODE_n$ resulted in significantly lower HC, CO, NOx, and PM emissions and superior engine performance. The higher difference in reactivities of natural gas and $PODE_n$ might be a possible explanation for this trend. Hararia et al. [228] explored a new strategy in which they used a mixture of CNG and compressed biogas (CBG) as the LRF, and different blends of thevetia-peruviana methyl ester (TPME) were used as the HRF. The main objective of this research was to enhance the load limit of RCCI combustion mode. They reported that this fuel combination resulted in relatively higher BTE and NOx emissions than the reference RCCI combustion mode fueled with CNG-mineral diesel; however, CO and HC emissions were marginally lower. Aydin [243] used CNG as LRF and biodiesel as HRF in the RCCI combustion mode. They used a higher fraction of safflower biodiesel (up to 95%) to achieve RCCI combustion and compared the results with conventional CI combustion mode. They reported that the RCCI combustion mode with a higher fraction of biodiesel as HRF resulted in a shorter combustion duration than the conventional CI combustion mode.

Due to the significant effect of fuel properties of HRF on the RCCI combustion mode, researchers also used DME as HRF in the RCCI combustion mode. Kakoee et al. [229] investigated the RCCI combustion mode using natural gas as LRF and DME as HRF. They added hydrogen to the natural gas to improve global reactivity and reactivity stratification control. They reported that DME decomposition during the RCCI combustion mode was an important phenomenon that affected the fuel mixture's CN and the HRR. A higher CN of DME becomes less effective in such conditions due to these changes in the mixture properties before the combustion, leading to lower combustion quality, higher emissions, and lower power output than baseline diesel [229]. Jin et al. [230] performed detailed experimental investigations to understand the ignition dynamics of the DME-methane mixture, which were explored as a potential fuel pair in the RCCI combustion mode. They explained the ignition characteristics of the DME-methane mixture and reported that the initial ignition events were similar in all cases of RCCI combustion mode. However, for the DME-methane case under turbulent conditions, the formation of high-temperature autoignition kernels was relatively advanced compared to a homogeneous mixture. They indicated the existence of typical tetrabranchial flames, such as cool flames, fuel-rich premixed flames, diffusion flames, and fuel-lean premixed flames, where the fuel-lean premixed flame branches finally trigger the premixed methane-air flames. Krisman et al. [244] also performed similar investigations using DME as HRF and reported that DME exhibited a two-stage ignition in a turbulent mixing layer. They reported that cooler regions in the combustion chamber affect the timings and location of ignition in the second stage auto-ignition.

Park et al. [231] explored the RCCI combustion mode characteristics using DME and ethanol as HRF and LRF, respectively. Few other researchers used DME in R-RCCI combustion mode to achieve superior combustion control than the RCCI combustion mode. They reported that the DME-ethanol fueled RCCI combustion mode resulted in lower emissions than biodiesel-ethanol and diesel-ethanol dual-fuel combustion. They reported a significant reduction in $ISNO_x$ in the DME-ethanol fuel pair without deterioration of IS_{soot}. Yao et al. [149] and Chen et al. [181] assessed DME as HRF in a modified single-cylinder diesel engine, in which methanol was injected directly into the combustion chamber as LRF. Yao et al. [149] reported that premixed DME enhanced the in-cylinder thermal atmosphere, which resulted in superior combustion to conventional RCCI combustion mode. Chen et al. [181] investigated methanol-DME fueled RCCI combustion mode by varying fuel injection parameters such as fuel injection timings of methanol, FIP, etc. They reported that varying SoI timings of methanol exhibited a weak effect on the LTHR in the DME as HRF; however, it significantly affected the HTHR. They suggested that the overall combustion duration of methanol-DME fueled RCCI combustion mode could be reduced by increasing the FIP of methanol [181].

RCCI combustion mode can be enhanced by using HRF, other than mineral diesel. HRFs such as $PODE_n$ have fuel properties that aid in making the combustion more controllable, thus allowing to cater to higher load operating points in the LTC mode. With biodiesel, higher combustion efficiency and BTE can be achieved. Additionally, these alternative fuels enhance the potential of reducing soot emissions while keeping the same level of NOx emissions as the baseline mineral diesel.

2.3.2. Alternative Fuels as LRFs in RCCI Combustion Mode

In the past five decades, many alternative fuels have been extensively explored for IC engines. Among those alternative fuels, alcohols have shown significant potential to be utilized as an alternative to mineral diesel and gasoline. However, alcohols have been promoted more as an alternative to gasoline for practical applications. In many countries, gasoline has been replaced with alcohol blended gasoline, in which 20% (v/v) alcohols are blended with gasoline. Unlike gasoline engines, the use of alcohol in diesel engines is challenging. In many studies, alcohols have been explored as a direct replacement of mineral diesel in CDC; however, igniting alcohols in diesel engines is very difficult due to their lower CN. In the literature, many techniques have been proposed for alcohol utilization, such as fuel blending [238–241,244,245], port fumigation [246–252], and dual-fuel emulsions [218]. The blending of alcohols with mineral diesel has attracted researchers due to its simplicity. In most blending strategies, the suitability of alcohol has already been justified [253–255]. However, most studies concluded that blending strategies can be used up to only a certain fraction of alcohol in diesel engines since high alcohol proportions are closely related to rust and corrosion issues in IC engines. In addition, due to the lower CN of alcohols, their ignition is challenging, requiring higher proportions of HRF. This reduces the advantage of RCCI combustion mode as higher proportions of HRF pushes the combustion peak closer to the CDC. Moon et al. [256] conducted experiments using mineral diesel-ethanol blends. They reported that lower ethanol blends with mineral diesel (< 30% v/v of ethanol) exhibited stable combustion without significant variations in fuel spray atomization characteristics. However, higher blends of alcohol with diesel pose several serious issues such as phase separation (especially for methanol), inferior fuel spray atomization, poor combustion, engine performance, and serious material compatibility issues with the fuel injection system. Therefore, alcohol blending with mineral diesel has not been implemented commercially. Methanol utilization was then explored using diesel-methanol compound combustion (DMCC) [247,248]. Yao et al. [247] performed DMCC experiments and reported a significant reduction in the NOx and soot emissions simultaneously. Haribabu et al. [249] investigated a similar combustion mode

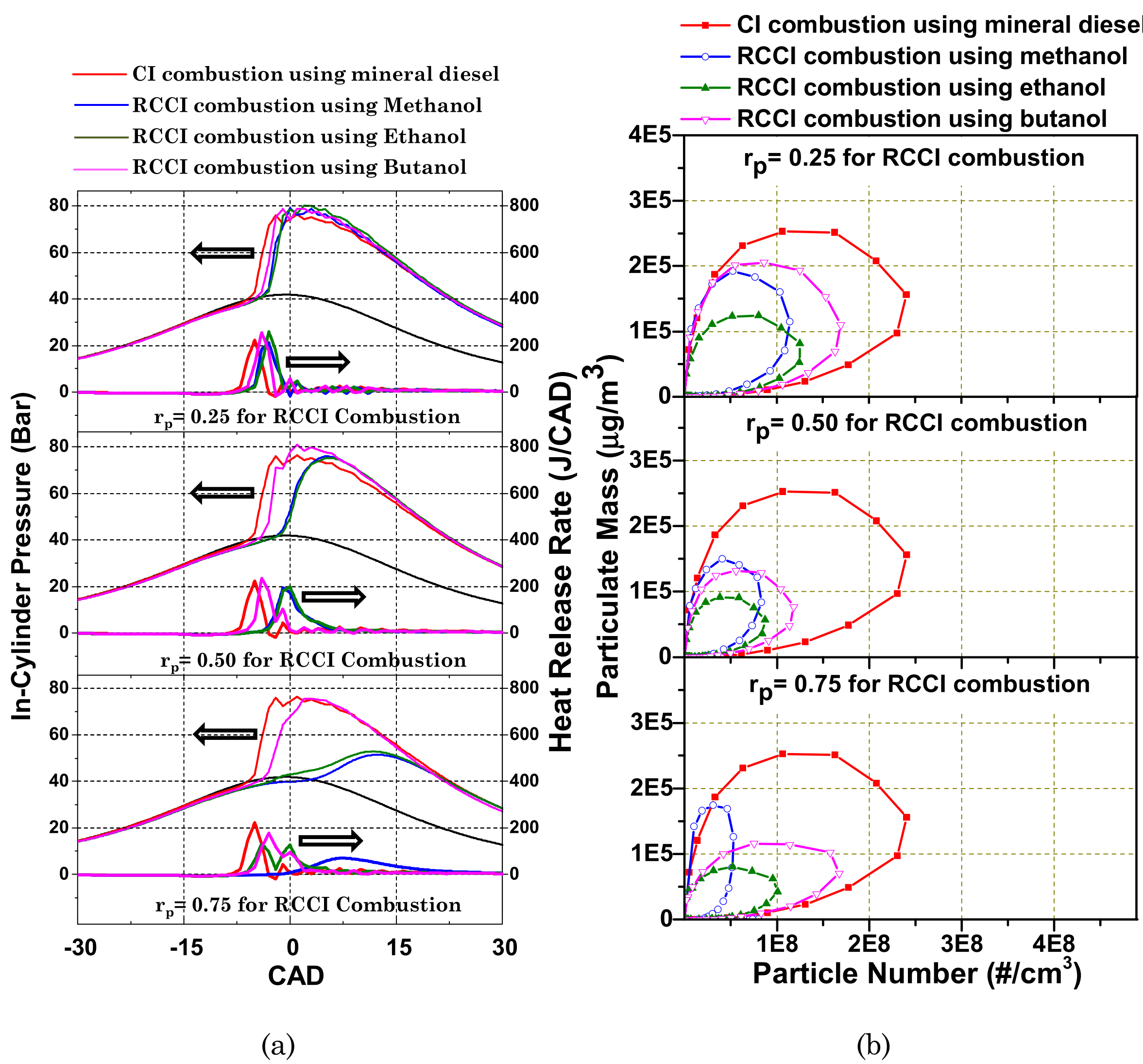


Fig. 17. (a) In-cylinder pressure and HRR variations w.r.t. CAD, (b) Number-size and mass-size correlations of particles at different r_p of Methanol, Ethanol, and Butanol at constant engine speed (1500 rpm) and load (3 bar BMEP) (Reprinted from [154] with permission of SAE).

using direct injection of methyl ester and port injection of methanol via carburetion. They also reported relatively lower NOx and smoke emissions and improved fuel economy.

The RCCI combustion mode showed the tremendous potential of using alternative fuels in the last few years. In RCCI combustion mode, CNG [257–260], syngas [261], alcohols including methanol [122,250, 262,263], ethanol [264,265], and butanol [177,266,267] have been injected into the port as LRF. These LRFs helped in extending the load range of engine operation due to their higher resistance to auto-ignition (less knocking), higher reactivity gradient (optimum combustion phasing), and charge-cooling effect (due to their higher latent heat of vaporization). In most preliminary investigations of the RCCI combustion mode, methanol was used as the LRF due to its lower reactivity (lower cetane), making it suitable for achieving a reactivity stratification with mineral diesel. The gasification of coal can be done to produce methanol; however, in the last few years, production of methanol from black liquor gasification, biomass gasification, and the reaction of hydrogen and CO_2 (directly from the atmosphere or from the coal-fired power plants, industrial flue gases, etc.) is attracting significant global attention [268]. The presence of relatively higher oxygen than the other alcohols also makes it more suitable, promoting soot oxidation, leading to lower particulate emissions [238].

Excessive LRF leads to incomplete combustion, resulting in higher HC and CO emissions, especially at low loads. Sayin et al. [239–241] investigated the RCCI combustion mode using a mineral diesel-methanol fuel pair and reported a significant reduction in NOx and PM than the CDC mode. They also emphasized PM reduction due to the use of methanol. They reported that the absence of a C-C bond was the main reason for lower PM emissions from mineral diesel-methanol-fueled RCCI combustion mode. A few studies reported that the lack of a C-C bond also reduces the formation of polycyclic aromatic hydrocarbons (PAHs), which act as soot precursors.

Like methanol, ethanol is also a potential candidate for LRF in the RCCI combustion mode. Hanson et al. [269] performed preliminary RCCI combustion mode investigations on a light-duty, multi-cylinder diesel engine using a blend of ethanol and gasoline (E20) as the LRF. They reported that the lower reactivity of ethanol was suitable for achieving a sufficient reactivity gradient for the RCCI combustion mode. They found lower PRR in E20-mineral diesel-fueled RCCI combustion mode than in gasoline-mineral diesel-fueled RCCI combustion mode. Apart from these findings, they also reported that E20 also helped in increasing the peak power output because ethanol is less prone to auto-ignition. Qian et al. [270] conducted RCCI combustion mode experiments using n-heptane as the HRF and three different LRFs, namely ethanol, n-butanol, and n-amyl alcohol. They reported that all three LRFs showed similar combustion characteristics at the lower r_p;

however, ethanol-n-heptane fueled RCCI combustion mode resulted in relatively retarded combustion events at higher r_p. This was also visible in the emissions characteristics, where n-heptane-ethanol-fueled RCCI combustion mode exhibited a relatively greater reduction in NOx and soot emissions than the other two fuel pairs. Agarwal et al. [150] performed RCCI combustion mode using n-butanol to explore its potential as the LRF. They performed RCCI combustion mode experiments at different r_p of n-butanol at varying engine loads. They reported that n-butanol exhibited relatively superior combustion characteristics even at higher r_p; however, they did not observe any significant variation in the performance parameters. They suggested that the relatively higher CN of n-butanol than other alcohols, such as methanol and ethanol, was the main reason for superior engine performance. To further explore the effect of fuel properties on the RCCI combustion mode, Agarwal et al. [154] performed a detailed experimental study on the RCCI combustion mode using methanol, ethanol, and n-butanol as the LRF.

They performed experiments at constant engine load and the r_p of the LRF. They found quite a similarity between the RCCI combustion mode fueled with methanol and ethanol as the LRF; however, butanol fueled RCCI combustion mode exhibited closer similarity with the CI combustion mode (Fig. 17 a). This was also remarkably reflected in the emission characteristics, where methanol and ethanol-fueled RCCI combustion modes showed greater potential for PM reduction than the butanol (Fig. 17 b). They concluded that all three alcohols could be used as the LRF in the RCCI combustion mode to reduce NOx and PM emissions simultaneously.

The use of biogas in IC engines is not a new area; however, the performance of biogas-fueled engines using conventional combustion modes has not been found satisfactory. Therefore, biogas has always been used in dual-fuel combustion mode engines to achieve acceptable performance and emissions. Prajapati et al. [271] investigated the potential of biogas utilization as the LRF in the RCCI combustion mode. They reported that the RCCI combustion mode engine fueled with mineral diesel-biogas fuel pair exhibited lower NOx and CO_2 emissions than the CDC mode. In another study by Bora et al. [272], a biogas-rice bran biodiesel fuel pair was used in the RCCI combustion mode, delivering superior engine performance due to superior reactivity stratification between the biogas and the biodiesel. In a comparative investigation by Verma et al. [273,274], the effectiveness of biogas, CNG, and hydrogen as the LRF was assessed. In the exergy analysis, it was observed that mineral diesel-hydrogen-fueled RCCI engines exhibited better engine performance than the ones using biogas. This might be due to higher free radicals such as O, H, and OH, enhancing the net reaction rate and a higher net HRR [275]. The use of hydrogen in RCCI combustion mode is also beneficial due to its higher laminar burning velocity, leading to a shorter combustion duration. Though, the replacement of mineral diesel with biodiesel showed slightly inferior engine performance. In a similar investigation by Khatri et al. [276], hydrogen addition in biogas in dual-fuel mode improved the engine's efficiency. Ebrahimi et al. [277] reported that the addition of hydrogen also helped in the dissociation of methane present in biogas, which resulted in a relatively shorter combustion duration of the RCCI combustion engine fueled with biogas-mineral diesel fuel pair. They also reported the fuel chemistry related to hydrogen addition in RCCI combustion mode. They observed that the formation of OH radicals was delayed due to an increasing fraction of hydrogen, leading to a relatively longer ignition delay.

For this reason, in many studies, reformed biogas was also investigated [278–280]. Mahmoodi et al. [281] investigated reformed biogas-mineral diesel-fueled RCCI combustion mode using 3D modelling. They used higher r_p of reformed biogas and observed significantly reduced mean combustion temperature in the RCCI combustion mode than in the baseline CI combustion mode. This might be due to the relatively weaker effect of CO_2 present in biogas on the LTC strategies than in the CDC mode [282]. CO_2 present in the biogas also affected the combustion characteristics due to the dominant role of thermal, reactive, and transport properties of CO_2 on the ignition characteristics. Nieman et al. [257] used the KIVA3V CFD software and CHEMKIN, the chemical kinetics analysis tools, to investigate the diesel-CNG dual-fuel RCCI combustion mode. They observed superior performance and lower exhaust emissions in the RCCI combustion mode at low engine loads without EGR. However, EGR controlled the RCCI combustion mode at higher engine loads. Aydin [243] also used CNG as LRF in RCCI combustion mode and reported that CNG could reduce high gas temperature, useful for reducing NOx. They reported that biodiesel/CNG fueled RCCI combustion mode reduced NOx emissions in proportion to the amount of CNG. Martin et al. [283] used a blend of propane and DME as LRF to achieve RCCI combustion. They reported that this fuel combination could result in an intermediate combustion mode, namely premixed dual-fuel combustion (PDFC), which lies in between CI and RCCI combustion modes. This study found relatively higher NOx from PDFC compared to RCCI combustion mode. However, relatively lower PRR makes this combustion technique more suitable at higher engine loads. Relatively higher BTE of PDFC was another important finding of their study.

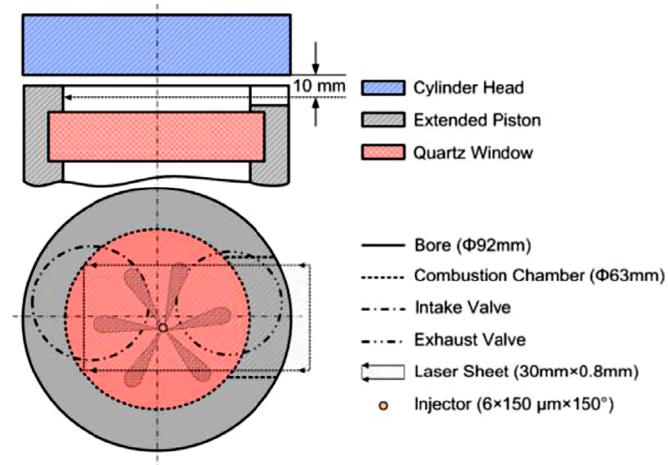


Fig. 18. Scheme of the combustion chamber and field of view of the optical diagnostics (Reprinted from [286] with permission of Elsevier).

This section reports the findings of several studies performed on the RCCI combustion mode using LRF other than gasoline. It could be concluded that though alcohols are potentially superior LRF than fossil fuels, their higher concentrations could hamper the combustion stability and cause complications such as fuel blend separation and other engine operational issues. Since alcohols have a lower CN, their ignition in RCCI combustion mode presents some challenges. Other alternative LRFs extend the operational range of RCCI combustion mode if they have higher CN, higher resistance to auto-ignition, higher reactivity gradient, and charge cooling effect, preventing some of the better-known issues of RCCI combustion mode, such as probable knocking. Using some of these alternate fuels as the LRF reduced soot emissions, especially when the C-C bonds are absent in the test fuel. Hydrogen was also tested as the LRF and helped enhance the RCCI combustion mode engine performance, as reported by some studies, due to higher net HRR caused by free radicals such as O, H, and OH.

2.4. Fundamental Investigations of RCCI Combustion Mode

Literature shows that LTC strategies have been extensively assessed in various all-metal engines, including heavy-duty engines. Most experimental studies in metal engines have focused on improving the combustion, performance, and emission characteristics of LTC strategies. However, RCCI strategy requires a finer understanding that can only be explored using fundamental experimental investigations. RCCI combustion mode is a kinetically-controlled combustion, significantly

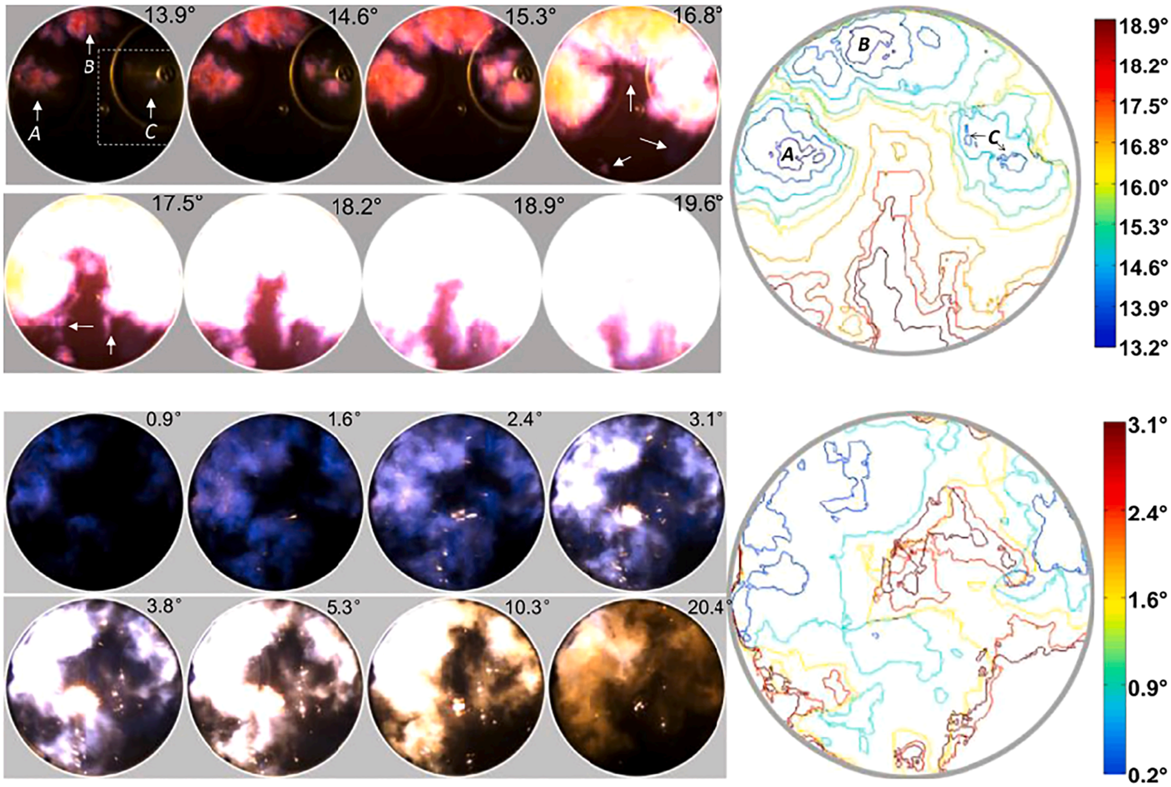


Fig. 19. Time-resolved single-shot true-colour images and flame boundary evolution of PPC and RCCI combustion modes firing cycles. [Top] PPC [Bottom] RCCI (Reprinted from [286] with permission of Elsevier).

affected by the in-cylinder processes, including charge stratification, reactivity gradients, ignition, and flame evolution, which need to be explored in detail using advanced optical diagnostic techniques [135, 284,285]. This section deals with the methodology and results of fundamental optical investigations using different test fuels, control techniques, engine loads, etc.

Liu et al. [286] performed a comparative investigation of PPC and RCCI combustion modes using different optical diagnostics techniques such as fuel-tracer planar laser-induced fluorescence (PLIF) imaging, high-speed natural flame luminosity (NFL) imaging, and Formaldehyde and -OH PLIF imaging. The researchers used a quartz window-fitted piston engine for the optical studies (Fig. 18).

Their objective was to focus on the flame development process and assess the distribution of intermediate combustion products and free radicals. The authors used iso-octane as the LRF and n-heptane as the HRF to achieve the RCCI combustion mode; however, PRF70 was used as the test fuel in the PPC mode. They kept most of the operating conditions identical in both combustion modes. The RCCI combustion mode exhibited a shorter ignition delay than the PPC mode. The RCCI combustion case has a higher minimum equivalence ratio (0.54) and fuel-rich regions with higher reactivity where the fuel is more easily auto-ignited (thus reducing the ignition delay). Optical investigations of NFL at 70% r_p (70% LRF) revealed that RCCI combustion mode does not exhibit flame front propagation (Fig. 19). The locally high reactivity regions in the RCCI combustion form flames in the periphery of the cylinder during the first crank-angle degrees (CAD). However, in the central region of the cylinder, the reactivity of the mixture is lower. After 3.1° CAD, the flames reach the central region. In the case of PPC, auto-ignition kernels appear at 13.9° CAD in local fuel-rich zones, which later show flame fronts. The difference between the two modes in terms of flame propagation lies in the fact that the fuel reactivity stratification of the RCCI combustion mode induces more auto-ignition in the earlier combustion stages, which translates into a faster combustion rate than PPC mode, whose auto-ignition depends on the local equivalence ratio (the difference between fuel-rich and fuel-lean-regions). The researchers later increased the r_p (only 5% of the input energy coming from the HRF), finding that the rate of combustion of RCCI combustion mode decreased. RCCI combustion with higher r_p gets closer to the combustion in SI engines. The increase in r_p also results in a flame front propagation in RCCI combustion mode; however, the laminar flame speed was lower than the PPC mode. The authors concluded that the degree of fuel stratification played a crucial role in auto-ignition and flame front propagation in the RCCI combustion mode and that different fuel injection strategies can control the degree of stratification.

In RCCI combustion mode, different optical diagnostics techniques are limited by the PRR and soot emissions. In most optical investigations, experiments are limited to a certain part-load condition, depending on the strength of optical components. The optical diagnostics of RCCI combustion mode under higher engine loads have higher practical significance since it provides vital information about the limiting criteria. Kokjohn et al. [135,287] performed optical experiments only up to 4.2 bar gross IMEP due to the limited optical strength of the optical components of the engine using high-speed imaging and PLIF techniques to investigate the RCCI combustion mode. They concluded that local fuel reactivity controlled the ignition sites. They found that ignition sites were more dominant in high reactivity zones; therefore, they appeared downstream of the injected fuel jets. In such conditions, HRR can be controlled by fuel stratification. They further extended this work and included chemical kinetics modelling [135] to explore the role of fuel reactivity stratification, equivalence ratio, and temperature on the RCCI combustion mode control. They concluded that the fuel reactivity was the main parameter, which controlled the ignition delay, and fuel concentration and temperature stratification exhibited a relatively weaker effect on the combustion. The combustion images captured by the high-speed camera also showed that the RCCI combustion mode was significantly affected by the degree of fuel stratification. Upon comparing high-speed imaging, they reported that PLIF imaging was more informative and provided valuable

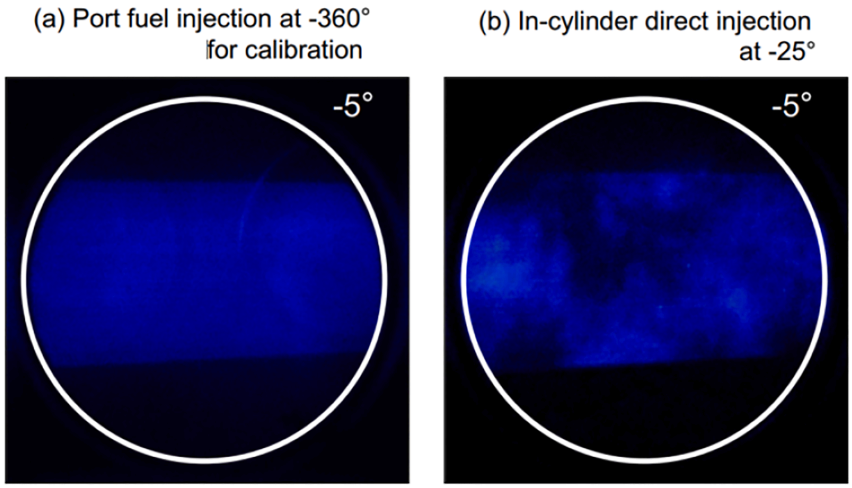


Fig. 20. PLIF images acquired at -5°CA aTDC with (a) PI and (b) DI. The results are averaged from 10 single-shot PLIF images (Reprinted from [288] with permission of Elsevier).

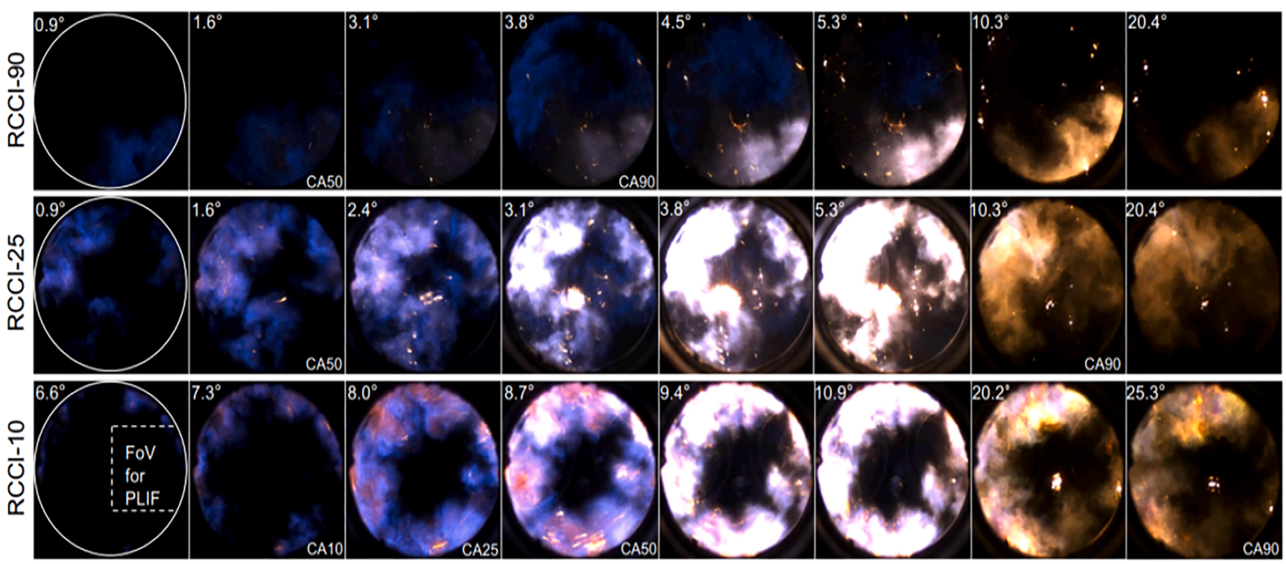


Fig. 21. True-color natural luminosity (NL) image sequences for the cases of RCCI-90 (top), RCCI-25 (middle) and RCCI-10 (bottom), respectively (Reprinted from [288] with permission of Elsevier).

information about forming formaldehyde and OH radicals in the two-phase (LTHR and HTHR) ignition of RCCI combustion mode. However, the high-speed imaging provided only a two-dimensional natural flame luminosity in a three-dimensional reaction zone. This technique also had another limitation. It could not distinguish between the LTHR and HTHR phases, an important aspect of RCCI combustion mode. However, this issue of high-speed imaging was reported only in a few studies related to RCCI combustion mode at different degrees of fuel stratification.

Tang et al. [288] also explored the effects of fuel stratification on the RCCI combustion mode using several optical diagnostics techniques such as PLIF under non-reactive conditions, Time-resolved natural combustion luminosity imaging, and single-shot OH PLIF imaging in a light-duty optical engine. PLIF technique was used to quantify the fuel-air equivalence ratio and PRF number. Experiments were performed at different injection timing (Fig. 20).

The authors reported that higher fuel concentration and reactivity regions were affected by the SoI timing, which moved downstream to the edge of the combustion chamber with retarded SoI timing of n-heptane from -90°CA aTDC (RCCI-90 case) to -10°CA aTDC (RCCI-10 case). They reported that combustion took place in two stages in the case of RCCI-10. An auto-ignition was observed around the combustion chamber in the first stage due to high reactivity regions. In the second stage, auto-ignition occurred in the low reactivity regions in the central part of the combustion chamber. This two-stage heat release process was the main reason for relatively lower PRR in the RCCI combustion mode. Formaldehyde and OH PLIF images exhibited a more stratified fuel distribution in retarded SoI timing of n-heptane. Another important observation of their investigation was a relatively slower formaldehyde consumption rate and formation of OH radicals at retarded SoI timings of n-heptane.

In Fig. 21, a time-resolved, single-shot NL image sequence for each case from a typical engine cycle. The crank angle position for 50% (CA_{50}) and 90% (CA_{90}) cumulative heat-release (CHR) were marked on the lower right side of each corresponding image. The boundary of the combustion chamber is shown with a white circle at the start of each image sequence. The field of view (FoV) used in the OH and formaldehyde PLIF imaging was shown with a white dashed line at a 6.6° crank angle for the RCCI-10 case.

Fig. 22 shows that the combustion duration (from CA_{10} to CA_{90}) in

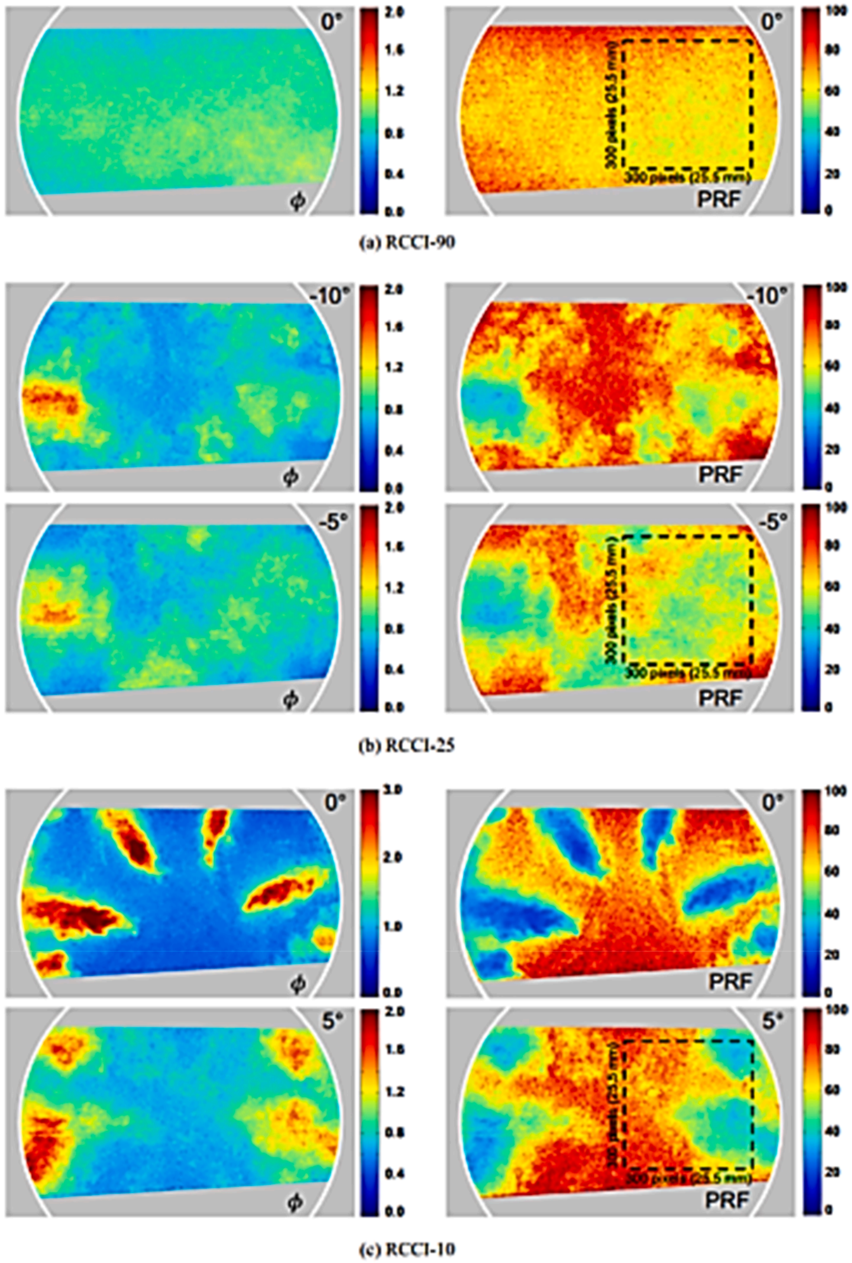


Fig. 22. Fuel-air equivalence ratio (left column) and PRF number (right column) distribution under different n-heptane direct injection timings (Reprinted from [288] with permission of Elsevier).

RCCI-10 was longer than RCCI-25 case. A higher degree of fuel stratification was the main reason for a relatively longer combustion duration in RCCI-10, promoting higher soot production and reduced PRR. OH PLIF images exhibited that the HTHR phase of RCCI combustion mode could be extended up to the central part of the combustion chamber. However, Musculus et al. [289] used a low-load LTC conceptual model and suggested that no HTHR occurs in the central part of the combustion chamber, where the HC forms mainly.

Tang et al. [288] investigated the OH PLIF signal using different SoI timings (Fig. 23). They reported that OH radicals begin to emerge at 1°, 1°, and 7°CA for RCCI-90, RCCI-25, and RCCI-10, respectively, in the regions where formaldehyde vanishes. In RCCI-90, the most field of view was occupied by the OH radicals after 3°CA, showing a uniform OH radical distribution. After that, the OH PLIF signal remained uniform for the next several crank angle degrees; however, the signal intensity became weak after 30°CA. A comparison of OH PLIF signals of different SoI cases showed relatively slower development in the RCCI-10 case than in the other two cases. At 8°CA, OH radicals first occupied the regions around the combustion chamber, where n-heptane resides, fuel reactivity was high, and there were no signs of OH radicals in the regions of lower reactivity, as shown by white dashed lines marked by 'b' (Fig. 23). However, in the following crank angles (9°, 10°, and 11°), a weak OH PLIF signal appeared gradually in this region. By 11°CA, OH PLIF signal of intense non-uniformity occupied the most field of view. Kokjohn et al. [135] explored the role of equivalence ratio, temperature, and fuel reactivity stratification on the HRR using a combination of optical diagnostics and chemical kinetics modelling. They used iso-octane as the LRF, injected during the intake stroke to provide sufficient time for fuel-air premixing, and then injected n-heptane as the HRF at the end of the compression stroke. Results showed that the ignition first occurred in the squish region and then expanded towards the centre of the combustion chamber. They also used PLIF to get

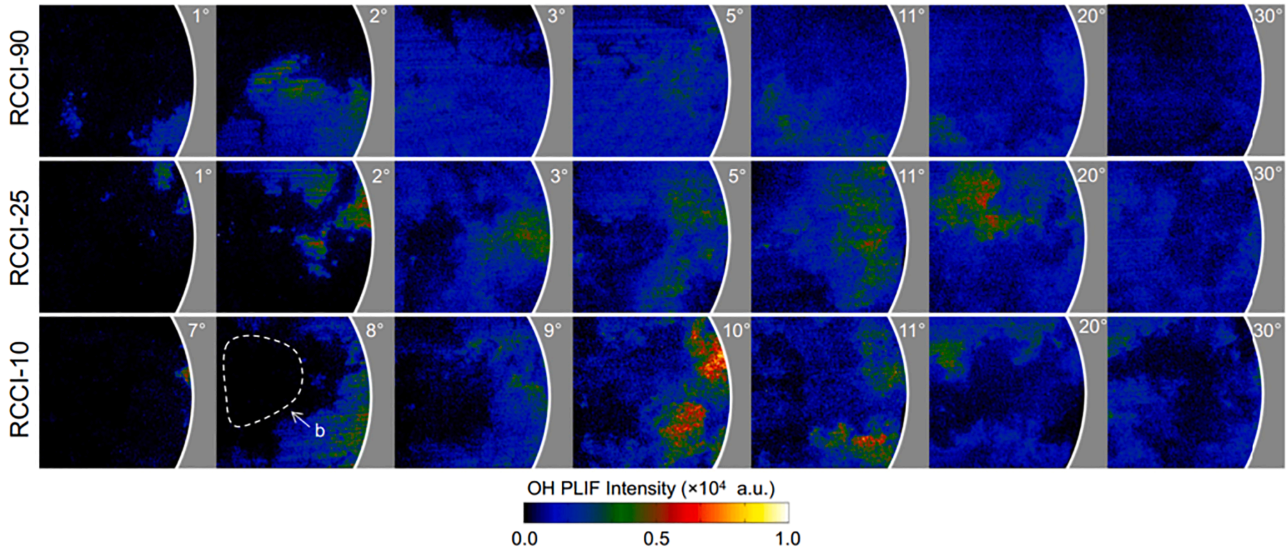


Fig. 23. HTHR process shown by single-shot false-colour images of OH-LIF for different direct injection timings (Top row: RCCI-90, Middle row: RCCI-25, Bottom row: RCCI-10). The white dashed line marked by "b" shows low-reactivity regions not occupied by OH radicals (Reprinted from [288] with permission of Elsevier).

quantitative information about the fuel concentration distributions before the ignition. PLIF showed that the combustion events followed the direction of the reactivity gradient, which was an important characteristic of the RCCI combustion mode (Fig. 24).

The researchers employed a modelling technique to explore the effects of temperature, equivalence ratio, and PRF number stratification on the RCCI combustion mode. They reported that reactivity stratification was the prime factor that controls the combustion chamber's ignition location and growth rate. Compared to reactivity stratification, a relatively weaker effect of equivalence ratio and temperature on the RCCI combustion mode was another important finding of this investigation.

In other studies, the RCCI combustion mode was compared to a diesel-pilot injected natural gas-fueled engine. A diesel injection near the TDC is a strong ignition source for an overall lean natural gas-air mixture. In diesel-pilot initiated combustion, the combustion is presumed to occur via deflagration [290]. Still, in RCCI combustion mode, the role of flame propagation is unclear as only one study has experimentally explored this question. Kokjohn [287] used laser ignition to initiate flame propagation. It was found that flame propagation could exist in the RCCI combustion conditions, as high-speed chemiluminescence showed luminous regions propagated away from the ignition spot. Flames did not propagate in every cycle, but the probability of successful flame kernels increased with increasing equivalence ratio, suggesting that higher equivalence ratio regions were more likely to support the flame propagation. The flame growth rate was similar to the autoignition reaction fronts, indicating that flames could propagate from the auto-ignition sites. Still, the two combustion fronts were not discernable with current optical measurements. KIVA 3D CFD tools allowed newer insight into the combustion process, combustion of intermediate species, and sources of inefficiency and losses. RCCI combustion mode temperatures were much lower, and the peak temperature locations were farther away from the piston and cylinder walls. This reduced the heat losses to the cooling system, which was one of the reasons for the increased thermal efficiency of the RCCI combustion mode [144].

Tang et al. [185] performed RCCI combustion mode investigations to extract detailed information about the RCCI combustion mode and chemical interactions between the LRF and HRF in different engine operating conditions. They performed experiments in an optical engine using premixed iso-octane, assisted by the two-stage reactions of n-heptane. They used high-speed imaging to visualize the NFL and Cantera-based data processing to understand detailed combustion kinetics. They observed a shifting combustion behaviour from two-stage ignition to three-stage ignition dominated by the retarding of the iso-octane injection timing due to reduction of local air-fuel mixture reactivity and weakening of chemical interaction between the HRF and LRF, which lowers the HRR. These three stages of heat release correspond to LTHR, HCHR and PCHR. The Cantera analysis of the PCHR showed that varying the HRF injection timings towards later injections resulted in different combustion reactions with more soot precursor formation potential (which is also observed in the higher luminosity of the NFL results) especially in the primary reacting regions. This is correlated with the higher stratification of both the fuel and flames associated with earlier injections. Results in Fig. 25 exhibited that the advanced injection timings (SoI = -27° aTDC and -7° aTDC) resulted in an earlier ignition kernel, primarily near the combustion chamber bowl. In SoI = -27° aTDC, dominant premixed combustion was observed, which was seen as blue ignition kernels appearing at 3°CA. A significant heat was released rapidly at this stage, due to which the whole combustion chamber was occupied with the blue flames at 3.7°CA. After this, a dominant effect of fuel stratification was observed in the combustion images; however, after ~9°CA, the luminosity was dominantly controlled by the soot oxidation.

A combined analysis of n-heptane and iso-octane reaction mechanisms exhibited several reactive species formations from the premixed n-heptane promoted iso-octane, which further accelerated combustion. The consumption of n-heptane (injected at -360° aTDC) was similar, independent of the SoI of iso-octane. During the consumption of n-heptane, CH_2O and CO accumulated while OH remained at a low level until a rapid consumption of iso-octane led to prompt heat release.

Tang et al. [185] investigated the predicted distributions of temperature and recalculated chemical HRR for three SoI cases at 10 mm below the cylinder head (Fig. 26). They observed a much wider main heat release region in the space, which approached the central part of the combustion chamber when advancing the SoI timings for the iso-octane. The stratified heat release was observed at all SoI timings. At retarded SoI timings, the primary reacting locations moved towards the upstream spray, which is the fuel-rich region. The PCHR is dominated by the combustion of the iso-octane, as the heat release contribution from the n-heptane is low. In the premixed combustion of n-heptane, heat release was predominantly controlled by the reaction: $HCO + O_2 = CO + HO_2$; however, this reaction shifted to hydrogen-oxygen reaction ($H + O_2 (+M) = HO_2 (+M)$) in the presence of increased temperatures. Both

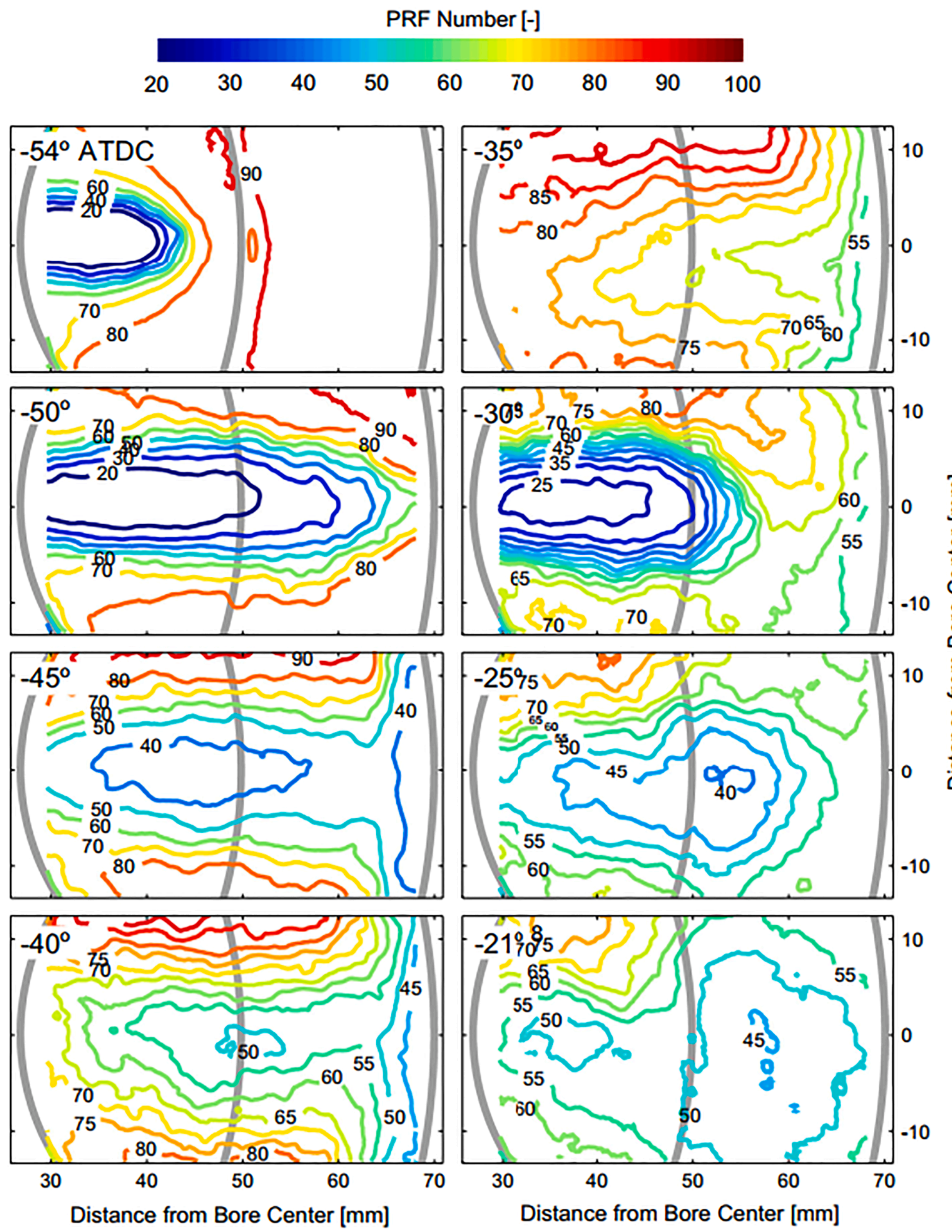


Fig. 24. Sequence of ensemble-averaged PRF maps at several crank angle positions during a common-rail injection event (Reprinted from [135] with permission of Elsevier). Each image's upper left-hand corner shows the time in crank angle degrees after TDC. The PRF maps were generated from the vapour-fuel concentration measurements with the camera viewing downward through the cylinder-head window. The relative error in the PRF number images is approximately 7% of the measured value with a filter size of 0.25 mm and 3.4% of the measured value with a filter size of 1 mm.

reactions belong to the intermediate-temperature reactions, and the equivalence ratio of the premixed mixture was low. Thus, the HRR remained low in these stages. The heat release was dominated by the $HO_2 + OH = H_2O + O_2$ reaction during the primary combustion stage. The formation of OH is propitiated by the $H + O_2$ (+M) = HO_2 (+M) reaction surrounding the primary reacting kernels and furthers the combustion process.

Fig. 27 shows the mass fractions of NC_7H_{16}, IC_8H_{18}, CH_2O, CO, and OH at the TDC at different SoI timings for iso-octane. In advanced SoI timings (SoI = −27° aTDC), the availability of more time for premixing resulted in relatively lower in-cylinder temperature and HRR, especially in the HCHR regions. This also led to greater cooling of the spray, resulting in higher non-reacting n-heptane and iso-octane. Results also showed that more CH_2O and CO were produced in advanced SoI timings than in the SoI = 3° aTDC case. However, the formation regions for these species were different in all cases. For SoI = −27° aTDC case, CH_2O and CO mainly formed in the large fuel regions; however, in SoI = −7° aTDC, CH_2O and CO mainly formed around the spray periphery. They also reported the formation mechanism of these species and suggested that reactions $C_7H_{15} + O_2 \rightarrow C_7H_{15}O_2$ and $AC_8H_{17} + O_2 => AC_8H_{17}O_2$ are important in the presence of higher n-heptane and iso-octane. This was mainly due to their close relationship with the low-temperature reaction pathways. However, heat release at a higher temperature was dominated by the reaction $HCO + O_2 = CO + HO_2$, which laid the foundation for intense HTHR. The higher reactivity is located in the CH_2O-rich regions, which the authors established a correlation to the whole mixture reactivity. In the earlier SOI cases, the formation of CH_2O is dominated by $C_2H_4 + OH = CH_2O + CH_3$ in the high-temperature region (with the delayed SoI scenario). The precursors are CH_2OH for the SoI = -27° aTCD and CH_3O for the SoI = -7° aTCD case. In all cases, it should be highlighted that the reaction of iso-octane enhanced the formation of reacting species that accelerated combustion.

In summary, this section focused on the fundamental investigations of RCCI combustion mode using optical diagnostics and simulations approach, explaining the combustion processes in totality, including the kinematic processes, control techniques, and flame evolution, among other descriptive conditions during combustion. Several studies that implemented optical diagnostics techniques have been included,

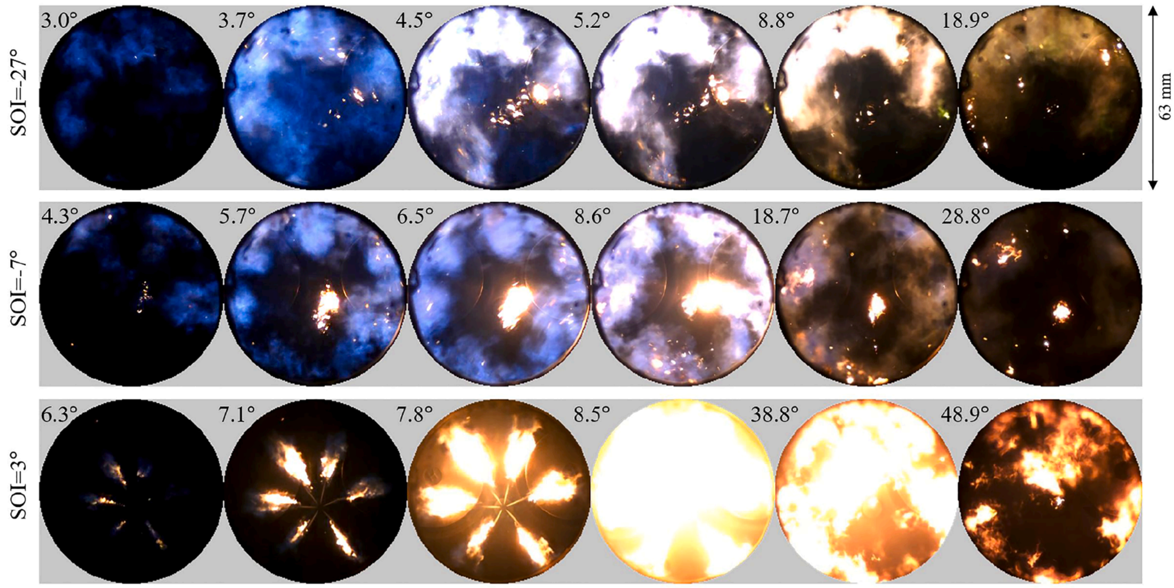


Fig. 25. Experimentally measured natural flame luminosity images at various crank angle positions (Reprinted from [185] with permission of Elsevier).

indicating that the flame propagation under RCCI combustion mode was highly sensitive to fuel stratification. Higher r_p promoted higher flame propagation. It also emerged from the review that local fuel reactivity controls ignition sites, with high reactivity zones being more likely to be ignited first, e.g., downstream of injected HRF jets. Optical techniques such as PLIF have some limitations regarding dimensionality and definition of the combustion phases. Some researchers provided information about the spatial distribution of formaldehyde and OH radicals during RCCI combustion mode. Also, using optical diagnostics, the effects of the SoI on different stages of the RCCI combustion mode were investigated, especially when the SoI was delayed more. In this case, the first stage autoignition occurred at the perimeter region of the cylinder (squish region), representing higher reactivity gradients. The second autoignition stage occurred in the combustion chamber's central region, representing lower reactivity gradients. Delayed SoI exhibited a slower rate of formaldehyde consumption and OH radical formation. A combination of optical and modelling techniques was also undertaken in some studies, which provided elaborate new information on the chemical kinetics of the RCCI combustion mode.

2.5. Effect of Control Parameters on RCCI Combustion Mode

Several studies investigated various parameters to control the RCCI combustion mode, among which the LRF quantity was an important one. A few studies indicated that RCCI combustion mode with optimum LRF quantity complied with steady-state NOx and soot emission limits imposed by the EURO VI emission regulations without using the after-treatment systems [151,156,291]. The LRF quantity must be low at low loads to increase the combustion stability and reduce the HC and CO emissions. At medium engine loads, a relatively higher LRF quantity could be used (which may reach the maximum levels); however, a moderated LRF quantity should be used to control the maximum in-cylinder pressure and the PRR [292]. Wang et al. [293] investigated the RCCI combustion mode in different r_p of methanol and reported that high r_p resulted in unstable combustion, especially at low engine loads. This led to misfire and significant cyclic variations, resulting in higher HC emissions and reduced fuel economy. Several interventions, including intake air heating, methanol heating, etc., were explored to avoid such events. Pan et al. [294] performed RCCI combustion mode investigations to explore the potential of intake air temperature variations as a control parameter. They reported that increasing intake air temperature resulted in relatively superior combustion stability. RCCI combustion mode exhibited lower HC, CO, and formaldehyde emissions at higher intake air temperature, especially while using higher proportions of fumigated methanol. The overall objective of implementing such techniques was to enhance the methanol vaporization, which directly affected the degree of completion of combustion. It has already been demonstrated that mineral diesel fuel injection parameters play an important role in modern common rail direct injection (CRDI) diesel engines. In a diesel engine, fuel injection parameters directly affect the spray characteristics. Fuel-droplet size distribution in the combustion chamber and fuel-air mixture homogeneity are critical for a diesel engine's good performance and emission characteristics. In diesel engines, pilot injection results in relatively lower peak HRR, leading to smoother combustion [295]. The pilot injection also results in superior engine performance and lower HC emissions [295]. Therefore, the effect of fuel injection parameters of the HRF, namely FIP, the SoI timing, and the number of injections, were also assessed to optimize the RCCI combustion mode for the varying engine load. Liu et al. [296] conducted the DMDF combustion investigations using mineral diesel and methanol. They explored the effect of FIP of diesel on engine combustion, performance, and emission characteristics. They reported that increasing FIP of mineral diesel resulted in relatively superior engine performance characteristics due to optimized combustion phasing (near TDC) and shorter combustion duration. Increasing FIP of mineral diesel led to relatively lower HC, CO, and smoke emissions; however, NOx emissions increased slightly. Higher CO_2 emissions at higher FIP of mineral diesel were observed due to improved DMDF combustion. The other important finding of this study was a relatively more dominant response of varying FIP in the DMDF combustion mode than in the CDC mode.

Singh et al. [152] explored the FIP variations of mineral diesel (as HRF) on the RCCI combustion mode at different r_p of methanol (as LRF). They reported that increasing FIP led to higher knocking in baseline CI combustion mode; however, RCCI combustion mode exhibited superior engine performance, especially at higher FIP and higher r_p. Another important observation was a relatively weaker effect of FIP variations on the HC and CO emissions from the RCCI combustion mode. However, the PM emissions correlated strongly with the FIP. Due to greater penetration of the HRF at higher FIPs, increasing FIP resulted in lower PM emissions from the RCCI combustion mode. They also analyzed experimental results to identify a suitable FIP for the RCCI combustion mode (Fig. 28).

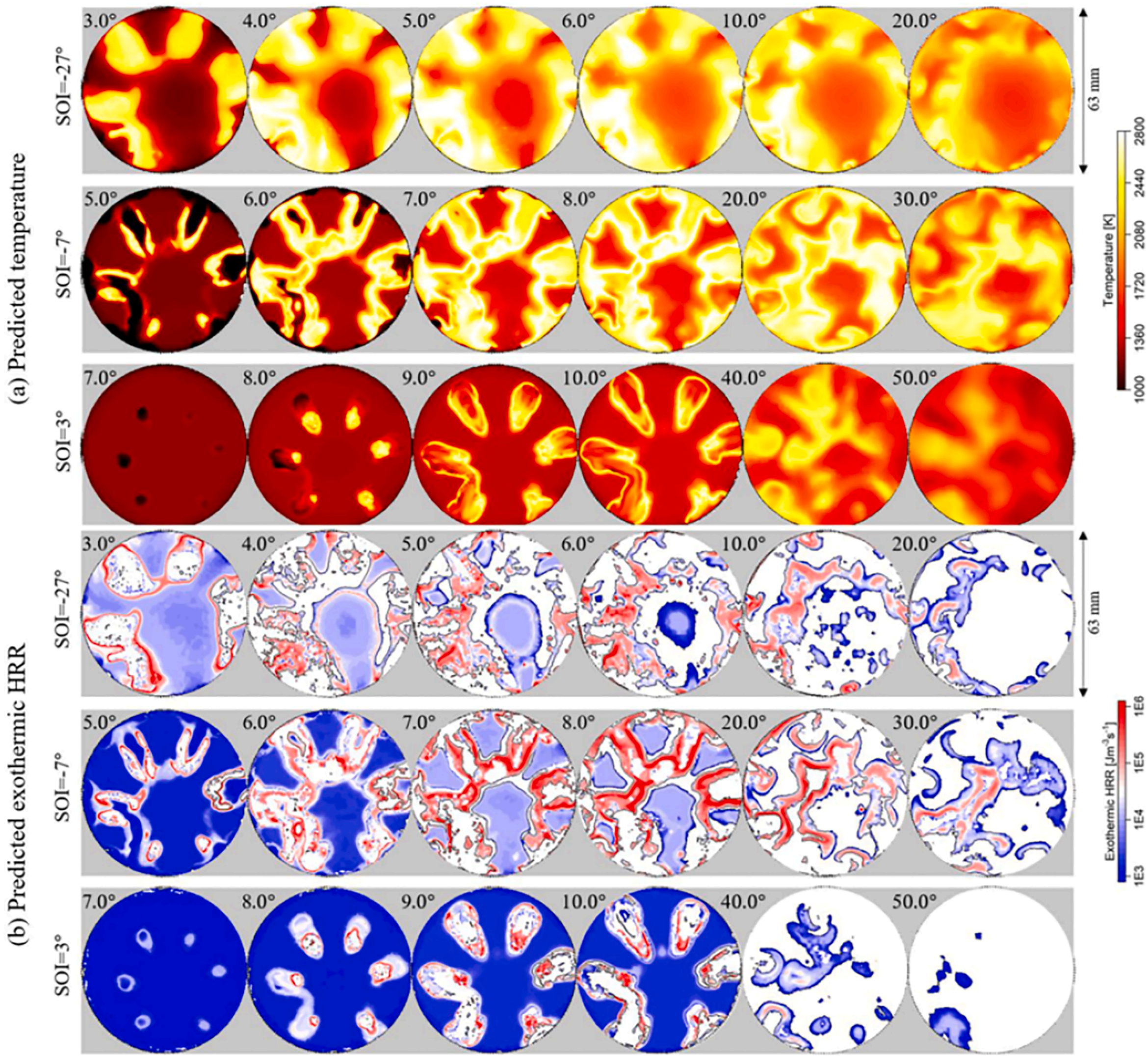


Fig. 26. Comparison of the predicted distributions of (a) temperature and (b) exothermic HRR at various crank angles. The predicted results are on a plane 10 mm below the cylinder head (Reprinted from [185] with permission of Elsevier).

This optimization exercise exhibited that the engine performance and particulate characteristics of RCCI combustion mode were significantly affected by the FIP of the HRF. Contour maps of TPN in Fig. 28 (a) showed a weak effect of FIP variations on the TPN emitted by both baseline CI and RCCI combustion mode engines; however, the r_p exhibited a relatively stronger correlation with the TPN, which decreased with increasing r_p of methanol. Combined analysis of TPN and TPM contours exhibited that a combination of 750 bar FIP at $0.50 < r_p < 0.75$ of the LRF was the optimum range for the lowest TPN and TPM emitted by the RCCI combustion mode. Fig. 28(b) showed a strong correlation of NOx emissions with the r_p of methanol; however, increasing FIP did not significantly reduce NOx emissions. Walker et al. [297] also explored the effect of FIP of mineral diesel on RCCI combustion mode. They reported that increasing the FIP of mineral diesel led to superior control over the combustion phasing. Li et al. [250] explored the effect of the SoI timings of mineral diesel on the RCCI combustion mode. They reported that advancing the SoI timings of mineral diesel resulted in relatively more stable combustion. At higher r_p of methanol, advancing the SoI timings of mineral diesel led to superior fuel economy. Mohammadian et al. [176] explored a single fuel RCCI combustion mode strategy fueled with isobutanol + 20% DTPB. They performed a detailed investigation to explore the effect of different DI parameters, namely SoI timing, FIP, and spray cone angle of the HRF and the r_p of the LRF. They reported that SoI timing was an essential parameter in single fuel RCCI combustion mode, which exhibited that advancing the SoI timing of the HRF (from 58° bTDC to 88° bTDC) resulted in superior engine performance and reduced emissions than the baseline case (SoI timing= 58° bTDC). The pilot injection was the other important parameter for controlling the RCCI combustion mode. Results showed that pilot injection and main injection resulted in relatively higher in-cylinder temperature and pressure than a single injection, which promoted the combustion at lower engine loads and higher r_p of the LRF.

Suh et al. [298] reported that pilot injection provided favourable conditions for the main injection, leading to superior ignition. Due to these advantages of pilot injection, several researchers explored the optimum pilot fuel quantity with respect to main injection fuel quantity because too much pilot injection quantity results in lower engine efficiency. Wei et al. [218] reported that increasing pilot fuel injection quantity resulted in relatively higher peak in-cylinder temperature. The use of pilot injection also changes the HRR pattern in the RCCI combustion mode, which becomes bimodal for higher pilot fuel injection quantity. The HRR characteristics also depend on the r_p of methanol. Increasing the r_p of methanol changes the HRR pattern from bimodal to uni-modal for a constant pilot fuel injection quantity. Wei et al. [218] suggested that the charge cooling effect of methanol (due to higher latent heat of vaporization) was the main reason for this trend, leading

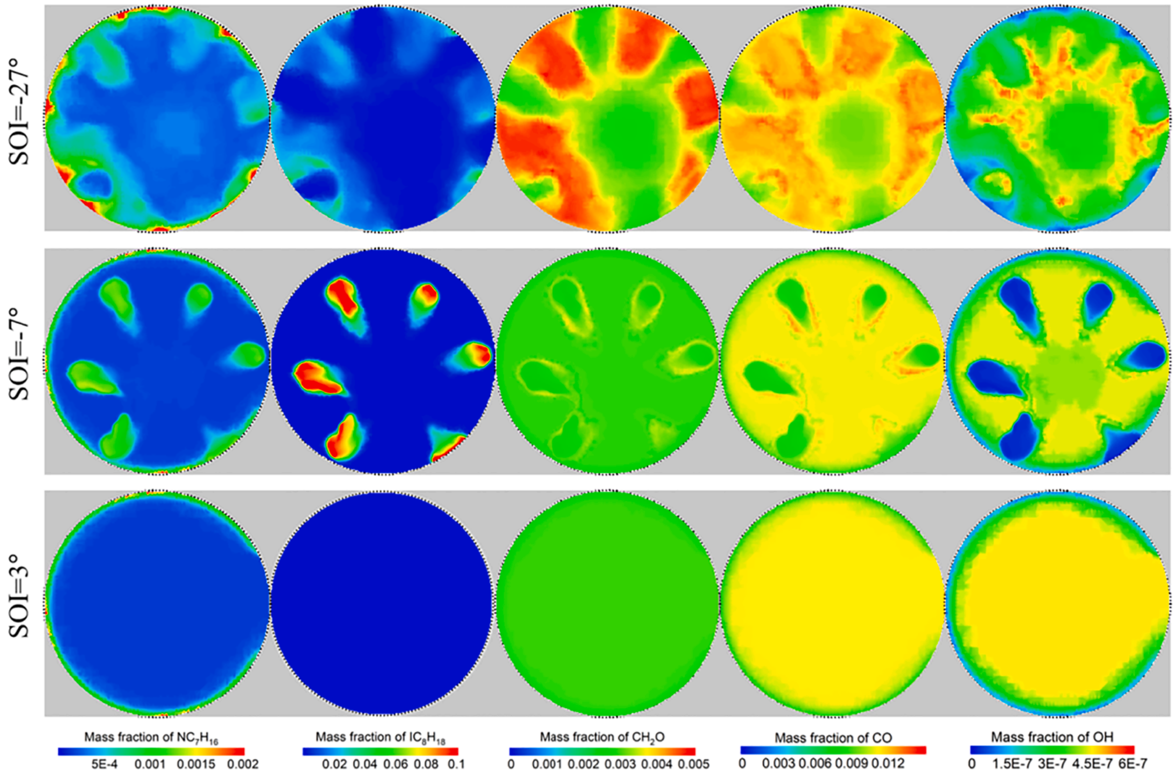


Fig. 27. Comparison of predicted distributions of mass fractions of NC_7H_{16}, IC_8H_{18}, CH_2O, CO, and OH at TDC at different SoI timings (Reprinted from [185] with permission of Elsevier).

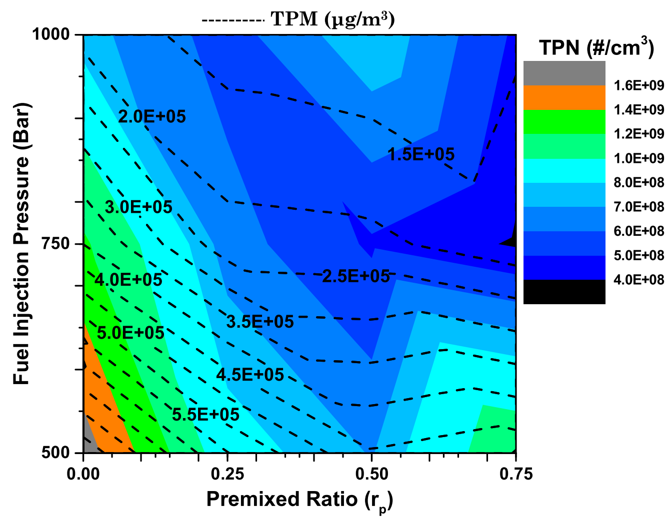
to dominant premixed phase combustion, resulting in a single peak in the HRR curve. SoI timing of pilot injection was also explored by many researchers, who reported that advancing pilot injection timing resulted in higher peak in-cylinder temperature. Wei et al. [218] concluded that a relatively larger pilot fuel injection quantity at advanced pilot injection timing was suitable for achieving higher fuel efficiency from the RCCI combustion mode at a higher r_p of methanol (M50). Their study also reported that most emissions, including CO, HC, and unregulated species, decreased with increasing pilot fuel injection quantity and advancing pilot injection timings. Jia and Denbratt [172] explored the effect of different methanol injection strategies in RCCI combustion mode. They compared the performance and emission characteristics of the RCCI combustion mode achieved by port-injected and direct-injected methanol. They reported that port injection of methanol resulted in lower CO and HC emissions along with higher BTE. They also varied different control parameters to check the suitability of direct methanol injection and reported that port injection of methanol was superior in most conditions. Zhao et al. [186] also explored the potential of direct injection of LRF in RCCI combustion mode by using a combination of butanol and biodiesel. They performed the experiments using different biodiesel injection timings, butanol energy ratios and butanol injection pressures. They reported that direct injection of both LRF and HRF provided better control on combustion due to two-stage heat release, including premixed combustion of butanol and biodiesel in the first phase and diffusion combustion of remaining biodiesel in the second phase. These two phases can be adjusted by controlling biodiesel and butanol injection parameters, along with the r_p of butanol.

Several studies evaluated boosting and EGR to achieve RCCI combustion mode at higher engine loads. Wang et al. [299] investigated biodiesel-gasoline-fueled RCCI combustion mode using boosting, EGR, and late intake valve closing (LIVC) control and reported a significant improvement in the upper load limit of the RCCI combustion mode. However, biodiesel-gasoline-fueled RCCI combustion mode exhibited inferior engine performance at higher load operation than mineral diesel-gasoline-fueled RCCI combustion mode. The charge dilution using air and EGR was also explored to improve RCCI combustion mode performance (lower torque output). Although EGR dilution is preferred for achieving RCCI combustion mode due to its simplicity, air dilution exhibited higher thermal efficiency. This was mainly due to a relatively higher specific heat ratio with air dilution than the EGR dilution [232]. Wang et al. [232] performed RCCI combustion mode investigations using experiments and theoretical thermodynamic modelling to evaluate the effects of air and EGR dilutions at high loads on the ITEg.

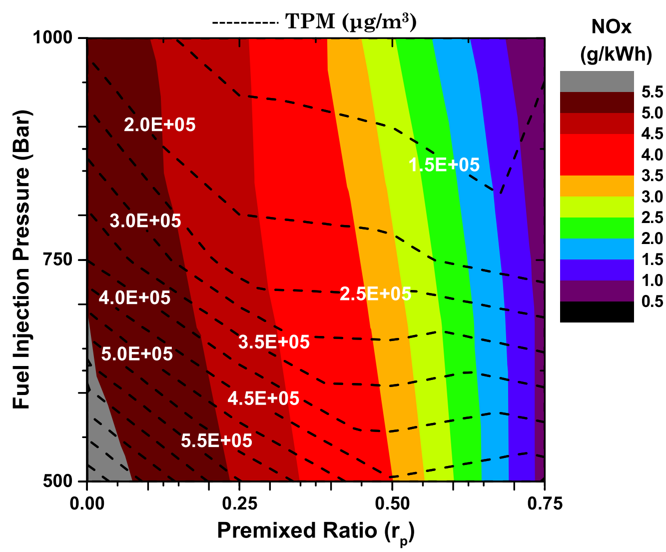
The authors reported that ITEg increased with decreasing EGR at a fixed intake air pressure of 240 kPa. The relatively stronger effect of EGR than the intake air pressure on ITEg was another important observation of this study, reflecting a more promising contribution of air dilution on ITEg improvement (Fig. 29). To better compare air and EGR dilution effects on ITEg, Wang et al. [232] used 'fuel-to-charge base equivalence ratio (Φ'),' which decreased with increasing EGR. Therefore, for maintaining a constant Φ', Φ should be increased with increasing EGR. Fig. 29 showed that the ITEg contours were closely related to air and EGR dilution at similar engine loads and combustion phasing. Results showed that ITEg increased with decreasing EGR; however, at a constant mass of air, ITEg decreased with increasing EGR dilution. They concluded that the thermal efficiency increased with increasing air dilution; however, thermal efficiency was hampered by the EGR dilution.

Olmeda et al. [142] also investigated the effects of mineral diesel's r_p, EGR, and SoI timing sweep. They reported that both EGR and r_p of gasoline affected the combustion efficiency significantly. Increasing the EGR beyond 20% at higher r_p of gasoline (more than 70%) resulted in inferior combustion, leading to higher HC and CO emissions. Although varying SoI timing of mineral diesel provided insights into its effects on the heat losses, however, for SoI timing variations from 30 to 60° bTDC, the heat losses did not show significant variations.

Desantes et al. [300] focused on the thermal efficiency of the RCCI combustion mode at lower engine loads. They performed simulations



(a)



(b)

Fig. 28. (a) TPM and TPN emissions, and (b) TPM and NOx emissions from RCCI combustion mode at different r_p of methanol and FIPs of mineral diesel w.r.t. baseline CI combustion mode (Reprinted from [152] with permission of ASME).

and experimental investigations to explore the effect of oxygen concentration on the RCCI combustion mode. They reported an improvement of ~1.5% in combustion efficiency using the combined effect of oxygen concentration and in-cylinder fuel blending (ICFB). Bora et al. [272] performed the RCCI combustion mode investigations to assess the effect of CR on the RCCI combustion mode and reported relatively superior engine combustion and performance; however, NOx and CO_2 emissions increased with increasing CR. This increase was due to improved combustion mainly, leading to higher peak in-cylinder temperature. Singh et al. [148] also explored the EGR and intake charge temperature (ICT) as control parameters to effectively control the RCCI combustion mode.

The authors performed experiments at different r_p and reported that increasing EGR resulted in relatively more stable combustion (Fig. 30 a); however, at higher r_p of methanol, combustion degraded due to the excessive cooling effect of the EGR. In contrast to EGR, the effects of ICT were significant at higher r_p to reduce the in-cylinder cooling and charge cooling effects of EGR and methanol, respectively. They concluded that

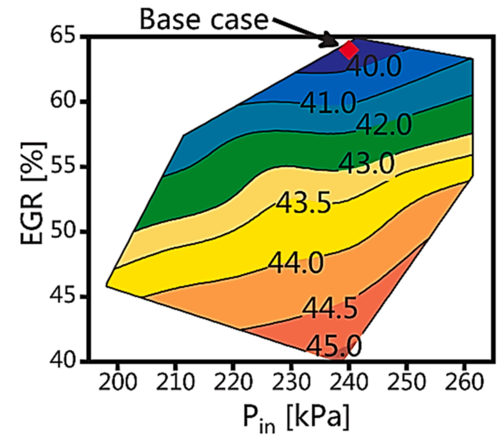


Fig. 29. Experimentally measured ITEg plot as a function of EGR and intake pressure (Reprinted from [232] with permission of Elsevier).

the combined effect of EGR and ICT variations could be an effective solution to achieving a stable RCCI combustion mode along with lower HC and CO emissions (Fig. 30).

This section presented an overview of recent literature on the effect of control parameters on RCCI combustion mode. By the definition of RCCI, the main control parameter explored is the reactivity that defines this combustion mode; hence the quantities of LRF and HRF were extensively investigated. Some studies indicated that by optimizing the proportions of LRF to HRF, the RCCI combustion mode engine should easily comply with steady-state EURO VI regulations for NOx and soot. It was revealed that the optimum r_p was dependent on the engine load, being the highest possible at medium loads since at low loads, higher proportions of LRF may lead to higher CO and HC emissions or unstable combustion. In contrast, at high loads, it could lead to excessive maximum pressure and PRR. Some other combustion control strategies were explored to enhance the combustion control and prevent relying exclusively on the reactivity gradients of the fuels. One such strategy was to increase the intake air's temperature, which resulted in higher combustion stability depending on the proportion of the LRF. Another commonly explored strategy was to optimize the fuel injection timing, the number of injections, and the FIP of the HRF. It was concluded that for the RCCI combustion mode, higher FIP could be beneficial for improving the combustion stability and reducing the PM emissions, contrary to the CDC mode. However, FIP didn't greatly influence CO and HC emissions. Advancing the SoI of the HRF also improved the combustion stability of the RCCI combustion mode. Pilot injection controlled the RCCI combustion mode, especially at lower loads, increasing the in-cylinder temperature and pressure, allowing higher r_p. Air management strategies studied by researchers to control the combustion and emissions in RCCI combustion mode were EGR and boosting. Albeit ITEg reportedly increased with reducing EGR rate. EGR helped to maintain a more stable RCCI combustion mode, though.

3. Challenges and Limits of RCCI Combustion Mode

HCCI and PCCI combustion modes have several drawbacks. The main difficulties lie in the combustion control, high load operating range extension, and high CO and HC emissions at low load. Diesel (or HRFs) facilitates the auto-ignition and combustion under the PCCI combustion mode at low loads; however, combustion phasing and excessive PRR pose a challenge at high loads. The RCCI combustion mode adopts in-cylinder fuel mixing using a PFI to induct the LRF and DI to induct the HRF. An ideal fuel reactivity stratification can be obtained by varying the LRF/HRF proportions over a wide range of engine loads and speeds [301]. This concept was preceded by Inagaki et al. [104], who worked on PCI combustion. Gradual combustion progress was achieved by

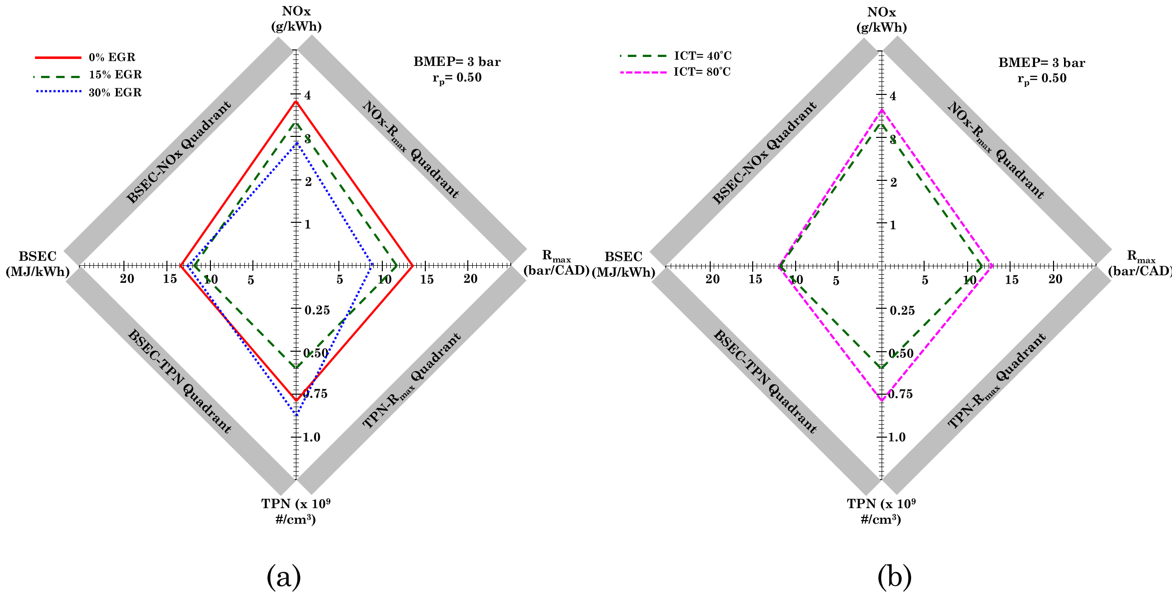


Fig. 30. Qualitative correlation between the combustion, performance, emissions, and particulate characteristics of RCCI combustion mode (at $r_p = 0.50$) at varying (a) EGR rates and (b) ICT (Reprinted from [148] with permission of SAE).

precise in-cylinder stratification, using a PFI system to supply the iso-octane and a DI system for diesel supply, and adjusting the mixture reactivity to meet the load requirements. This study effectively demonstrated that the operational map for the LTC concepts derived from the HCCI combustion concept could be extended for maintaining the soot and NOx at low levels and keeping the PRR under control. The PCI combustion concept evolved into the RCCI combustion concept after the advancements made by Kokjohn et al. [106,123,144], who coined this term and verified the results by Inagaki. Their work indicated the domination of the combustion sequence by successive ignition of zones having different mixture reactivities, from the most reactive to the least reactive ones. It was established that controlling the fuel blend for spatial stratification of the fuel reactivity could control the combustion duration. Several researchers stressed that the RCCI combustion mode could overcome many limitations of the HCCI and PCCI combustion mode [130,302]. RCCI combustion mode, however, was not immune to drawbacks. It has been extensively studied, and several reviews have been done on this topic [25,303], which cover some of the drawbacks and potential of the RCCI combustion mode concept. However, a comprehensive review on overcoming the challenges and real-world applicability is unavailable in the open literature. The next section summarizes studies related to the challenges which must be addressed for the commercial use of the RCCI combustion mode concept, proposed solutions to various challenges, a few examples of real-world applications, and the prospects of this new concept.

3.1. Challenges in Implementing RCCI Combustion Concept in Real Engines

There is general agreement in the scientific community about the main issues faced in the commercialization of RCCI combustion mode. These include higher CO and unburned HC emissions at low loads and excessive PRR at high loads. Other identified problems include slower flame propagation, knocking, lower thermal efficiency at low loads, lower exhaust gas temperature (EGT), which reduce the efficiency of after-treatment systems, and increased complexity for the fuel injection system and controls. Other issues include higher specific fuel consumption, which is closely related to the RCCI combustion mode engine's efficiency. However, the main issue lies in the operating range since both high and low loads cause undesired effects. Often, the strategies to extend the range in one direction negatively impact the other load extremes. Solutions proposed for the range issue are from hybrid combustion concepts that operate under one combustion mode or others depending on the load demands, such as the dual-mode dual-fuel (DMDF) concept given by Benajes et al. [304], to the utilization of alternative fuels with attractive properties that could potentially offset the negative effects at low and high loads [303], and the implementation of the so-called single fuel RCCI combustion mode, where through the addition of cetane improvers or reforming of the fuel on board, a fuel derivate with different properties can be obtained. The problems associated with RCCI combustion mode are identified, and solutions are discussed to develop this concept closer to reality.

3.2. Engine Speed-Load Limits in RCCI Combustion Mode

RCCI combustion mode is an LTC that uses at least two fuels with different reactivities to control combustion timing and phasing. This control is achieved by varying the ratio of the two fuels and adjusting their injection timing. This same principle also has undesirable side effects; for example, a high fraction of premixed combustion at high loads promotes excessive PRR, knock, and noise. The low combustion temperature at low loads does not allow complete fuel-air mixture oxidation and increases the CO and HC emissions. Another unforeseen consequence of the advanced LTC concept is the low exhaust temperatures, at times lower than the lighting-off temperature of DOC and air management system requirements. These requirements are almost impossible to achieve outside the test rigs, therefore requiring some complex system additions.

Defined operational limits for RCCI combustion mode have been investigated [305]. They reported that on a 17.1:1 CR, serial production 1.9 L engine platform at 1000 rpm, the combustion concept limits were within 2-5 bar BMEP, while at 3000 rpm, the limits were from 4-8 bar BMEP. These limits respect limitations such as smoke under 0.1 filter smoke number (FSN) and NOx below 0.4 g/kWh. However, the problem remains in extending the operational limit to the ranges of conventional CI and SI engines. For this reason, the main limiting factors are addressed in this review paper so that the focus can be objective-oriented. These limiting factors of discussed in the following sub-sections.

3.2.1. Intake Air Loop Requirements (EGR and Turbocharging)

For resolving the issue of excessive HRR, the researchers have been

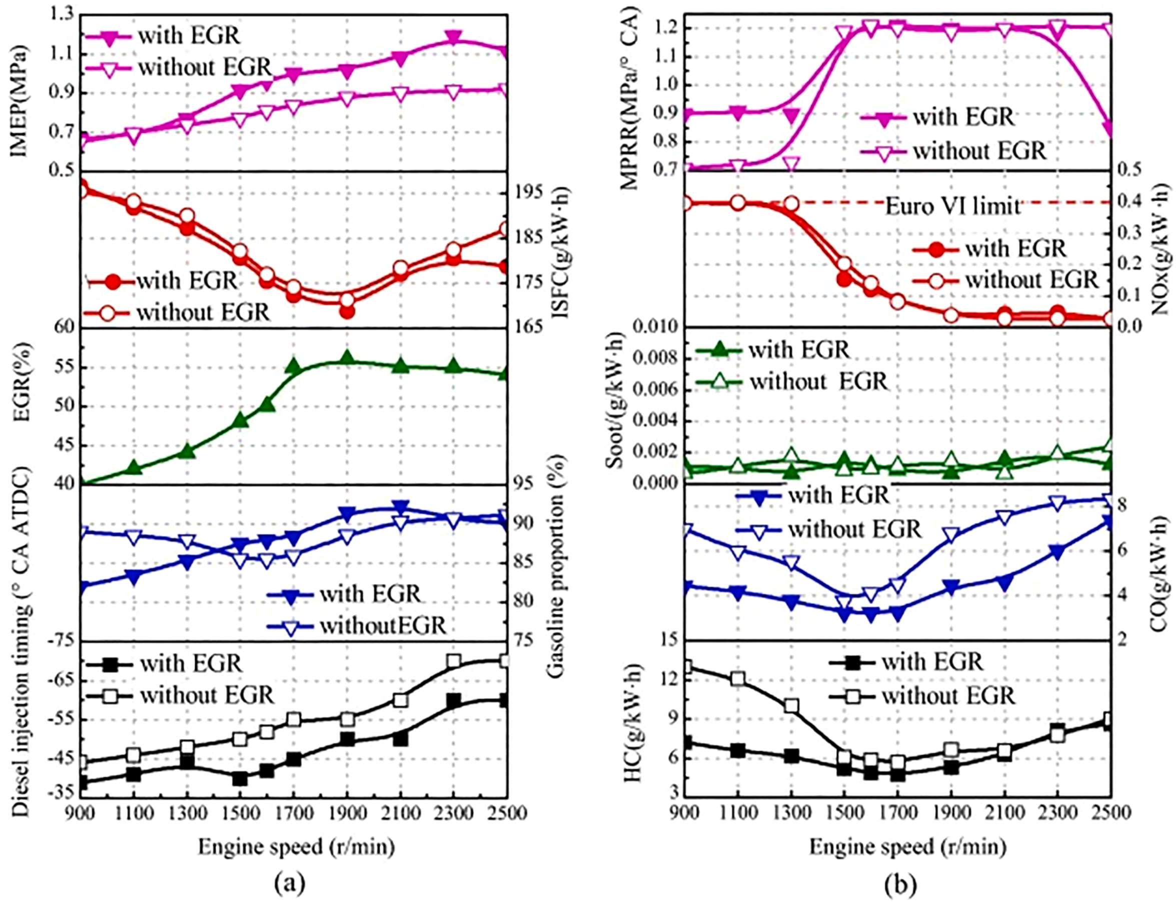


Fig. 31. Operational parameters and associated exhaust emissions corresponding to the maximum load conditions in RCCI combustion mode at each engine speed investigated (Reprinted from [307] with permission of Elsevier).

prompted to increase the charge dilution, and for managing this, excess air and EGR are proposed quite often. Wu and Reitz [306] reported that combustion and emissions improved with a higher boost while simultaneously reducing EGR requirement and sensitivity to emissions in RCCI combustion mode at high loads. High inlet pressures bring some implementation difficulties to the light. While a boosted engine is necessary to induct excess air, the existing turbochargers cannot always supply the required inlet air pressure and flow rates. This can be resolved by using a large air management system that surpasses any commercially installed turbocharger capacity. For a real-world application, the design of a higher capacity air management system would be necessary or a significant improvement in the control of RCCI combustion mode to not require high air-flow rates.

On the other hand, EGR induces similar flow problems with an additional factor of temperature limitations; hence any new system must consider these constraints. EGR reduces the PRR; however, RCCI combustion mode is greatly sensitive to variations in EGR rate, especially at high loads. Nevertheless, an early diesel injection can slightly reduce EGR sensitivity compared to a late diesel injection [306]. In this case, the early injection changes the charge composition, reducing the chances for auto-ignition of the premixed charge of the LRF. It is worth mentioning that contradictory results have been reported for boosting and EGR for HCCI combustion mode (and by extension to other LTC methods, like RCCI combustion mode). Some studies reported that the mixture dilution by Boost and EGR could help the combustion remain within peak pressure and PRR limits. In contrast, some other studies reported an increase in both these parameters. These contradictions have been noted [63]. The latter effect was attributed to reducing the heat transfer to the cylinder walls, increasing the combustion speed. It is, however, generally agreed that EGR extends the load limits for the LTC modes without breaching the imposed limits for peak pressure or the PRR.

High EGR rates are necessary to extend the load range. Nonetheless, large quantities of EGR at low loads must be accompanied by increasing the HRF fraction. Low load diesel-fueled LTC (IMEP ~0.23-0.26 MPa) has been investigated and compared with the diesel-gasoline RCCI strategy [307] on a single-cylinder heavy-duty engine. The results indicated that the proposed method was suitable for low load operations, albeit produced slightly higher NOx and soot than RCCI combustion mode. In Fig. 31 a, an increase in EGR was necessary for the increased load, and the injection timings were more advanced to prevent the LRF from generating knock. Fig. 31 b shows the emissions and maximum PRR, which is of particular interest in this study. A plateau existed at medium loads and speeds for HC and CO emissions.

With the application of EGR in RCCI combustion mode, CO and HC emissions reportedly increased in another study [227]. However, with an early diesel injection, a slight reduction in HC and CO was observed with a penalty of higher NOx emissions [306]. EGR rate is also highly dependent on the initial temperature [308]. If the initial temperature is high, EGR is essential to extract the NOx benefits and prevent undesirable autoignition of the LRF. EGR, however, has some effect on the ringing intensity of the engine when maintaining a constant combustion phasing. At different (higher) loads, only a small fraction of EGR would be required, further indicating that a fraction of LRF can control combustion [308]. RCCI combustion mode regime needs increased global reactivity of the in-cylinder charge (in the same way as CDC mode) to extend the low-load operating range. However, this neither prevents higher CO and HC emissions nor higher fuel consumption that LTC

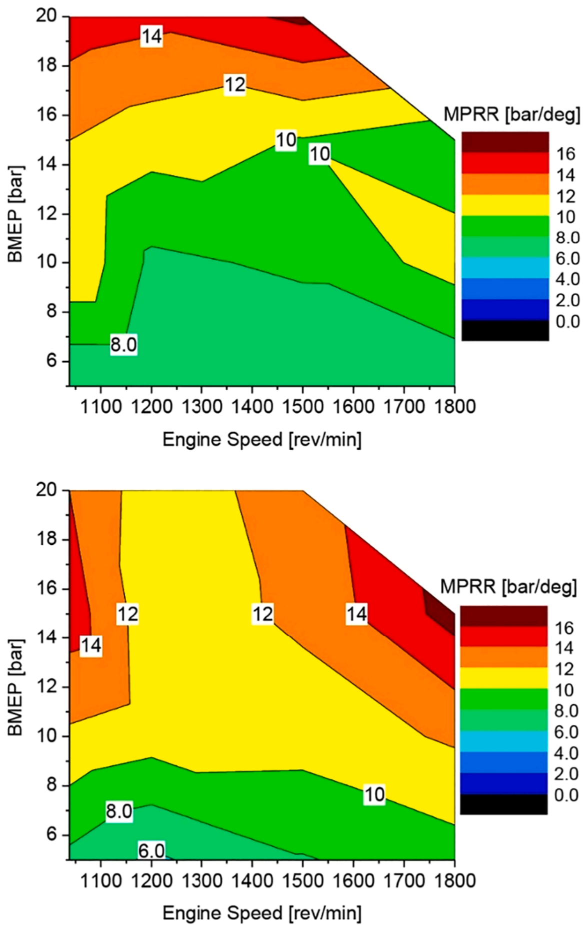


Fig. 32. MPRR for EGR (top) and non-EGR (bottom) operations over the entire engine operating map (Reprinted from [311] with permission of SAE).

strategies often suffer from [307]. A similar principle would be reviewed later to extend the RCCI concept by incorporating multi-mode operation, where more good combustion is obtained at each load and speed condition.

High boost pressure requirement emerged as a limitation to realising the full potential of the RCCI combustion mode concept at high load conditions [309] because the LRF fraction has to be reduced, and the HRF fraction has to be increased to cater to the necessary load conditions. However, a further study demonstrated that this might not be as limiting as previously thought. At full load, the boosting and EGR rate requirements were similar to those required in the production-grade engine (3 bar boost and 30% EGR) [310]. Hanson et al. [311] demonstrated the effect of these properties on the RCCI combustion mode while investigating the emissions and performance of a 13L multi-cylinder heavy-duty diesel engine modified for dual-fuel operation (PFI of natural gas and direct injection of diesel) both with and without the EGR over the EPA Heavy-Duty 13 mode supplemental emissions test. They confirmed that EGR reduced the engine noise to below 97 dB while simultaneously reducing NOx emissions by 48%, even though a slight increase in soot and thermal efficiency were observed. Using EGR in this study was to achieve the lowest possible NOx without significant thermal efficiency loss. Hence, only a minimum required EGR was applied to control pumping losses and soot emissions. EGR also reduced HC and CO emissions due to the engine's lower exhaust mass flow rate. With EGR, inlet temperature was also higher, which helped oxidize cycle-averaged CO and HC. With EGR, 1.53 g/kWh of HC emissions were observed, while without EGR, they were 2.27 g/kWh.

3.2.2. *Mechanical Limitations (PRR and Noise)*

High speed and high load conditions are problematic due to the mechanical limitations related to the structural integrity of the engine. Higher PRR and peak pressures are often a consequence of more efficient combustion, which heavily affect the engine noise characteristics and lead to higher stresses on the engine components, reducing their durability [312]. PRR is an important factor affecting engine knock and is an issue in SI engines. However, PRR can also affect the CI engines with premixed combustion in LTC modes. It is generally accepted that high PRR occurs because of the auto-ignition of gasoline (in this case, the LRF) before the flame front initiated by the spark plug (or autoignition of the HRF) reaches the combustible mixture. This, in turn, causes a ringing sound originating from detonation waves. Prolonged knocking may cause high wear of piston rings, cylinder-head erosion, the disintegration of the piston, and piston melting.

Hence PRR also limits the thermal efficiency achieved, besides undesirable noise [313]. High peak in-cylinder pressure is undesirable since it can exceed the safe operating pressure range, causing high stresses and fatigue in the engine components and adversely affecting the engine. High HRR in RCCI combustion mode at high-speed, high-load conditions can cause ringing. Studies on charge stratification by Li et al. [314] concluded that peak PRR could be reduced by retarding the injection timing. Researchers [307] examined the operational range for RCCI combustion mode from 900 to 2500 rpm engine speed and concluded that the engine load limit increases with the engine speed. Limiting factors for extending the high-load limit of RCCI combustion mode were excessive NOx formation at low engine speeds (because the LRF fraction needs to be reduced), excessive PRR at moderate speeds, and unacceptable in-cylinder peak pressure at high speeds. The exploration campaign found a low fuel consumption of 168.6 g/kWh at 1900 rpm speed and a 56% EGR rate. The importance of boosting for extending the operational range of RCCI combustion mode at low speeds was assessed. It reduced fuel consumption but generated excessive PRR. A high EGR rate and gasoline fraction were required at high speeds, and advanced diesel injection was required at low speeds to extend the RCCI combustion mode operating range. Fig. 32 shows the PRR achieved with and without the EGR [311]. The images show how EGR aided in reducing problematic levels of PRR across the entire operating map, reaching levels >12 bar/deg only in the higher load range. The absence of EGR promoted these values at medium loads for low and high engine speeds.

3.2.3. *Engine-Out HC and CO Emissions and Aftertreatment System Efficiency*

One of the main issues for RCCI combution mode is the higher emissions of incomplete oxidation products, i.e., HC and CO. The injected and premixed LRF at advanced crank angles in the RCCI combustion mode can get trapped in the crevices [19], increasing the unburned HC emissions due to incomplete burning. This can happen particularly at lower engine loads because the combustion propagation is weaker due to insufficient fuel-air mixing. RCCI combution and other LTC modes suffer from reduced CO conversion, posing challenges to the after-treatment systems. Although RCCI combution mode can help avoid the need for SCR after-treatment system due to its very low NOx emission levels; however, the DPF and the DOC become important to convert the higher quantities of CO and HC that remain unoxidized in the LTC modes. After-treatment systems need suitable boundary conditions to operate satisfactorily, and inlet temperature is one of those important boundary conditions. The EGT of the RCCI combustion mode engine is the inlet temperature for the after-treatment system. Obtaining adequate inlet temperature has been a challenge for efficient operation of the DOC, more so at low engine loads. Hence unburned emissions of HC and CO remain disproportionally higher. DPF regeneration also suffers from negative consequences because of lower inlet air temperature, leading to inefficient passive regeneration. The lower soot and the NO_2 concentrations do not properly activate efficient passive regeneration.

Therefore, active regeneration using higher oxygenation at higher temperatures effectively cleans the DPF.

Several attempts have been made to reduce engine-out HC and CO emissions before reaching the after-treatment devices. Among the options, direct injection of both the HRF and the LRF is thought to simplify the reactivity and equivalence ratio independently [315], controlling the fuel quantity in the crevice regions and reducing unburned HC. This strategy reduced higher NOx emissions than the traditional incorporation of LRF via PFI. However, direct injection of LRF did not show the desired reduction of unburned HC and CO emissions. Appropriate spray targeting and control of the crevice flow [19,144] reduced the CO and HC emissions by reducing the localized fuel-rich zones.

Higher CO and HC emissions in RCCI combustion mode might not impede commercial implementation of this concept [315,316]. The oxidation of CO and HC can be achieved by using conventional DOCs with RCCI combustion mode , as long as the EGT remain $>200^{\circ}C$ to ensure catalytic activity. Advancing the evaluation of the performance of DOCs with RCCI mode, Garcia et al. [317] developed and calibrated a 1-D model for the DOC with RCCI combustion mode to define the device's size to comply with current emission standards. They analyzed the response of DOC in a vehicle system simulation under different driving cycles to find that the CO and HC emissions at the DOC outlet surpassed the desired range. The researchers then sized the device to achieve an acceptable emission level. They concluded that a volume four to six times bigger would be necessary to comply with the prevailing emission standards. It can be summarized from these studies that although DOC reduces exhaust emissions, to comply with current and future emissions regulations, resizing the DOC is necessary, along with fine-tuning the combustion to reduce engine-out CO and HC emissions.

3.2.4. Transient Cycles and Control Systems

Consequently, vehicle operation and engine operation are not restricted to stationary conditions. Gross and Reitz explored the complexity of transient RCCI combustion mode operation [318]. They indicated an expressed need for additional controls to avoid undesirable effects to achieving the RCCI combustion mode. A comprehensive review of the transient operation under RCCI combustion mode was presented by Paykani et al. [319]. The main theme was that the RCCI combustion concept was mostly tested under stationary conditions, and there could be differences under transient conditions. They showed how open-loop (OL) control systems based on maps could be more expensive to calibrate than the close-loop (CL) control based on the in-cylinder pressure signals. The authors referred to Saracino et al. [320] and Hanson [321] to suggest that CL systems can account for the variability in EGT and EGR among the cylinders. The literature indicated that transient operation in RCCI combustion mode is possible by adjusting the EGR, airflow rate, engine speeds, intake pressure, and pedal position for a wide range of engine operating conditions. Control systems are vital for implementing the RCCI combustion concept. Controllers have demonstrated an accurate tracking performance for desired combustion phasing. The study focused on a single fuel RCCI combustion mode using gasoline and a cetane improver to increase the reactivity of the directly injected gasoline [318]. They performed a step change of load from 1 to 4 bar BMEP at 1500 rpm. Before transient operation investigations, experiments at four steady-state points helped assess engine performance and emissions. Intermediate points were interpolated to improve smooth transitions in the instantaneous step changes. These points were then calibrated by changing the injection strategy, EGR, and fuel rail pressure to reach a predefined combustion phasing (CA_{50}). A CL calibration was used for these tests by employing a next-cycle (NC) controller to adjust the PFI fraction of each cycle to obtain the preset CA_{50} values. In contrast, engine parameters were adjusted following the 2-D maps. Results indicated that the single fuel transient operation is possible without significantly increasing the emissions.

In summary, the RCCI combustion mode emerged as an important LTC mode to resolve the issues posed by HCCI and PCCI combustion modes. RCCI combustion mode indicated that varying the HRF and LRF quantities could obtain an ideal charge reactivity. However, researchers also experienced some limitations that need to be resolved. This section reflects the challenges in adopting the RCCI combustion mode for commercial applications. From the emissions perspective, it is generally agreed that the main shortcoming of the RCCI combustion mode is relatively higher CO and HC emissions. Charge dilution strategies such as deploying higher EGR to increase the operational limits and prevent other problems can be useful if EGTs are not sufficiently high. Other factors limit the RCCI combustion mode, such as knocking probability at high loads and high r_p, low thermal efficiency at low loads, and unstable combustion. Low EGTs are the other effect of RCCI combustion mode, which remains to be addressed. Low EGTs can hamper the after-treatment system operations, such as for DOCs. With these main issues in mind, appropriate solutions must be devised. These solutions include superior spray targeting and control of crevice flows, which reportedly reduce HC and CO emissions. Several studies on the after-treatment system capacity have been undertaken to assess whether higher CO and HC emissions are a limiting factor for the commercial application of RCCI combustion mode. These studies considered reduced EGTs of the RCCcombustion mode engines to find whether the after-treatment systems could efficiently operate within these limitations. Finally, the control systems must be more robust for operations under transient conditions due to relatively higher injection system complexity for the RCCIcombustion mode engines than the CDC mode engines. Experimental studies have shown that though RCCI combustion mode operations might be more complex, they will soon be a real commercial possibility.

4. Implementation of RCCI Combustion mode in Real Engines

Most studies on RCCI combustion mode address the combustion control issues in reactivity because it is the core of the working principle of LTC. Li et al. [322] classified reactivity into two types: (i) global reactivity and (ii) reactivity gradient or stratification. The first was determined by the quantity and characteristics of the fuel (CN, octane number, LHV, etc.), while the second was dependent on the injection strategy, spray penetration, and entrainment of the HRF in the premixed charge. Kokjohn et al. [135] found that reactivity stratification is a leading factor in controlling the combustion phasing and ignition location, followed by equivalence ratio having a significant influence. At the same time, the temperature stratification effect was negligible. Because of the importance of the reactivities of the fuels in combustion characteristics of the RCCI combustion mode, their management, injection strategies, control, and concept extensions are worthy of investigations for engine implementation. For the RCCI combustion concept to work, at least two fuels with different reactivity are required. The source for these can vary from a single fuel to multiple fuels. The need for different fuels in the same engine/ vehicle platform increases control system complexity and cost. There are hurdles in adapting current generation engines to the RCCI combustion mode or designing clean sheet RCCI engines, but this concept has become feasible for commercialization. Hanson et al. [323,324] demonstrated the RCCI combustion concept using gasoline and ultra-low sulfur diesel in a hybrid platform on a 2009 Saturn Vue vehicle. Besides this, one engine control system supplier has a commercially available vessel that affirms the RCCI combustion mode retrofitting capabilities [325]. Recently, Argonon [326] showed promising commercial application possibilities of the RCCI combustion concept in large size engines by retrofitting a Caterpillar 3512 engine on an inland vessel MTS, which reduced up to 10% energy consumption when operated with biofuel instead of diesel while also complying with emission regulations without the need for DPF. The vessel's engine took advantage of the fuel flexibility of the RCCI concept to refuel without completely emptying the tank. This implied that it is possible to have a commercially available RCCI engine that relies on the fuel flexibility of the RCCI combustion concept for its use. However, it is important to explore possible solutions for the downsides of the RCCI combustion concept using the current state-of-the-art technologies available.

4.1. Multi-Mode Concepts to Cover the Entire Engine Map

RCCI combustion concept has limitations over the load extremes, both high and low. Hence researchers have focused their efforts on finding a solution for these limitations without sacrificing benefits (such as reduced NOx and PM emissions) or further deteriorating the weaker aspects (such as higher HC and CO emissions). Another boundary condition of RCCI combustion mode limits the peak pressure or PRR. The multi-mode concept offers an alternate solution to this issue. The multi-mode concept limits RCCI combustion mode operation to an engine operating range where the combustion is optimized. Then, the engine transitions to another mode, which would offer superior performance under other engine operating conditions. Not only have that, but mode-switching RCCI combustion engines also retained the advantage of lower NO_x emissions compared to the CDC mode.

4.1.1. Dual-Mode Dual-Fuel (DMDF) Concept

The dual-mode dual-fuel (DMDF) combustion mode extends the RCCI combustion mode over the high and low loads. Different load zones are then catered by different combustion modes depending on constraints imposed by the PRR, NOx, soot, and maximum in-cylinder pressure. These modes could be fully premixed combustion and dual-fuel diffusion combustion modes. PRR and in-cylinder pressure limitations are imposed to prevent mechanical failures. The optimized multi-mode combustion strategy combines a fully and highly premixed RCCI combustion regime at low and medium loads and dual-fuel diffusion combustion at full load [155,156]. In the cited studies, the authors indicated that this strategy maintains PRR below 15 bar/CAD and maximum in-cylinder pressure below 190 bar while simultaneously covering the entire engine map with engine-out NOx emissions below EURO VI limits [155]. The authors could reach up to 14 bar IMEP and maintain soot emission under 0.8 FSN in most engine operational zones, reaching values as low as 0.02 FSN at 7 bar IMEP. DMDF combustion relies on reducing the CR to resolve the limitations of the RCCI combustion mode at full load [327]. The reduced CR then helps mitigate the undesirable ignition of LRF and reduce the PRR. Benajes et al. [327] emphasized extending the RCCI combustion mode because of the emission improvements in the entire global engine map. To do that, they evaluated the RCCI/CDC mode-switching vis-à-vis CDC, depending on the coverage of the RCCI combustion mode regime, while trying to maximize its share to complete the Real Driving Emission cycle. Garcia et al. [291] compared the DMDF combustion to the RCCI/CDC mode-switching combustion to cover the unattainable load range of RCCI combustion mode with CDC mode. They reported that DMDF reduced specific fuel consumption by 7%, and engine-out NOx emissions complied with EURO VI limits and were 87% lower than the RCCI/CDC mode. DMDF still showed higher CO and HC emissions (up to 10 times higher than other modes) which could be addressed by using exhaust gas after-treatment systems.

4.1.2. RCCI/CDC Mode Switching

Benajes et al. [301] explained that the dual-fuel RCCI/CDC concept hinges upon switching between the RCCI combustion and CDC modes to cover the entire engine map. They evaluated the performance and emissions of this concept by simulating vehicle systems operating under different driving cycles (Real Driving Emissions (RDE) Cycle, Worldwide Harmonized Light Vehicle Test Cycle (WHTC), Federal Test Procedure (FTP-75), and Japanese cycle (JC08)), with the experimentally obtained diesel-E85 and diesel-gasoline engine maps on a light-duty diesel engine having a CR of 17.1. Their results indicated that this concept could be used in flexible fuel vehicles (FFV). They concluded that E85 as LRF could extend the operating limits of the RCCI combustion mode with lower NOx and soot emissions but with higher HC and CO emissions.

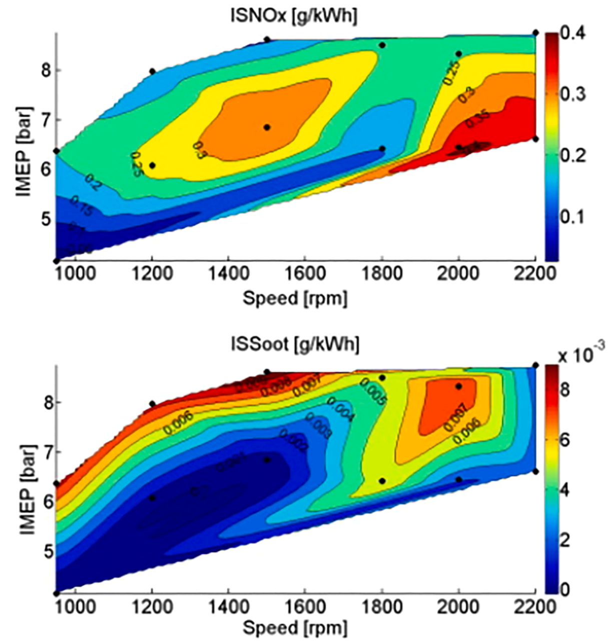


Fig. 33. NOx and soot emissions mapping of RCCI combustion mode operation on a high CR EURO VI engine (Reprinted from [310] with permission of Elsevier).

Another interesting conclusion of the real-world application of this concept was that the amount of fuel necessary for dual-mode RCCI combustion mode using gasoline was almost the same as required for the CDC mode. Hence no additional fuel storage space would be required in the vehicle. A 32.5 L tank for diesel and 27.5 L tank for gasoline would be sufficient while maintaining the same vehicle range as the CDC. Prikhodko et al. [316] and Benajes et al. [328] indicated dual-mode RCCI combustion mode as an alternative combustion mode (CDC in this case) without the need to reduce the CR when the RCCI combustion mode operating window was rather limited.

Benajes et al. [310] indicated that the operating range of RCCI combustion mode was between 25% and 35% load because of the limits imposed by PRR and peak in-cylinder pressure. The rest of the engine operating conditions in the engine map were catered by CDC mode, improving the overall oxidation of HC and CO while maintaining their peak values at ~37 g/kWh and 23 g/kWh, respectively. It is worth noting how the CO and HC emissions are superior to the results produced by a higher CR engine [309]. The investigations were conducted in a single-cylinder engine using a gasoline-ethanol blend (80%-20%) as LRF and diesel with 7% biodiesel as HRF. The RCCI operational range provided a 2% improvement in gross indicated efficiency. Very low NOx and soot emissions from RCCI combustion mode complied with EURO VI emission norms, as shown in Fig. 33.

Recent works on LTC mode switching explore how to cover the entire engine map by switching between CDC, conventional dual-fuel combustion, HCCI, RCCI, PCCI, PPCI, and piston-split dual-fuel combustion (PDFC) modes [329]. The authors divide the operation modes according to the proportion of LRF and HRF and the timing of the injections of the HRF to early or advanced CAD, as shown in Fig. 34. Test and driving cycle simulation results indicated that selecting the correct mode for each zone of the operational engine map can decrease NOx and soot emissions by ~20% compared to CDC and, as the thermal efficiency is increased, also provide a reduction of ~1% CO_2 emissions. It should be understood that since the multi-mode concept uses the CDC approach to expand the operating map of the RCCI combustion mode engine, the SCR and DPF cannot be eliminated from the after-treatment system of the engine. Finally, the RCCI combustion mode engine emerged as a very attractive concept exhibiting excellent efficiency and reduced fuel

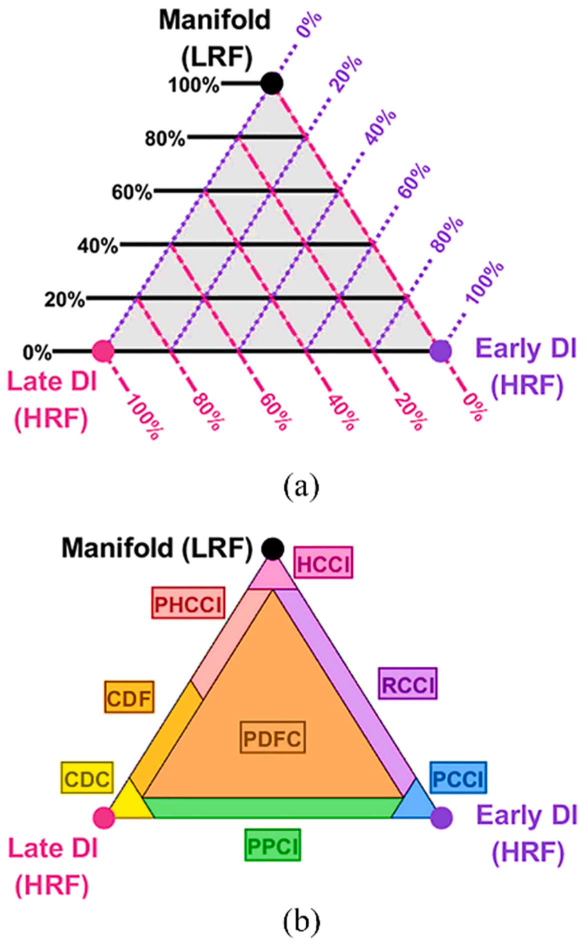


Fig. 34. The Manifold/ Early DI/ Late DI Triangle diagram or 'MELT'' diagram plots the distribution of the total fuel energy into the three primary injection types considered (a) shows isolines of fuel energy percentage, while (b) applies acronyms from the literature of combustion modes that are expected to occur with certain fuel distributions (Reprinted from [329] with permission of SAGE).

consumption.

4.1.3. Hybrid RCCI Combustion Mode Coupling

Electric vehicles (EVs) are supposedly important in decarbonising the transport sector and reducing urban air pollution. Light-duty vehicles have been substituted by plug-in electric vehicles (PHEV) without significantly impacting the end-user experience. New developments in batteries and energy administration controls have increased the range to a market average of 315 km [330]. The availability of publicly accessible charging stations increased by 60% in 2019 compared to the previous year [331], and energy costs have become comparable to conventional vehicles [332]. Even though EVs are gaining relevance in the transport sector [332], IC engines will remain the most important workhorses for the transport sector globally in the near and medium-term future. IC engines have not yet encountered a feasible electric powertrain challenge to fulfil the distance-range and cargo-weight demands of the medium- and heavy-duty vehicle segments while maintaining operational and logistics costs. As more EVs penetrate the automotive fleet, an intermediate step between full electrification and straight IC engines is required. Hybrid Electric Vehicles can serve as a bridge between both realms, the EVs and the ICEs, and offer a superior solution, taking advantage of both. A hybrid RCCI-electric powertrain is highly attractive because of its potential to get emissions to regulation levels and even lower while also reaping the benefits of lower fuel consumption. Additionally, the electric operation can serve as a primary power source in

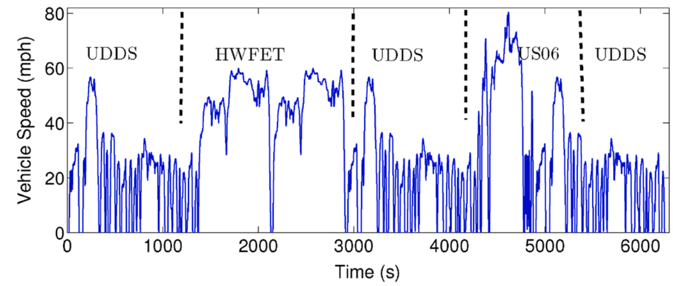


Fig. 35. The combined driving cycle for evaluating the designed EMC (Reprinted from [333] with permission of MDPI).

the load range where the RCCI combustion mode has trouble.

Hybrid powertrains would allow RCCI combustion mode operation with its load constraints, as demonstrated by Solouk and Shahbakhti [333]. They designed and implemented three energy management control (EMC) strategies, namely rule-based control (RBC), dynamic programming (DP), and model-predictive control (MPC), to improve the fuel economy of the hybrid system combined over a combined driving cycle entailing the urban dynamometer driving schedule (UDDS), highway fuel-economy test (HWFET), and US06 driving cycles (Fig. 35) [333]. The investigators considered switching modes between the electric operation and combustion leads to a fuel penalty, adding this insight into their results. The work was focused on fuel optimization, and it was observed how the RCCI-series hybrid electric vehicle (SHEV) exhibited superior fuel economy than a modern SI engine (12.6%) and a CI engine-based HEV (2.2%). This work also discussed the state of charge (SOC) influence on the fuel economy. A battery at low SOC in charge sustaining mode has a fuel economy advantage over a higher initial SOC. The authors indicated that the battery at low SOC demands the engine to run longer to compensate for the higher battery energy loss, providing the RCCI combustion mode with instances to save fuel that the tested SI and CI combustion modes do not offer.

Advancements in hybrid powertrains combined with dual-fuel RCCI combustion mode were explored by Benajes et al. [334]. They evaluated Mild hybrid (MHEV), Full hybrid (FHEV), and Plug-in hybrid (PHEV) vehicles with diesel-gasoline RCCI combustion mode using 0-D numerical modelling. The simulation inputs were based on the calibration maps of a CDC mode and diesel-gasoline dual-fuel RCCI combustion mode of a GM 1.9L light-duty engine. The simulated vehicle consisted of a Class D passenger car, which had an additional electric motor and battery in the model since the vehicle was not originally an electric vehicle. Hybrid powertrain results were compared to their non-hybrid counterparts under the World Harmonized Light Vehicles Test Procedure (WLTP). They demonstrated that it is possible to attain NOx and soot levels below the EURO VI regulation. They achieved a 5% reduction in fuel consumption for only dual-fuel combustion mode while maintaining the NOx levels same as CDC mode. With the same fuel consumption that the OEM has, a 30% reduction in NOx emissions was obtained (Fig. 36). The dual-fuel combustion increased CO emission to 1.6 g/km compared to 0.8 g/km from the CDC mode. It was possible to obtain similar emissions from a hybrid powertrain as that of the OEM because of the use of high BMEP for recharging the batteries, thus promoting CO conversion. HC emissions did not improve by using a hybrid powertrain and exhibited values higher than those reported by the OEM. Benajes et al. [334] performed a lifecycle analysis (LCA) for these systems. They indicated that the RCCI combustion mode-based PHEV reduced CO_2e-missions by 30% compared to the baseline CDC mode vehicle in the cradle-to-grave approach.

Hanson et al. [323,324] tested a series-hybrid vehicle with an RCCI engine coupled to a 90 kW AC motor that worked as a generator for a 14.1 kWh Li-ion traction battery pack powering a 75 kW drive motor (Fig. 37). They reported that HC, CO, and NOx emissions were similar to those reported in previous laboratory tests. With appropriate

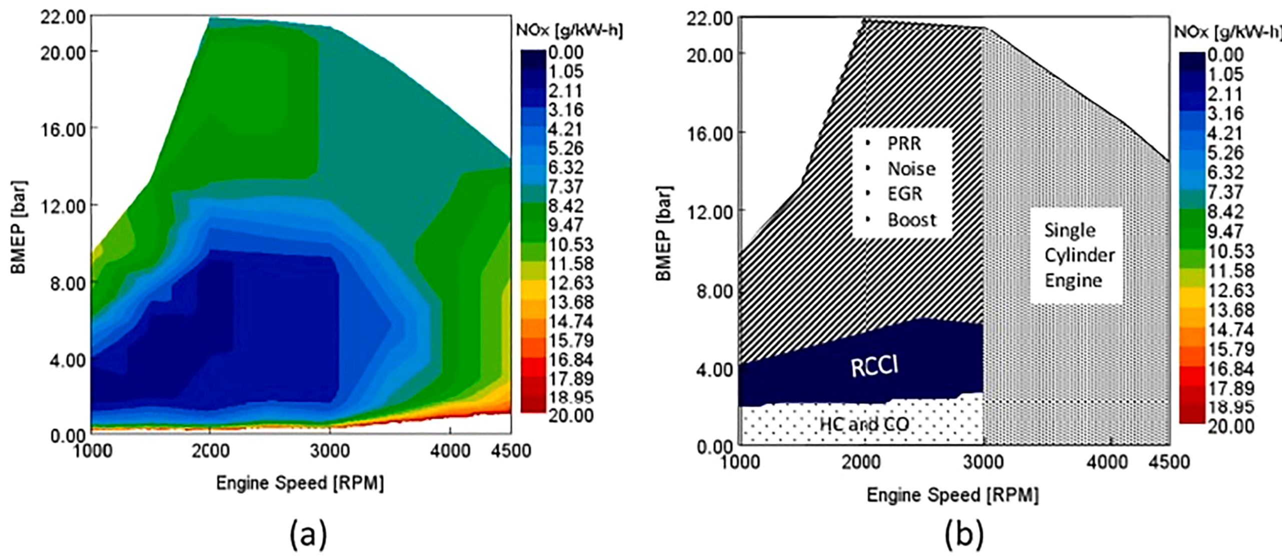


Fig. 36. (a) NOx emissions for the CDC mode calibration map and (b) reactivity-controlled compression ignition (RCCI) gasoline calibration map (Reprinted from [334] with permission of Elsevier).

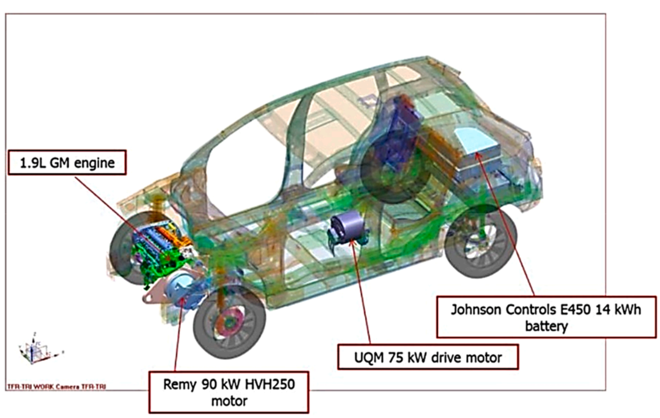


Fig. 37. UW hybrid vehicle drivetrain (Reprinted from [323] with permission of SAE).

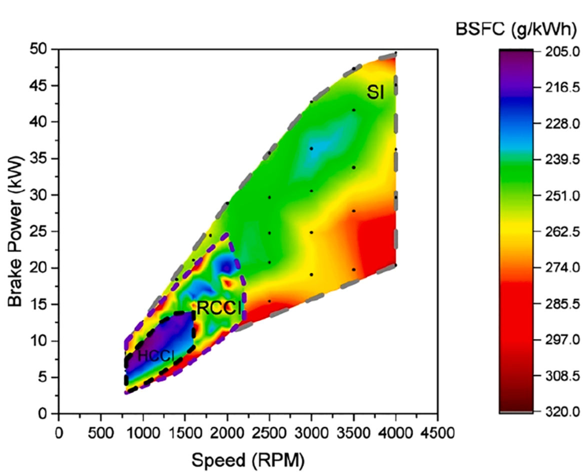


Fig. 38. BSFC map of multi-mode LTC-SI engine (Reprinted from [335] with permission of SAE).

modifications, the vehicle could comply with US EPA Tier-2 bin-5 NOx and CO levels for several test cycles with a fuel economy ranging from 4.4 L/100 km to 7.47 L/100 km. Additionally, they reported that exhaust gas after-treatment systems could reduce HC and CO emissions by 98.5% in the HWFET with a hot start. During the ORNL HWFET, the NOx emissions were higher than the steady-state laboratory tests because of a probable unoptimized fuel injection calibration. Fuel consumption was reduced to 0.3 L/km when operating at higher engine power demands because of the increased thermal efficiency it represented.

Solouk et al. [335] experimentally coupled different LTC modes with an Extended Range Electric Vehicle (EREV). The EREV allowed the decoupling of the engine from the drivetrain, thus making the LTC operation achievable only within its optimum limits. Their experimental setup allowed HCCI, RCCI, and conventional SI combustion modes. They used the data acquired to generate a BSFC map and determine the load limits for each combustion mode (Fig. 38). Tests were performed for single-mode EREV coupling (HCCI-EREV, RCCI-EREV, and SI-EREV) and a multi-mode LTC-EREV. They reported 9% and 10.3% improvements in fuel economy in a single-mode RCCI-EREV compared to SI-EREV coupling for the city driving cycle (UDDS) and the highway driving cycle (HWFET), respectively. However, with HCCI-EREV, the improvement was higher (12% and 13.1%). The improvement from one driving cycle to the other was explained by the longer engine operation duration in the high-power demand cycle (HWFET). There were no reported noise vibration harshness (NVH) or ringing issues, primarily because the LTC modes (RCCI and HCCI) were kept within the low engine speed range where these issues are not prominent. The LTC modes exhibited improvement over the SI combustion mode since the LTC modes at low vehicle speeds could operate at low engine speeds while the SI combustion mode has to do so at high engine speeds. Another conclusion from this study was that the LTC-EREV multi-mode operation could improve the fuel economy in the vehicle more than the single-mode operation could. Thus, the RCCI+SI-EREV emerged as the best combination in the high-power demand driving cycles, and HCCI+RCCI emerged as the best in the low to mid-power range driving cycles (Fig. 39).

4.2. Single fuel RCCI Combustion mode

Functionally, the concept of RCCI combuston with two different fuels can have substantial advantages in obtaining NOx and soot emissions

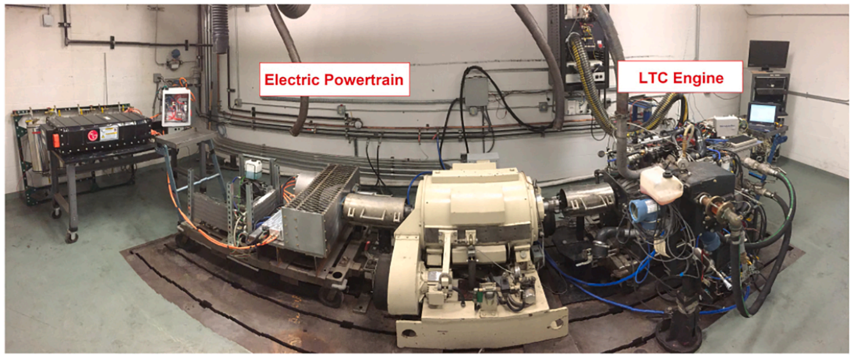


Fig. 39. LTC-Hybrid electric powertrain experimental test-bed utilizing a double-ended 465 hp AC dynamometer at Michigan Technological University (Reprinted from [335] with permission of SAE).

below the regulatory limits; however, operationally, the use of two fuels can bring in some complications. These include (i) the need for two fuel tanks in the vehicle, (ii) different depletion and recharging rates for the fuels, which could potentially deter users from adopting the mode due to slightly increased fuel recharge scheduling complications, (iii) increased complexity of fuel injection control systems, as well as the use of two separate fuel injection systems, which would increase the cost of the vehicle. The single fuel RCCI combustion mode was explored as a viable alternative for these reasons. There are several ways to realize single fuel RCCI combustion mode. The main approaches for this include use of fuel additives and reforming. Both methods seek changes in the fuel composition and reactivity of the primary fuel so that the derivate fuel can act as a second fuel to manage the reactivity stratification. These approaches are discussed in the following sub-sections.

4.2.1. *Single fuel + Additives*

To apply the RCCI combustion mode concept with a single fuel tank, a method to change the reactivity of the fuel to allow reactivity stratification can be used. Cetane improvers are the most common path to achieving single fuel RCCI combustion mode. These additives could be DTBP and 2-EHN [283]. Cetane improvers are applied to a portion of the LRF, which is then directly injected into the cylinder as the HRF. The rest of the LRF is injected through the PFI system [131,133]. Many studies have been done on single fuel RCCI combustion mode engines using cetane improvers [76,122,177]. Single fuel RCCI combustion mode could be accomplished by using only the LRF (e.g., gasoline, methanol, ethanol, iso-butanol) with a cetane improver, which enhanced the autoignition characteristics of the LRF. The required fractions of cetane improver were reportedly quite low, ranging between 0.75% and 5% v/v. However, the amount of secondary fuel (the HRF obtained by the mixture of LRF and additives) will vary significantly from application to application. In the case of iso-butanol with DTBP, high concentrations of DTBP were required to raise the reactivity sufficiently [177]. Higher quantities of DI fuel were required (60-70%) compared to lower diesel quantities required for similar conditions (25-40%). Splitter et al. [133] tested DTBP with gasoline at mid-load conditions and used relatively smaller quantities of improver (0.75%, 1.75%, and 3.5% v/v). The same study reported that a lower fuel injection pressure (400 bar) for the HRF was necessary with gasoline-DTBP than a diesel counterpart (800 bar). Peak gross indicated efficiency of 57% was obtained with acceptable emissions in this study.

It was reported by Liotta [336] that an a2-EHN fraction increase caused a linear increase in NOx emissions from the engine. The work states that the response was due to the presence of the nitrate group chemically bonded to the additive. To realize ultra-low NOx emissions from conventional RCCI combustion engines, Kaddatz et al. [179] used a light-duty single-cylinder engine with 10% ethanol-blended gasoline (E10) and 2-EHN as an additive (up to 3%v/v) in RCCI combustion mode. They then compared the results with dual-fuel RCCI combustion using E10 and diesel. Since the percentage of 2-EHN in this study was low, the NOx penalty was not very high due to single fuel and additive use. NOx penalty was below 1 g/kWh over a range of loads. In addition, the best peak indicated efficiency was quite similar to that of dual-fuel RCCI combustion mode at 50% load (9 bar gross IMEP). Single fuel RCCI was also investigated under transient conditions in a General Motors (GM) Z19DTH 1.9-L diesel engine having a CR of 13.75 [318]. Gasoline mixed with 2-EHN (3% v/v) was compared to conventional dual-fuel RCCI combustion mode under transient conditions. HC emissions were maintained under 1500 ppm, NO below 10 ppm, and smoke lower than 0.1 FSN while simultaneously keeping a 10 bar/CAD limit on the PRR and a moderate maximum noise up to 95 dB during calibration. The researchers proved that single fuel RCCI combustion mode operation could be achieved with similar HC levels to the dual-fuel RCCI mode counterpart, but with NOx penalty, while using gasoline with 2-EHN as the HRF.

Regarding cetane improvers, one aspect that should be highlighted is the possible effects changing the fuel reactivity could cause into mixtures' storage stability and safety. Cetane improvers have been included in fuels since 1940 [336]. More than two decades ago, using these peroxides with dual-fuel applications that included liquefied petroleum gas (LPG) and diesel [337] was tested, revealing the thermal stability of a 5% LPG-diesel CI engine configuration was comparable with an engine running exclusively on diesel. It is important to comment that cetane improvers usually are reactive and exothermically unstable peroxides in large concentrations. It is conceivable that some of these properties can be transferred to the fuel it is blended. Nonetheless, for applications of RCCI combustion, concentrations of these composites remain relatively low, and no significant effects were found on the fuel stability. Eng et al. [338] tested concentrations as high as 15% v/v, finding that the addition of larger concentrations (> 2%) did not produce large changes in the fuel ignition timings.

4.2.2. *Single fuel + Reformate*

Although cetane improvers theoretically require only one fuel tank for the RCCI combustion mode to work, the single fuel alternative does require an additional tank of a similar size as that of a diesel exhaust fluid (DEF) [319], which would be required to be recharged periodically. The fuel reformation strategy to obtain a secondary fuel having different reactivity from the same fuel to achieve RCCI combustion mode has been proposed to bypass this hurdle completely. The concept takes a fraction of the primary fuel and directs it to an onboard fuel reformer to produce a reformed gaseous mixture of partially reacted hydrocarbon species, hydrogen, and CO [339]. Catalytic partial oxidation (CPOX) is a reforming where a rich fuel-air mixture is reacted over a catalyst to produce CO, H_2, and other partially combusted hydrocarbon species [340]. Another reforming alternative is steam reforming (SR or SMR for the specific case of methane). This option requires a high-temperature heat source, steam, and fuel (methane), which are reacted over a catalyst to produce CO and H_2 [339]. SMR and CPOX are different. The first is endothermic, while the second is exothermic. Variable valve actuation can also lead to in-cylinder reforming using negative valve overlap (NVO). The NVO principle increases the auto-ignition capabilities of high-octane fuels at a CR typical of SI engines without the need to preheat the air [341] because the exhaust mass is compressed. This concept is useful for on-board fuel reforming because the hot recompressed exhaust provides the necessary conditions to initiate fuel reformation reactions gas [342]. The last reforming method mentioned in this review is the thermo-chemical fuel reformer (TFR) from dedicated EGR (D-EGR). D-EGR has a dedicated cylinder running stoichiometrically, whose exhaust gases are supplied directly into the intake manifold [289]. It can be concluded from various studies that single fuel RCCI combustion mode is feasible by using onboard fuel reforming. However, the addition of hardware elements increases the complexity of the fuel and control systems.

4.3. Dual Direct Injection (DDI)

Research shows that the most common RCCI configuration is using an engine with a DI injector in addition to a PFI. However, other configurations are also proposed, e.g., the dual direct injection (DDI) configuration. Unlike the DI/PFI configuration, the DDI involves direct injections for the LRF and the HRF at different injection timings. The intention behind direct injection of the LRF is to avoid fuel entrapment in the crevice region, which would reduce CO and HC emissions [343]. Lim and Reitz [344] performed RCCI combustion mode engine simulations using direct injection of iso-octane using a gasoline direct injection (GDI) injector. The LRF injected directly into the cylinder reduced CO and HC emissions by 27.1% and 7.1%, respectively, increasing combustion efficiency. DDI enhances the control over the mixing process and combustion phasing. It provides greater flexibility in reactivity stratification due to the possibility of distributing the LRF across multiple injections throughout the compression stroke. Some of the previously mentioned noise issues can be mitigated by this strategy while reducing the total unburned hydrocarbon emissions by up to 91% [345]. The direct dual-fuel stratification (DDFS) strategy reduced the combustion noise and improved combustion stability with lesser EGR required than conventional RCCI combustion mode, with a combustion phasing near TDC [346].

Direct injection of the LRF and HRF stratifies the fuel directly in the engine cylinder and reduces the charge premixing to maintain it under desired limits [347]. This approach is quite different from GCI combustion since the RCCI combustion mode has superior control over the combustion phasing. Besides this advantage, higher loads (up to 21 bar IMEP) were attained in the simulation of DDI of the RCCI combustion mode using iso-octane and n-heptane [348]. An ITE_g of 48.7% was obtained. On the other hand, n-heptane (as the HRF) mass and injection timing were reported to have a larger effect on the injection control by allocating a smaller quantity of HRF in the squish region before introducing the rest of the HRF to promote the ignition delay. Some drawbacks of the DDI are the need for higher injection pressure for both injectors, which adds complexity to the fuel injection and control systems. Additional space is also required in the cylinder head for mounting both the injectors. Yang et al. [349] proposed a low-pressure dual-fuel direct injection (LPDDI) concept based on air-assisted direct injection (AADI) technology (Fig. 39). The AADI consists of a built-up nozzle that incorporates a liquid injector and a gaseous injector, injecting diesel and gas mixture. The work substituted the compressed air traditionally used with the AADI system with methane, thus adapting the system for RCCI combustion mode application. The proposed system can produce direct injections for both the LRF and the HRF. The authors demonstrated that LPDDI could achieve RCCI combustion mode with diesel/CH_4. They underscored the importance of the LRF, and the HRF injection timings over the combustion phasing, specifying that optimized combustion was achieved by advancing the HRF timings (−250 °CA aTDC) and retarding the LRF injection timing (−112 °CA aTDC).

To summarize, commercially available RCCI engines are not yet a reality. However, the real-world implementation of the RCCI combustion mode has made significant advances in recent years. One of the main tasks to implement the RCCI combustion mode in commercial vehicles is adapting the dual-fuel injection system, but this increases the complexity of the vehicle systems and the control paradigms. Even with these design hurdles, the implementation of the concept has already been tested in cars and maritime vessels. In addition to the real-world proof of concept, other alternatives have been investigated to bring the RCCI combustion mode closer to commercial feasibility. For example, implementing multi-mode concepts, such as DMDF and RCCI/CDC, intends to resolve the limitations of the operational range by switching to another combustion mode when RCCI combustion mode proves to be inefficient or unattainable. Another proposed strategy was combining RCCI combustion mode in hybrid form for road vehicles. In particular, this strategy has proven to comply with US EPA Tier-2 bin-5 NOx and CO emissions in several driving cycles and indicated that the after-treatment systems could address the CO and HC emissions.

On the other hand, some studies focused on simplifying the systems necessary for obtaining the RCCI combustion mode. Strategies for single fuel RCCI combustion mode required improving the CN of the LRF, which two approaches could obtain: adding additives to the LRF and then using it as the HRF, and fuel reforming. Fuel reforming consisted of applying on-board processes to modify the reactivity of the LRF to facilitate controlled auto-ignition. Finally, the dual-direct injection was applied to achieve RCCI combustion mode. Using DDI technology, it was possible to reduce CO and HC emissions while maintaining high thermal efficiencies.

5. Conclusions, Recommendations, and Way Forward

The RCCI combustion mode has gained significant attention from the engine research community in the last few years due to its excellent combustion stability, efficiency, and emission characteristics. RCCI combustion mode could be adopted in production-grade engines offering a unique possibility of adopting alternative fuels. This review article has covered all features of RCCI combustion mode from end to end. The basic mechanism of all LTC variants is the same; however, the RCCI combustion mode deals with the reactivities of two different fuels, which are mainly governed by chemical kinetics. In extensive research related to LTC, it has emerged that all LTC modes have significant potential for emission reduction; however, a comprehensive analysis of HCCI and PCCI combustion modes of combustion revealed several limitations, especially related to their application at higher engine loads. RCCI combustion mode exhibited significantly greater potential in these aspects, explored in many research studies off-late, including in-cylinder combustion and optical diagnostics. This review article discusses detailed experimental methodologies, conclusively showing that the RCCI combustion mode has great potential to be adopted in production-grade engines. The dual-mode operation has been explored significantly among various other techniques due to its lesser complexity and greater applicability to cater to full engine load operations.

Research and development studies have demonstrated good potential for RCCI combustion mode to be applied to various engine platforms. Both light-duty and heavy-duty applications showed that the engine-out NOx emission levels were well below the limits imposed by stringent emissions regulations. In addition, lower soot emissions were observed without the need for after-treatment devices such as DPF and SCR. RCCI also delivers higher thermal efficiencies than other LTC modes. A flexible array of fuels can be used in this combustion mode, which opens the possibility of using fuels synthesized from renewable resources, thus reducing the overall carbon footprint of the transport sector. Besides these advantages, the RCCI combustion mode has some challenges, such as higher HC and CO emissions at lower loads and excessive PRR at higher loads. These issues restrict the full map

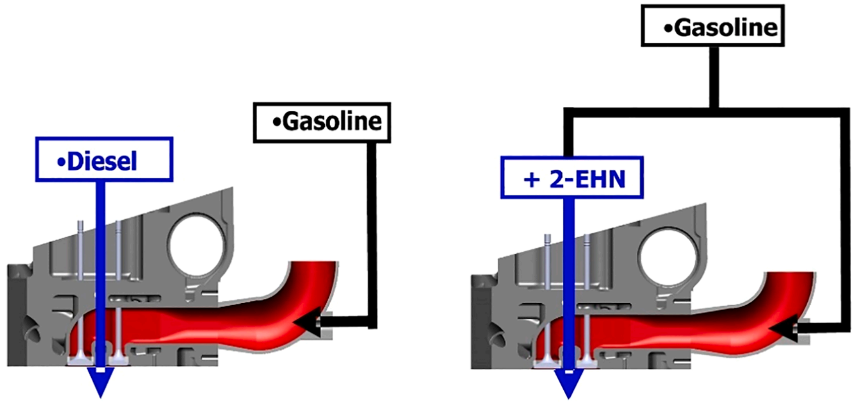


Fig. 40. Examples of dual-fuel and single fuel injection strategies for RCCI combustion mode (Reprinted from [131] with permission of SAE).

application to only medium engine loads. RCCI combustion mode offers several advantages; however, it still has some problems to overcome. These challenges limit the engine operation due to high PRR/knocking. Before commercialising the RCCI combustion mode engines, these issues must be resolved. Literature review shows that researchers are working to resolve these problems to make the RCCI combustion engine work in a low-environmental impact combustion mode, having good reliability and higher energy conversion efficiencies. This could make these RCCI combustion mode engines commercially attractive to the consumers and reduce the CO_2 footprint at the same time.

RCCI combustion offers very low NOx emissions, which can eliminate dependence on SCR systems and reduce PM emissions. Higher HC and CO emissions are a penalty because of low combustion temperatures. However, the after-treatment systems have shown excellent conversion efficiencies to comply with the emission legislation.

Various combustion control strategies have shown promising control over the PRR, such as retarding the HRF injection timings and introducing the EGR. These strategies help increase the mixture's resistance (previously introduced as LRF) to autoignition and help charge dilution. Higher boost pressures have shown a reduction of PRR, which prompts finding a way to obtain the required airflow under real-world operating conditions. Pathways to resolving these issues are partially completed by developing mixed combustion modes such as DMDF, which go from the pure RCCI combustion mode concept to combustion suitable for varying loads. Electrical hybridization of the RCCI combustion mode engine is also hugely beneficial since the electric motor can work over a load-speed range where the RCCI combustion mode engine struggles. The transient operations were verified under the discussed concept. The next logical step towards commercial implementation would be proving this concept in real driving conditions. Retrofitting current engines has also been of some interest to researchers and manufacturers. To further improve the prospect for this concept, the use of unconventional fuels with suitable properties facilitating the RCCI combustion mode could be explored. These would promote ultra-low soot emissions due to higher oxygenation (like OMEx) or a closer to net neutral CO_2 footprint due to its production cycle and even tank-to-wheel operation but are aspects that need to be explored further.

Once technical problems are resolved, commercial adoption of the RCCI combustion mode concept would require addressing functional issues such as reliability and fuel flexibility. A technical solution should be developed for an efficient control method to operate the engine in different combustion modes if one of the two fuels is unavailable. Further research and development of a single fuel RCCI combustion mode engine development would be a topic of interest for researchers, which would resolve several hurdles of having different fuel systems for the LRF and the HRF. Since costs are an important decision factor for the consumer and corporations, reducing the complexity of the fuel injection and control systems would also be essential for the commercialization of the RCCI combustion mode engine technology. With increasingly stringent emissions limitations, a methodology should be developed to assess whether the improvements in NOx and soot emissions are worth the CO and HC penalty and the increased system complexity and possible costs of having two distinct fuel injection systems. Besides some of the representative examples of real-world operation, the concept of RCCI combustion has not been commercially widespread, and other, simpler alternatives have been prioritized by manufacturers that can offer similar increases in thermal efficiency or emissions. The conversion efficiency of the after-treatment systems for the dedicated RCCI combustion mode engine would be an important development soon, as it could help determine whether the concept can make possible the emissions reductions necessary to reach global targets. (Figs. 40 and 41)

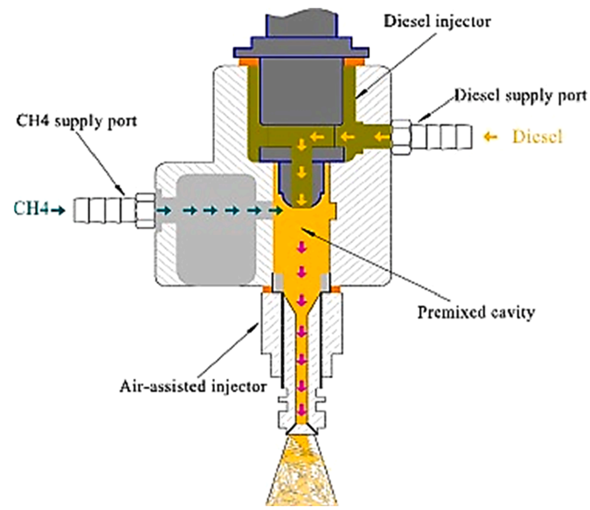


Fig. 41. Cross-section of AADI diesel/CH_4 dual-fuel injector model (Reprinted from [349] with permission of Elsevier).

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.

Supplementary materials

Supplementary material associated with this article can be found, in the online version, at doi:10.1016/j.pecs.2022.101028.

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