Chapter 5
Combustion Control Variables and Strategies

Abstract Low temperature combustion (LTC) engines need different enabling technologies depending on the fuel and strategy used to achieve combustion of the premixed fuel–air mixture. Controlling the combustion rate is one of the major challenges in LTC engines, particularly in HCCI combustion engine. To achieve higher thermal efficiency, the desired phasing of combustion timings is essential even at moderate combustion rates. Present chapter describes the combustion control variables and control strategies used for LTC engines. Various methods demonstrated to control the LTC engines can be categorized in to two main strategies: (i) altering pressure–temperature history and (ii) altering fuel reactivity of the charge. Temperature history of the charge in the cylinder can be altered by several parameters such as intake conditions (temperature and pressure), EGR, variable valve timing (VVT), variable compression ratio (VCR), water injection, supercharging and fuel injection strategies. Fuel reactivity of charge in the cylinder can be altered by various parameters such as equivalence ratio (Φ), fuel stratification, fuel additives, ozone additions and dual fuel. All these combustion control strategies are discussed for utilizing gasoline-like fuels in HCCI, PPC and RCCI combustion mode engines.

Keywords Combustion control • HCCI • EGR • PPC • NVO • Dual fuel • Additives • VVT • VCR • RCCI • LTC

5.1 Altering Time Temperature History

Time–temperature history of the charge in the cylinder determines the combustion timings, which essentially governs the combustion rate and thermal efficiency. In different LTC strategies, charge temperature history is controlled to achieve desired combustion phasing and extend the engine operating load. Temperature and pressure history of charge in the cylinder is correlated by gas equations. The time–temperature history of the charge is affected by several engine operating parameters discussed in the following subsections.

R.K. Maurya, *Characteristics and Control of Low Temperature Combustion Engines*, Mechanical Engineering Series, https://doi.org/10.1007/978-3-319-68508-3_5

5.1.1 Intake Thermal Management

Appropriate thermodynamic and chemical in-cylinder conditions are required in a HCCI combustion close to TDC position to achieve the autoignition at desired combustion phasing. Additionally, the air–fuel mixture in HCCI combustion needs to be sufficiently dilute to keep combustion rates and maximum charge temperature low enough to achieve acceptable ringing intensity (see Chap. 7) and NO_x emissions. Fuel typically used in HCCI combustion is gasoline-like fuels, which have higher autoignition temperatures. Intake preheating of charge is often used to enable the autoignition and phase the HCCI combustion at adequate crank position. Basic idea is to increase the temperature of charge at intake valve closing (IVC), where compression of charge starts. This temperature can be increased by increasing intake temperature or trapping the hot residual gases in the cylinder by negative valve overlap (NVO). Since in HCCI combustion engine is mainly kinetically controlled, it has the sensitivity to the charge temperature. Several techniques are used to increase the initial temperature of the charge at compression start position such as electrical preheating, exhaust heat exchanger, glow plugs etc. [1–4].

Figure 5.1 shows the required inlet temperature at IVC position as a function of engine speed and load in naturally aspirated HCCI engine using ethanol for different compression ratios. Higher intake temperature advances the combustion phasing, which increases the pressure rise rate drastically and sometimes above acceptable limits. Lower intake temperature retards the combustion phasing, which may leads to partial burn or misfire due to low combustion temperature. The lower temperature at retarded combustion phasing leads to the lower combustion efficiency. Operating limits shown in Fig. 5.1 used the acceptable pressure rise rate <5 MPa/ms and combustion efficiency $>85\%$. At particular engine load (IMEP), the temperature shown in Fig. 5.1 corresponds to most efficient inlet temperature (T_{IVC} with highest thermal efficiency). Figure depicts that higher T_{IVC} is required at lower IMEP for every engine speed at given compression ratio. Higher inlet temperature required is due to leaner mixture operation (lower reactivity) at lower IMEP conditions. Richer engine operating conditions require relatively lower intake temperature. For higher engine load conditions, comparatively lower T_{IVC} is required at higher compression ratio (Fig. 5.1). The higher compression ratio increases the compression temperature required for autoignition. Outside the operating range (Fig. 5.1), at higher intake temperature, the volumetric and thermal efficiency of the engine is very low. Intake temperature is required to be adjusted along with change in engine speed and load conditions as shown in Fig. 5.1.

Intake temperature control is usually thought as too slow response parameter for closed-loop control of transient HCCI combustion. The most direct implementation of intake temperature control is an electric heating element in the intake air path. The thermal inertia of the heating element itself causes a time constant of the order of seconds, which is indeed too slow for closed-loop control of transient HCCI combustion (as engine cycle typically completes in few milliseconds) [6]. A more sophisticated solution to this problem is fast thermal management (FTM) [7]. Fast thermal management can be used to control the temperature of the mixture at the

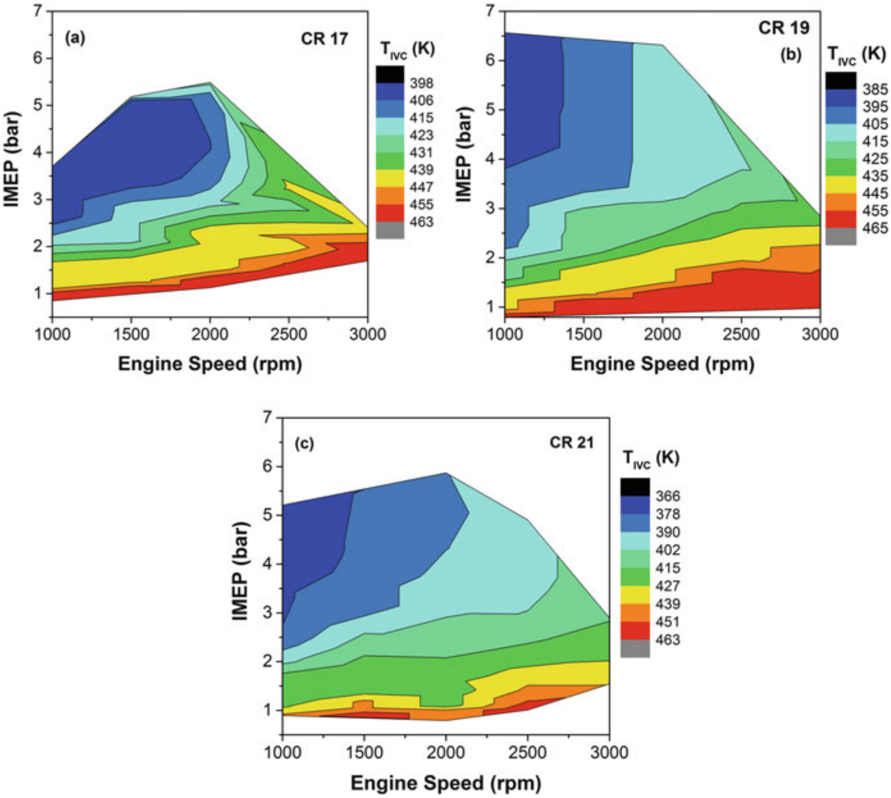

Fig. 5.1 Engine T_{IVC} (inlet temperature) as function of engine load and speed in ethanol HCCI combustion at different compression ratios [5]

beginning of compression stroke. The inlet temperature strongly affects the combustion phasing. It is demonstrated that thermal management could be used for HCCI control in a wide range of operating conditions [8]. Figure 5.2 shows the schematic of FTM system used for control of HCCI combustion. To achieve fast thermal management control of the inlet temperature, a source of cold ambient air and a source of hot air (heated electrically or by exhaust heat recovery) is used. By controlling the valves of the hot and cold airflows, cycle-to-cycle control of the inlet temperature can be achieved. The benefit of this method is that it does not require major engine modifications or use of fuel additives. The major drawback of FTM is that the engine system has to be fitted with a heater (electrical or exhaust heat exchanger) [10]. The mixing of hot can cold air can be inside or outside the cylinder (Fig. 5.2) by independently controlling the valves.

To avoid the electric heater for intake preheating in HCCI combustion, heat exchangers are used to extract the heat from engine exhaust. Figure 5.3 illustrates the system used for extracting heat from engine exhaust and coolant to preheat the intake air. At lower engine loads, the difference between exhaust gas temperature

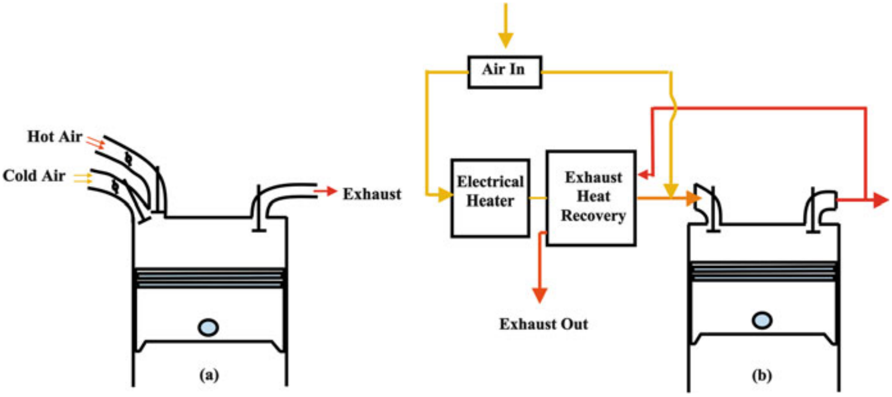

Fig. 5.2 (**a**) Schematic of heated and cold air arrangement in engine with two intake valve (Adapted from [9]), and (**b**) schematic of FTM system (Adapted from [7])

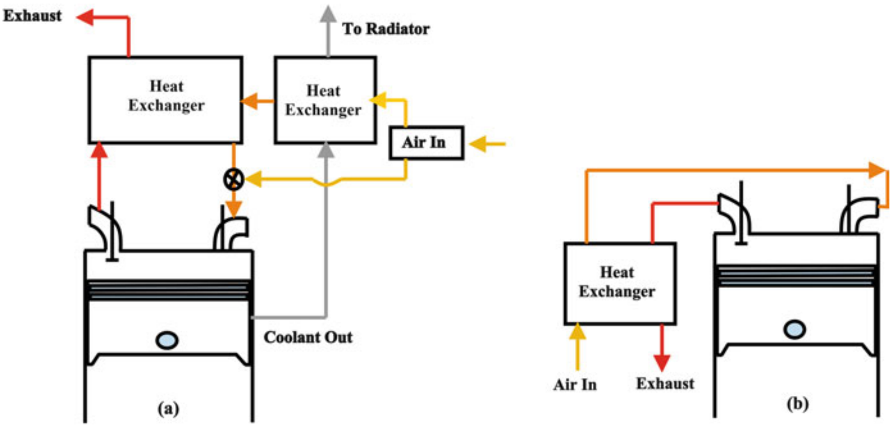

Fig. 5.3 Schematic of air preheating system by using heat exchanger taking heat input from (**a**) both exhaust and coolant (Adapted from [11]), and (**b**) exhaust (Adapted from [4])

and intake air temperature can become insufficient for intake preheating. Intake throttling is necessary to compensate for this temperature difference. At lower engine load, exhaust temperature is also lower, and typically, higher intake temperature requirement is also higher at low engine loads. Figure 5.4 shows the interaction between inlet air temperature, combustion phasing and exhaust gas temperature in HCCI combustion with exhaust heat recovery. Higher inlet temperature leads to advanced combustion phasing, which results in lower exhaust temperature. Lower exhaust temperature leads to the decrease in intake temperature resulting into delayed combustion timings. Delayed combustion timings then increase the exhaust temperatures (Fig. 5.4). The exhaust heat recovery-based system shown in Fig. 5.3 is not typically operated during engine start-up [4, 12]. In general idea of using heat exchanger may have difficulty in adjusting the intake

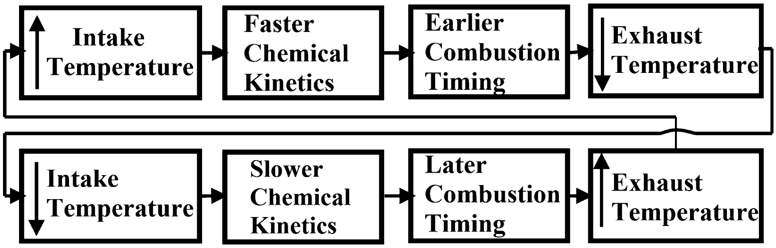

Fig. 5.4 Interaction between inlet air temperature, combustion phasing and exhaust gas temperature in HCCI combustion with exhaust heat recovery [12]

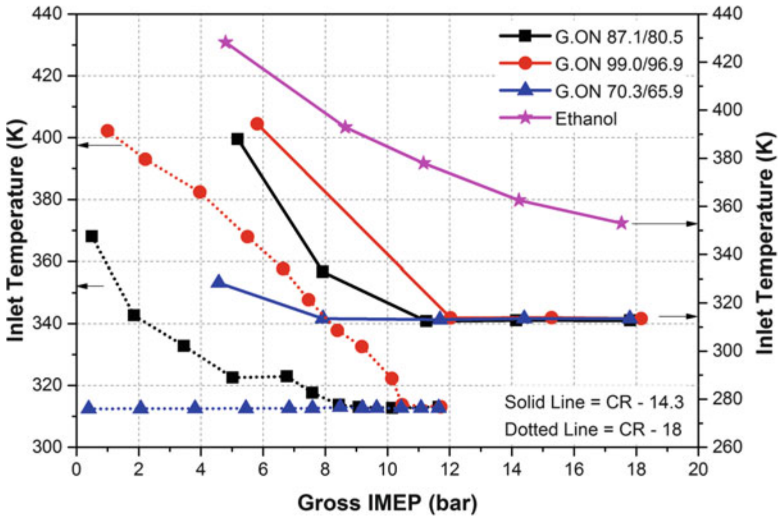

Fig. 5.5 Intake temperature as a function of engine load and fuel type in PPC engine for different compression ratios (Adapted from [13, 14])

temperature rapid enough to meet the control requirement during transient operation due to the thermal inertia of heat process. The FTM is presented as an acceptable alternative of exhaust heat recovery method.

Intake temperature requirement depends on the engine load as well as combustion strategy. Figure 5.5 shows the intake temperature requirements in PPC engine using different gasoline-like fuels at 1300 rpm for two compression ratio. Fuels designated as G.ON x/y represents gasoline octane number, where ‘x’ is RON, and ‘y’ is MON. Figure depicts that intake temperature requirements decrease with increase in engine loads. To achieve higher engine loads, typically higher intake pressure is employed. Gasoline becomes Φ sensitive at higher intake pressure (see Chap. 2), and stratification can be used to increase the load range. In PPC engine, heavy stratification of gasoline is used, and thus, lower intake temperature is required at higher engine load. At higher engine load (>10 bar IMEP), intake temperature requirement is close to ambient temperature (Fig. 5.5). The lower octane fuel is the relatively more reactive fuel and thus requires comparatively lower intake temperature for PPC combustion. Figure also depicts that at lower engine loads relatively higher intake temperature is required for all the fuels in PPC combustion.

The PPC engine operation at idle/lower load using higher octane gasoline is a major challenge. To achieve autoignition at lower load using high octane gasoline, higher intake temperature or boost (higher intake pressure) is required. Available boost pressure with standard turbocharger is limited at low engine load and speed conditions. Higher intake temperature requirement at lower engine load is also illustrated in Fig. 5.5. The usefulness of negative valve overlap (NVO) is investigated in PPC engine at lower engine load conditions [15]. The purpose of NVO is to trap hot residual gases in the cylinder for increasing the charge temperature to enable auto-ignition while using high octane gasoline. At lower engine load, exhaust temperature is also relatively lower, and thus, trapping residual gases of lower temperature results into relatively lower charge temperature. To increase the charge temperature for facilitating autoignition, higher amount of residual gases by larger NVO is required. Increased NVO also increases the EGR (internal) fraction in the cylinder. Higher amount of EGR leads to lower charge temperature by increasing specific heat of the charge [15]. Hence, there exists optimum NVO duration that can be used to operate the PPC engine at lower load condition.

Figure 5.6 shows the combustion instability (represented by standard deviation of IMEP) and ignition delay variation at different NVO operating conditions with and without the use of glow plug. Glow plug is used in combination with NVO, and glow plug is continuously on during the experiment. Figure 5.6 depicts that glow plug operation lowers standard deviation in IMEP (improves the combustion

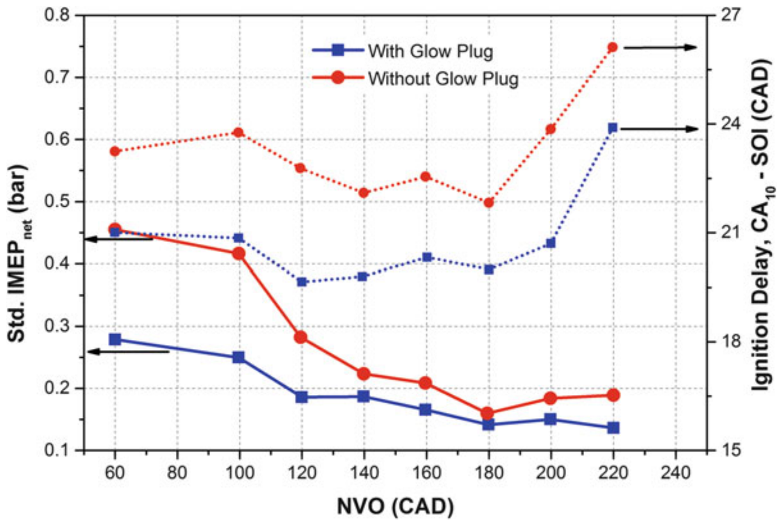

Fig. 5.6 Standard deviation of IMEP and ignition delay variations (with and without glow plug) in PPC engine using gasoline (Adapted from [15])

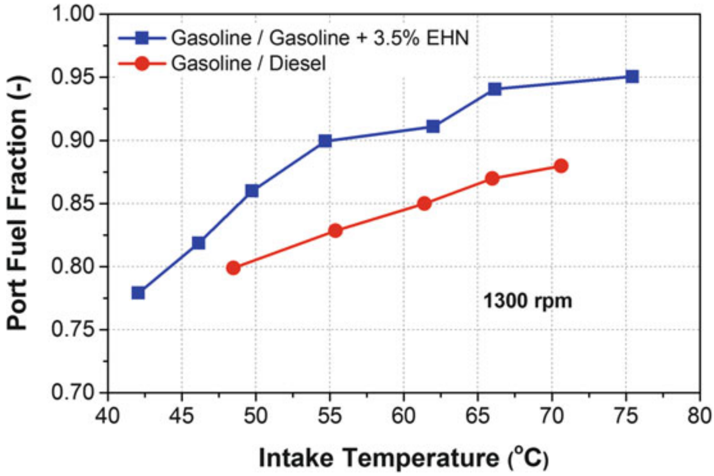

Fig. 5.7 Intake temperature requirement in RCCI engine with port fuel fraction (Adapted from [16])

instability) and reduces the ignition delay due to additional increase in charge temperature by glow plug. Study [15] showed that the lowest ignition delay is observed from approximately 120 CAD up to 200 CAD NVO (corresponding residual gas fraction is 20–45%). The NVO duration of 180–200 CAD (40–45% residual gas fraction) resulted into shortest combustion duration in PPC mode. This is the result of several factors such fuel, EGR and temperature distribution in addition to global charge temperature. Considering a limiting value of standard deviation on IMEP for minimum achievable load, the low load limit of PPC engine can be extended from 3.2 bar IMEP down to 2.2 bar IMEP by increasing NVO from 60 CAD to 180 CAD [59].

The RCCI combustion engine requires relatively lower intake temperature than other LTC strategies. In RCCI combustion, typically a high reactivity fuel is injected to enable the combustion in the chamber. The intake temperature requirement depends on the amount of low reactivity fuel injected in the cylinder. Figure 5.7 shows the variation of intake temperature with port injected fuel fraction in RCCI engine operation with gasoline/diesel and gasoline/gasoline +EHN. Intake temperature requirement increases with increase in the low reactivity fuel to enhance the autoignition. Similarly, intake temperature requirement is higher in single fuel RCCI with additives (EHN) as overall low reactivity fuel increases.

5.1.2 *Exhaust Gas Recirculation*

Exhaust gas recirculation (EGR) is a well-established method for facilitating control over ignition and combustion phasing, which affects the combustion performance in CI engines [17]. The EGR is broadly divided in two category (internal and external EGR) based on induction method in the cylinder. In external EGR (eEGR), a faction of exhaust gases from engine tailpipe is recirculated back into the intake manifold by typically passing through an EGR cooler. However, in case of internal EGR (iEGR), hot residual gases are trapped in the cylinder by changing the valve timings. The effect of EGR in HCCI engine is discussed in Sect. 2.2.4.5 of Chap. 2. The use of EGR changes the charge composition and charge mixture properties as the thermodynamic and chemical properties of EGR are significantly different than air. EGR typically consists of significant amount of complete combustion products (CO_2 and H_2O) and a large variety of incomplete combustion products, some of which is trace quantity. The EGR as a diluent has the potential to significantly alter and affect combustion characteristics including burn rates, ignition timing, combustion and engine thermal efficiencies. Effect of EGR and its constituents on ignition timing for gasoline and PRF80 fuel in HCCI combustion is shown in Fig. 2.22 (Chap. 2). Chemical effect of incomplete combustion production on the start of combustion in HCCI combustion is also shown in Fig. 6.29b (Chap. 6). The EGR is also used to increase the engine operating load range of HCCI combustion (Fig. 2.23 of Chap. 2).

In internal EGR, trapping of hot residuals from the previous cycle is achieved by the using flexible valve trains. Desired amount of internal EGR is obtained by adjusting the valve lift and valve timings (typically NVO) depending on engine operating conditions. In NVO strategy, the exhaust valve is closed early and a fraction of exhaust gas trapped in the cylinder undergo “recompression” due motion of piston towards TDC. The trapped residual gases subsequently mix with the cooler fresh incoming charge. Mixing of the fresh charge with the hot residual gases leads to overall increase in the charge temperature, which facilitate to achieve autoignition in HCCI combustion using high octane fuels [18]. In conventional engines, in contrast to NVO, the positive valve overlap (PVO) is typically used where both the exhaust and intake valves are open simultaneously for a short time period around the intake TDC position. Figure 5.8 illustrates the PVO and NVO valve lift profiles along with corresponding typical cylinder pressure trace. The PVO shown in figure has negligible overlap between intake and exhaust events, but it can vary depending on engine speed. Figure 5.8 also shows that advance in exhaust valve closing (EVC) timing is complemented by a corresponding and equal retard in intake valve open (IVO) timing in case of the NVO, and this strategy is called “symmetric” NVO. This strategy is employed to minimize the pumping losses related to recompression.

Another strategy for internal EGR is known as exhaust reinduction or rebreathing. In exhaust reinduction, fraction of exhaust gases from the previous engine cycle are inducted back into the cylinder during the intake stroke along with fresh air by a short exhaust valve event during the intake stroke. A study demonstrated the viability of reinduction HCCI and its potential for effective combustion phasing control [20]. The sensible energy of exhaust inducted in the cylinder is utilized to initiate the HCCI combustion. However, one of the main challenges with reinduction HCCI is the cylinder-to-cylinder coupling, which occurs by mixing of exhaust from different cylinders in the exhaust manifold and mixed exhaust is

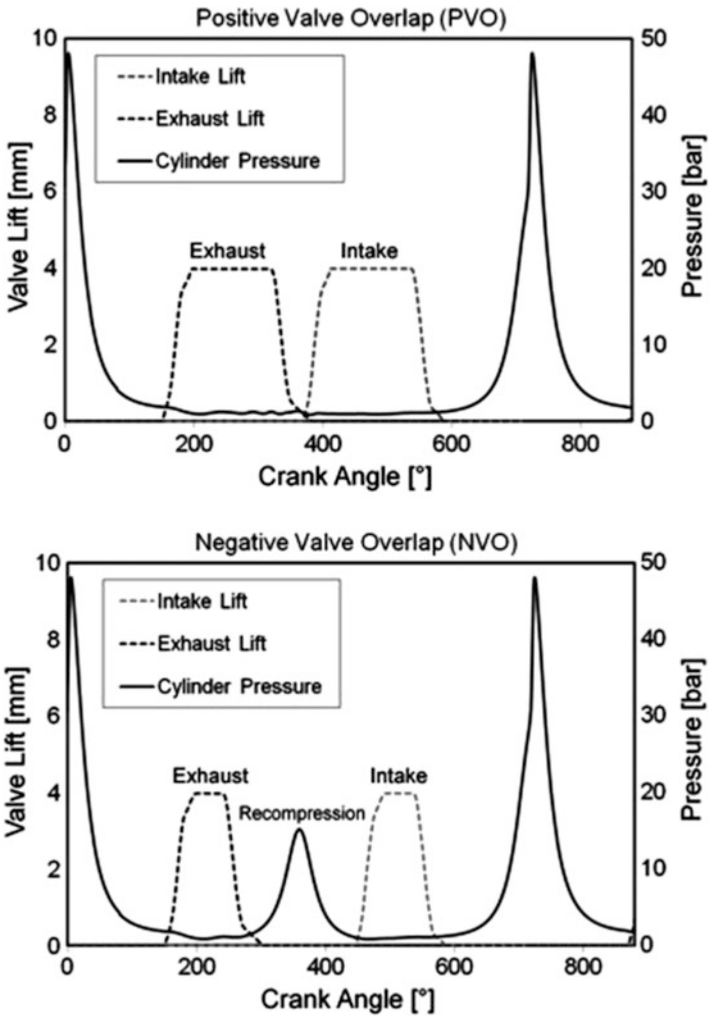

Fig. 5.8 Illustration of PVO and NVO valve lift profiles along with typical cylinder pressure curve [19]

reinducted into the cylinders. This mixing couples the contents from one cylinder to the contents of the other cylinders of the engine [21]. A model and control strategies are developed to handle the cylinder-to-cylinder coupling concerns accompanied with exhaust reinduction HCCI [22]. However, exhaust recompression (NVO strategy) does not have cylinder to cylinder coupling issues altogether.

The HCCI combustion is significantly affected by the homogeneity of EGR. The HCCI engine tends to tolerate more heterogeneously distributed EGR. For the same EGR level, power out can be increased by heterogeneous EGR as it lowers the pressure rise rate due to thermal stratification. Inhomogeneity among fresh charge and residual gas is favourable condition because temperature in EGR rich regions of

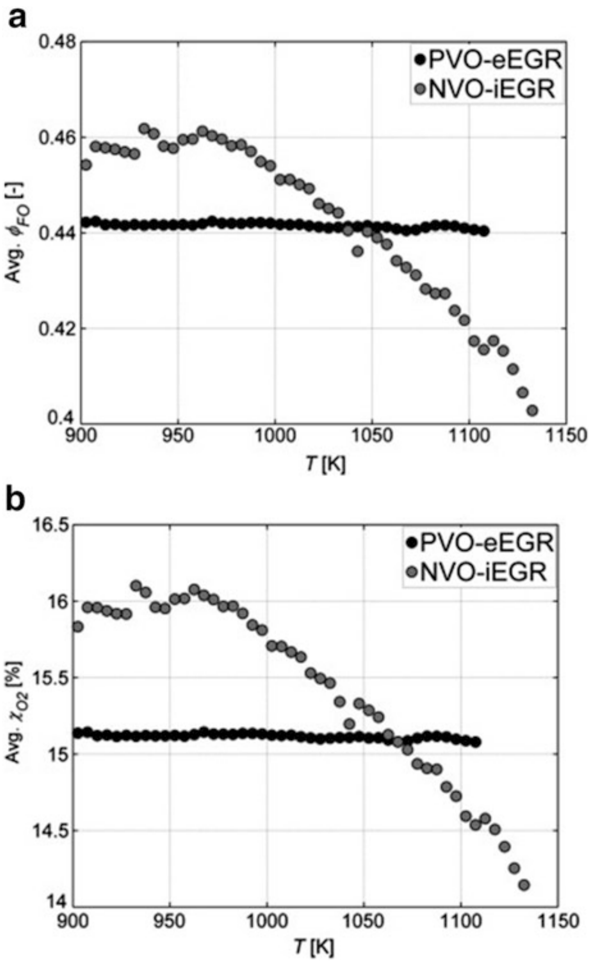

Fig. 5.9 Pre-ignition (10° CA bTDC) distribution of the average Φ_{FO} and χ_{o2} within a zone versus charge temperature (T) for PVO-eEGR and NVO-iEGR [19]

the chamber is higher than those attained in completely homogeneous in-cylinder charge. The small inhomogeneity could be beneficial in reducing heat release rate (see Sect. 2.2.4 of Chap. 2) [23].

Figure 5.9 illustrates the mean values of fuel oxygen equivalence ratio (Φ_{FO}) and oxygen mole fraction (χ_{o2}) as a function of charge temperature for understanding the relation between the thermal and compositional stratification for internal and external EGR conditions. This is determined by computing the average values of Φ_{FO} and χ_{o2} of all CFD (computational fluid dynamics) cells within a particular temperature range. The figure depicts that there is insignificant variation in composition (Φ_{FO} and χ_{o2}) with respect to temperature in the PVO case with external EGR. However, in case the of NVO internal EGR, both Φ_{FO} and χ_{o2} are negatively correlated to the stratification in temperature. The hotter zones have a lower Φ_{FO} and lower χ_{o2} than the colder regions, which have higher fuel to oxygen equivalence ratios and less residual dilution [19].

A study showed that the composition of the EGR gases is different from that of the engine exhaust gases due to an evolution inside the EGR pipe resulting from several mechanisms: fragmentation, conglomeration and chemical reactions for unburned hydrocarbons, oxygenates and NO_x [24]. A weak evolution of CO, CO_2 and O_2 concentration is observed in the EGR pipe. Conglomeration of some unburned hydrocarbons is found, which could not be interesting to control the combustion because PAHs are not very reactive and present in the EGR gases in very low concentration. Presence of aldehydes and ketones could impact HCCI combustion phasing because of their high reactivity [24].

EGR has thermal and chemical effect on HCCI combustion. However, EGR is used in HCCI combustion mainly for thermal effect to reduce the charge temperature in the cylinder. The EGR dilution also reduces the oxygen content in the charge, and at higher engine loads, oxygen deficiency limits can also be encountered (see Chap. 7). At higher engine loads, very high amount of EGR is required to control the heat release rate. A PCCI combustion system used 54% EGR to retard the combustion phasing and improve the obtained IMEP [25]. Another premixed charge combustion study used high EGR rates up to 68% to effectively control the start of combustion [26]. However, higher EGR amount decreases the thermodynamic efficiency, and it also creates problems in engine transient response and temperature-stability characteristics. Current research on LTC strategies are trying to reduce the quantity of EGR required to control the heat release rate at higher engine loads.

Figure 5.10 presents the EGR flow rate as a function of engine load and fuel type in PPC engine for different compression ratios. At higher compression ratio (18), EGR flow rate around 50% is required for higher engine load, and low octane fuel has lower requirement of EGR. At lower compression ratio, lower octane gasoline

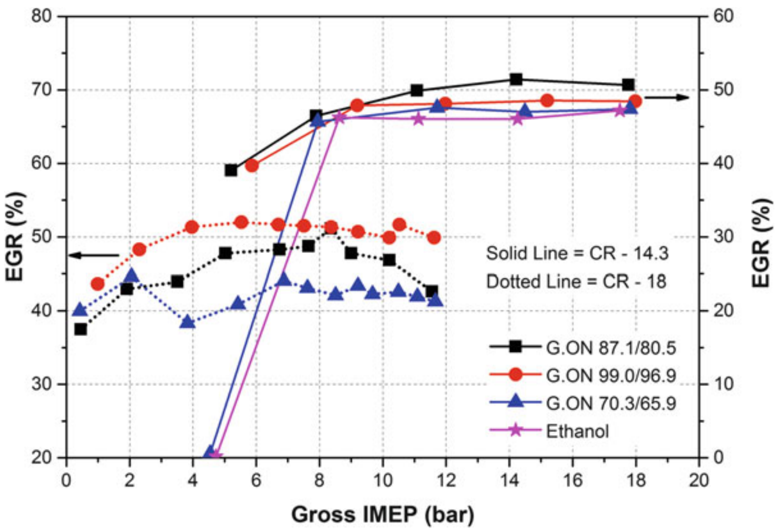

Fig. 5.10 EGR rate as a function of engine load and fuel type in PPC engine for different compression ratios (Adapted from [13, 14])

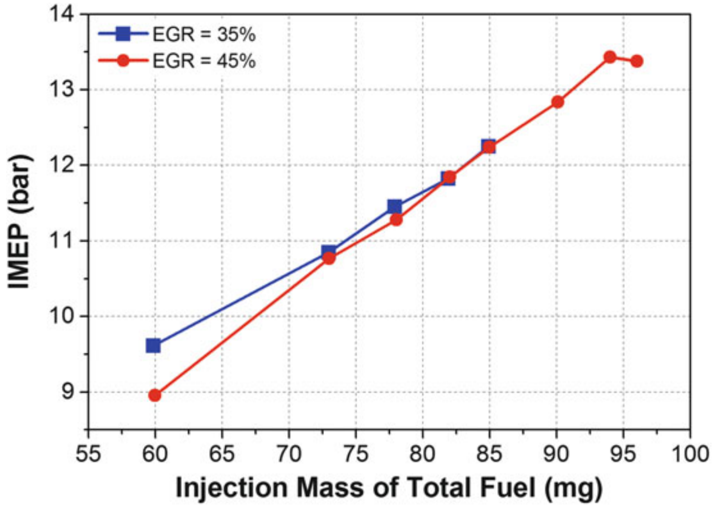

Fig. 5.11 IMEP as a function of direct injection fuel mass (at fixed premixed ratio 85%) in RCCI engine for two EGR flow rates (Adapted from [27])

and ethanol have lower EGR requirement for low load conditions. The EGR requirement depends on fuel, compression ratio, injection timings and intake pressure used for PPC combustion. The EGR rate shown in Fig. 5.10 is for injection strategy (two direct injection per cycle) developed in reference [13].

The RCCI combustion engine requires relatively lower EGR rate because the combustion rate is controlled by varying the reactivity of charge by using two fuels with significantly different reactivity. Figure 5.11 shows the IMEP achieved with two different EGR flow rates at fixed premixing ratio of 85%. Figure 5.11 depicts that more than 12 bar IMEP can be achieved with 35% EGR flow rate, while around 50% EGR rate is required in PPC combustion for the same IMEP (Fig. 5.10). Typically, in the optimized RCCI operation conditions require the EGR assistance, particularly at higher engine load to control the pressure rise rate, and up to 50% may be required depending on engine load and fuel.

5.1.3 Variable Valve Actuation

Variable valve actuation (VVA) offers a very fast method of varying the breathing process of the engine. In variable valve actuation strategy, mainly variable valve timing (VVT) and/or variable valve lift (VVL) is used to (i) control the residual gases in the cylinder and (ii) vary the effective compression ratio of the engine. Compression ratio control can be achieved over a wide range of HCCI operating conditions as the compression ratio strongly affects the combustion phasing. The HCCI engine has typically higher compression ratio and can obtain lower compression ratios by delaying the IVC during the compression stroke. For HCCI control purposes, two major methodologies, residual gas control and effective compression ratio control, are used very often [6]. A fully flexible VVA also offers the advantage of changing the temperature and composition of charge by rebreathing the residual gases from the previous cycle into the cylinder. With a full VVA system, the timing for the IVO, IVC, exhaust valve opening (EVO) and EVC can be changed to any desired timings [10].

Effect of different VVA strategies is investigated in premixed diesel combustion for their performance and cost/complexity [28]. Study compared several configurations such as VVTe (variable valve timing exhaust), VVTei (variable valve timing exhaust and intake), VVLei (variable valve lift exhaust and intake), DEe (double event of exhaust valves during intake phase) and SEe (single event of exhaust valve during intake phase) using VVA in premixed diesel combustion (Fig. 5.12). First three strategies (VVTe, VVTei and VVLei) are exhaust recompression strategy, and last two (DEe and SEe) are exhaust reinduction strategy to control the residual gases in the cylinder. In VVTe configuration, only the timing of exhaust valve is advanced while maintaining the same lift and duration, to trap the residual gases. In case of VVTei, valve timing is adjusted for both intake and exhaust valve symmetrically. In case of VVLei, IVC and EVO timings are kept fixed, while IVO and EVC timings are adjusted along with valve lifts. Reduction in effective compression ratio is avoided by keeping the fixed timing of IVC. Fixed EVO timing remove very early opening of the exhaust valve, which avoids the reduction in the expansion phase of the engine cycle. SEe and DEe are the two configurations used for reinduction of exhaust gases in

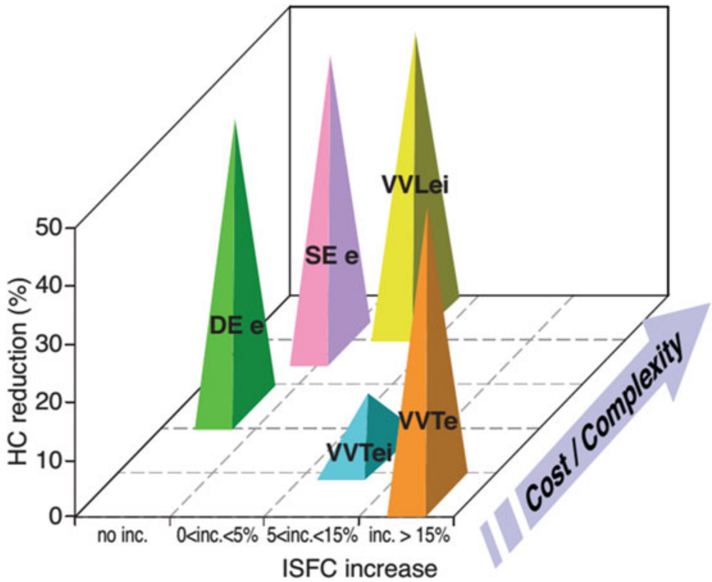

Fig. 5.12 Trade-off effectiveness/cost at very low load for different VVA strategies [28]

the cylinder by reopening of single and both (double) exhaust valves, respectively, during the intake stroke of the engine cycle [28].

Utilization of internal EGR reduces the CO and HC emissions at lower engine loads by increasing the charge temperature. Figure 5.12 compares different VVA configurations with respect to effectiveness in reducing HC emissions versus increase in indicated specific fuel consumption (ISFC) along with cost/complexity of the system with reference to fixed valve actuation (commercial configuration). The VVTe configuration is effective in decreasing HC emissions (50%) and relatively lower complexity, but it increases the ISFC (>15%) because of the early opening of exhaust valve leading to reduction in expansion work. The VVTei is slightly better in ISFC but relatively lower HC reduction in this strategy. In case of reinduction strategy, there is no reduction in the expansion phase. The double event of exhaust (DEe) during intake appears to provide the best trade-off between reduction in HC emission, ISFC increase and cost of the tested configuration [28]. A more complete discussion can be found in the original study.

Technological limitations of air path (compressor behaviour at the high-pressure ratio and low airflow, EGR rate vs. turbine energy, EGR cooling capacity and circuit permeability) and combustion limitation (torque reserve at high equivalence ratio, particulate emissions and combustion noise) limit the engine load range of highly premixed combustion [28]. Compression ratio of engine affects the charge temperature, and thus, NO_x and noise emissions are higher at the higher compression ratio. Reduction in compression ratio leads to increase in engine load range. The effective compression ratio can be reduced by varying the IVC timings because effective compression ratio of the engine is directly related to the cylinder volume at IVC. The compression affects only the cylinder volume at IVC and not the entire engine displacement volume. Figure 5.13 shows the summary of trade-off observed in reduction of effective compression ratio using VVA by presenting positive as well as negative impacts.

The reduction in effective compression ratio must have a positive effect on NOx emission due to decrease in pressure and temperature in the engine cylinder. However, a major effect is the decrease in volumetric efficiency resulting into decrease in EGR rate at particular intake pressure and equivalence ratio, which leads to higher NOx. Similar trade-off is observed for smoke emissions. Reduction in compression ratio increases the ignition delay thus more premixed combustion leads to reduction in smoke, while decrease in the volumetric efficiency produces opposite result. In case of HC and CO emissions, the combustion temperature increase leads to decrease in these emissions but the lower effective compression ratio also lower the charge temperature during injection phase leading to lower vaporization, which creates a negative effect.

Different types of VVA technologies are used in the engine such as cam phasing, piezoelectric, hydraulic, permanent magnet, electromagnetic and thier combinations [29]. A summary of the main advantages and disadvantages of different VVA technologies are provided in Table 5.1. A more complete discussion can be found in the original study [29].

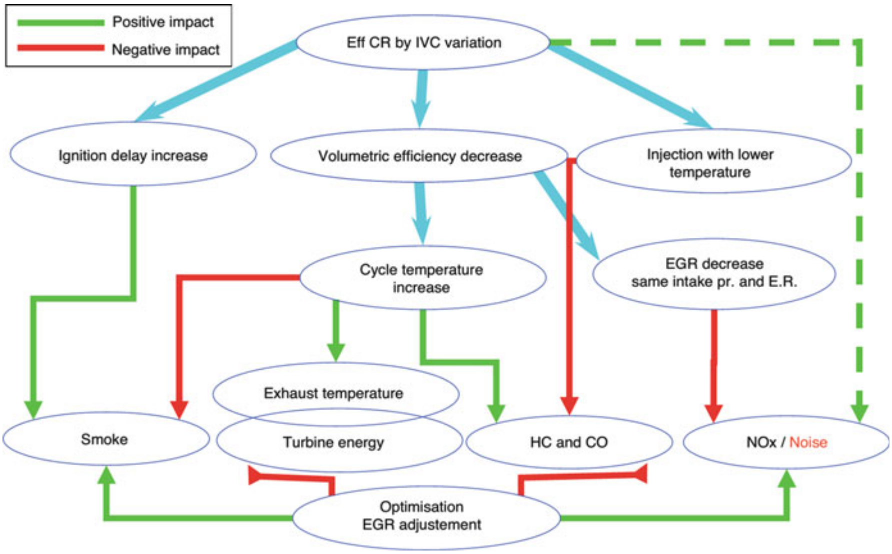

Fig. 5.13 Summary of trade-off observed in reduction of effective compression ratio using VVA strategy [28]

Table 5.1 Advantages and disadvantages of different VVA technologies [29]

Technology	Advantages	Disadvantages
Cam phasing	Reliability; variable valve lift	Cycle-by-cycle actuation only at low engine speed
Piezoelectric	Fully variable timing	Small displacements; complicated control
Hydraulic	Fully variable timing; soft landing; variable valve lift	Space requirements; temperature sensitivity; precise/expensive valves; high-power consumption
Permanent magnet and motors	Fully variable timing	Low force generation; weight; low valve speeds
Electromagnetic	Fully variable timing; space requirements	Additional electronics; soft landing needed; weight

5.1.4 Variable Compression Ratio

Compression ratio has very strong influence on intake temperature requirement for control of combustion timings in HCCI combustion engine. The variable compression ratio (VCR) can be used to control combustion timings by varying the compression ratio, which leads to variation in the charge temperature after compression. The fuel–air mixture auto-ignites at early crank angle position at a higher compression ratio due to higher charge temperature. VCR can be achieved by several methods. One method is to tilt the upper part of the engine block, the mono head. A hydraulic motor, controlled by an electronic valve turns the eccentric shaft that changes the engine configuration to vary the compression ratio [30]. Another method is to mount a plunger in the cylinder head whose position can be varied to change the compression ratio [31]. The main drawback of VCR system is that it presently does not allow individual cylinder control that is necessary to obtain good combustion timing control. The VCR systems are also expensive and complex [10].

The compression ratio is a key structural parameter that affects the engine efficiency. The LTC engine has typically higher efficiency, and relatively lower compression ratio is used to increase the load range of LTC engine (see Figs. 5.5 and 5.10). In order to extend the engine operating load, a RCCI engine is optimized at a very low compression ratio of 11.7 [32]. Another RCCI combustion study investigated two different compression ratios of 14 and 17 at different engine speeds [33]. At lower compression ratio, NOx emission reduced and unburned hydrocarbon emissions are increased in RCCI engine. At compression ratio 14, longer ignition delay, combustion duration and also lower maximum HRR are achieved. Therefore, compression ratio 14 is preferred at high load due to the limitation on peak pressure rise rate of the engine [33, 34]. A study also investigated the RCCI combustion by varying effective compression ratio by changing IVC timings [35]. The shape of heat release curve varies with IVC timings. The IVC timings can be optimized depending on fuel injection timings and fuel premixing ratio at particular engine speed and load conditions.

5.1.5 Water Injection

HCCI combustion control has been demonstrated using water injection. Water injection in the port is investigated in HCCI engine, and study indicated that evaporative cooling in the intake manifold (particularly at higher intake temperatures) offers the control of ignition timings by controlling the intake valve closing temperature [36]. The study confirmed that it is possible to control the ignition timing in a range of engine operating conditions with the use of water injection but at price of an increase in the already higher emissions of unburned hydrocarbons and carbon monoxide. Atomization, vaporization and distribution of injected water are the key parameters for system optimization [37].

A study used water injection to control diesel HCCI combustion timing and demonstrated that water injection could reduce the heat release rate significantly [38]. Direct water injection (using diesel fuel injector) is used to extend the load range of premixed compression ignition [39]. The study showed that the combustion suppression effect increases with increase in water injection quantity, and excessive water injection leads to higher THC emission along with lower thermal efficiency. Although the reaction suppressing effect of water injection enhances with advanced timings, an optimum water injection timing occurs. With more advanced timings, water vaporization is delayed, which decreases the reaction suppressing effect. Thus, amount of water injection must be limited to the minimum required for sufficient suppression of oxidation reactions at excessively advanced combustion timings [23].

A new strategy called thermally stratified compression ignition (TSCI) is proposed to artificially create the thermal stratification in the cylinder. In this combustion mode, water is directly injected in the cylinder to control the mean temperature along with temperature distribution in the cylinder, which offers cycle-to-cycle control of start of the combustion and HRR in premixed compression ignition [40]. More detailed description of TSCI strategy is provided in Sect. 2.4 of Chap. 2.

5.1.6 Boosting

Boosting (increasing intake air pressure) is considered as an effective technique to increase the engine operating load and extend the operational range of air–fuel ratio in HCCI combustion mode. However, boosting is accompanied by a higher peak cylinder pressure, which may lead to reach peak pressure limit of the engine. Higher intake pressure can enable an increase of the overall dilution of fuel–air mixture and affect the combustion characteristics through chemical effects related to the higher pressure of combustion chamber. At particular fuelling rate, relatively larger amount of inducted mass at higher intake pressure leads to higher level of charge dilution leads to reduction in pressure rise rates in HCCI combustion. For a fixed pressure rise rate limit, relatively higher amount of fuel can be burned in the cylinder and thus the maximum load limit increases via intake boosting. The higher load limit achieved in different LTC strategies is discussed in Sect. 7.1.2 of Chap. 7.

A very higher intake pressure (up to 3.6 bar) is required to achieve HCCI engine operating load (IMEP ~ 20 bar) comparable to conventional engines (Fig. 7.6 of Chap. 7). Figure 5.14 illustrates the intake pressure requirement as a function of engine load in PPC strategy at two different compression ratios. The figure depicts that very high intake pressure (~3.6 bar) is required at higher engine load. This level of intake boost is not practical with current turbochargers installed on marketed engines. Thus, boost requirement needs to be reduced by further research. Figure 5.14 also demonstrates that at relatively lower compression ratio, higher engine load can be achieved in PPC engine.

Turbocharger/supercharger providing the intake boost pressure needs to be adequately sized and matched to the requirements of LTC engine to achieve particular engine load and efficiency. Recent studies [41, 42] demonstrated the partial fuel stratification method to reduce the boost pressure requirement in low temperature gasoline combustion (LTGC) engines. Different direct injection strategies can be utilized to create the fuel stratification, which reduces the boost to load ratio in LTC engines [41, 42].

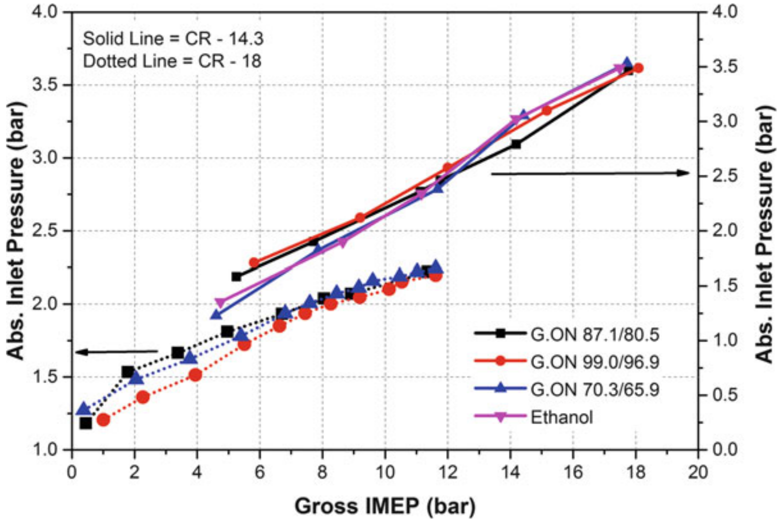

Fig. 5.14 Absolute inlet pressure as a function of engine load and fuel type in PPC engine for different compression ratios (Adapted from [13, 14])

5.1.7 In-Cylinder Injection Strategies

The local charge temperature and the air–fuel ratio are the key parameters controlling the initiation of HCCI combustion process. The direct injection of fuel into the cylinder provides the potential to control the combustion process by altering the local fuel concentration by varying the injection timings. The gas temperature is also altered through the charge cooling from fuel evaporation. Early fuel injection timings provide sufficient time for fuel vaporization and mixing with the air to create a premixed fuel–air mixture. The late pilot injection in the cylinder during compression stroke can control the combustion phasing by creating fuel stratification (increasing the local fuel concentration in some regions) in the cylinder [23]. Additionally, rich mixture decreases the ratio of specific heat and thus amount of disposable compression heating in the charge. Consequently mixture has to be further compressed to reach the autoignition temperature. The capability of split injection can combine these two functions.

Figure 5.15 illustrates the consecutive flame images of the gasoline auto-ignition by single and double injections (at three pilot injection timings of −20, −16 and −12 aTDC) with fixed main injection timing at TDC and fuel injection pressure at 400 bar, intake pressure 1.4 bar at 1200 rpm. Heat release rate and the normalized spatially integrated natural luminosity (SINL) are shown to investigate the correlation between the two (Fig. 5.15b). The natural luminosity of a flame generally consists of the chemiluminescence (emitted from the electronically excited gaseous species including CH, C_2 and formaldehyde) and the soot incandescence [43]. Figure 5.15 depicts that the single injection gasoline compression ignition (GCI) has a

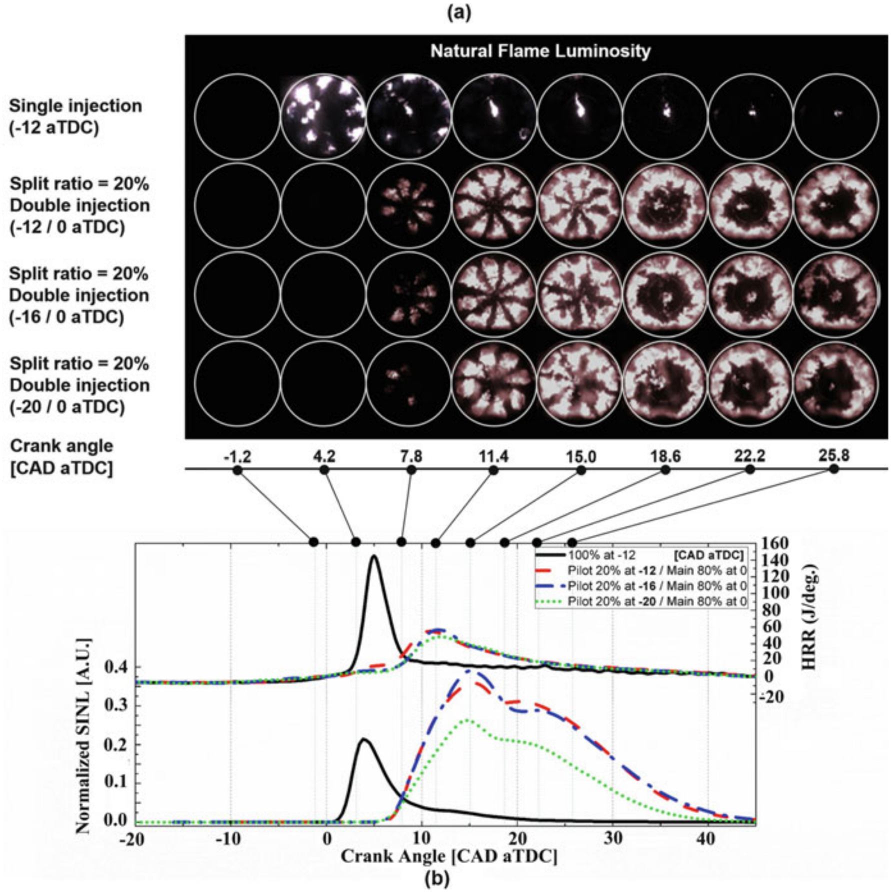

Fig. 5.15 Natural flame luminosity and heat release rate in gasoline combustion with single and double injection at 1200 rpm with total fuel injection quantity of 20 mg/stroke [43]

weak natural luminosity by the chemiluminescence from the premixed flame, while the double injection GCIs showed a strong natural luminosity similar to the diesel-like diffusion flame of the main injection by the soot incandescence. Double fuel injection creates a relatively more stratified charge in the cylinder leading to higher soot formation. Figure 5.15 also depicts that double injection strategy reduces the heat release rate of GCI combustion.

Fuel injection timings and number of injection govern the fuel stratification in the combustion chamber and thus heat release rate in LTC combustion. In case of diesel-like fuels, typically direct fuel injection strategy is used to achieve premixed diesel combustion. However, both port and direct fuel injection strategies are used in gasoline premixed compression ignition engines. Detailed charge preparation strategies using gasoline and diesel are discussed in Chap. 4. Comparison of fuel injection strategies in different LTC modes is provided in Figs. 4.12 and 4.20 (Chap. 4).

5.2 Altering Fuel Reactivity

Altering reactivity (autoignition characteristics) of fuel–air mixture is another strategy to control the combustion rate and combustion phasing in LTC engines. Mixture autoignition properties are affected by fuel types, fuel concentration, blending of two or more fuels, fuel additives, residual rate and residual reactivity, boosting etc. In this section, LTC controlled by altering the fuel reactivity using different strategies is presented.

5.2.1 Fuel–Air Equivalence Ratio

The amount of fuel burned in the cylinder mainly control by the power required from HCCI engine. In HCCI engine, throttle is not used to control the inducted air quantity therefore changing the quantity of fuel at different engine loads leads to change in equivalence ratio. Influence of equivalence ratio on HCCI combustion is discussed in Sect. 2.2.4.1 of Chap. 2. Higher equivalence ratio (richer mixture) advances the combustion phasing at fixed inlet conditions. Overall fuel–air equivalence ratio (Φ) of inducted charge governs the engine load in LTC engines. The Φ-sensitivity (autoignition reactivity) of charge also depends on intake boost pressure of engine (see Fig. 2.15 of Chap. 2). For example, gasoline is less sensitive fuel at naturally aspirated conditions, and fuel reactivity changes at higher intake pressure. With Φ-sensitive fuels, stratification of fuel can be used to increase the engine operating load for fixed pressure rise rate limits. Several LTC strategies such as PFS HCCI, PPC and RCCI uses fuel stratification in the cylinder by using direct fuel injection. Internal charge preparation in these LTC strategies is described in Sect. 4.2 of Chap. 4.

5.2.2 In-Cylinder Fuel Stratification

Fuel stratification in the cylinder is created by direct injection of fuel. Fuel stratification is one of the main methods to control the combustion rate by increasing the combustion duration. Fuel stratification is typically used to increase the engine operating load of LTC engines (see Sect. 7.1.2 of Chap. 7). A recent study demonstrated the direct injection of gasoline (by dual pulse) increases the thermal efficiency of partially stratified HCCI combustion [42]. PPC combustion also creates heavy stratification in the cylinder by using fuel injection system of diesel engine for gasoline-like fuels. In RCCI combustion, fuel stratification as well as reactivity stratification is created in the engine cylinder. Fuel injection strategy is an important variable among many factors in improving the performance of LTC engines. In RCCI combustion, injection strategy (number of pulse and fuel injection timing) affects the fuel and reactivity stratification in the cylinder. The number of fuel injection pulses in the injection strategy can affect the high reactivity fuel distribution and consequently shape the reactivity distribution in the combustion chamber. Single, double and triple fuel injection pulses have been investigated in different RCCI studies [34]. Fuel injection timing of high reactivity fuel is most important variable which affects the mixing process of fuel spray with mixture of air and low reactivity fuels. This mixing defines the reactivity distribution in the cylinder which affects the combustion characteristics in RCCI engine. Early injection of high reactivity fuel leads the combustion more prone to reactivity controlled, and relatively late injection results in combustion more similar to mixing/diffusion controlled. In direct injection dual fuel stratification (DDFS) strategy, both the high and low reactivity fuels are directly injected into the cylinder to create the stratified charge in the cylinder. Fuel injection strategies of different LTC modes with different stratification levels are presented in Fig. 4.20 (Chap. 4).

5.2.3 *Dual Fuel*

Fuel autoignition property variation can be used as a method for HCCI combustion control. The combustion phasing can be controlled by varying the fuel properties, and thus operating range can also be expanded. A study investigated the effect of fuel properties on low and high temperature heat release of HCCI combustion and its performance [44]. Fuel composition affects the low temperature heat release (LTHR) and high temperature heat release (HTHR) values. The effect of LTHR on HCCI combustion is discussed in Chap. 2. Fuel stratification in the cylinder leads to retarded combustion phasing, and it provides an additional actuator for HCCI combustion control [45]. However, very large stratification may lead to unstable combustion.

Dual fuel method can be used to actively vary the fuel octane number. The idea of using dual fuels is to use two fuels with different auto-ignition reactivity. The dual fuel system has a main fuel with a high octane number and a secondary fuel with low octane number (high cetane number) [46, 47]. This feature can then be used to control the combustion phasing in HCCI as blending the two fuels at different fuel ratio changes the auto ignition properties. Injection of low octane fuel in higher quantity leads to earlier autoignition in the cylinder. For advanced combustion phasing, the secondary fuel quantity is increased. For individual cylinder control of the combustion phasing, each cylinder must have two injectors for two different reactivity fuels. The benefit with dual fuels operation is that it provides an accurate control without any large engine modifications and cost. Only additional injectors are required. One demerit of this system is the requirement of two fuel tanks and its refuelling. However, secondary fuel consumption is very low and could be refuelled only in the maintenance intervals [10]. Typically, this system is demonstrated with fuel injection in the port by external blending of the fuels.

The RCCI combustion is another strategy, where in-cylinder blending of two different reactivity fuels occurs. With in-cylinder blending of fuels, fuel stratification as well as reactivity stratification can be controlled in this strategy. Fuel ratio of low and high reactivity fuel is one of the variables that affect the reactivity stratification in the combustion chamber. Reactivity in the cylinder can be varied by changing the fuel ratio, and consequently it affects the ignition delay. Ignition delay affects the mixing time of fuel and thus homogeneity of the charge in the cylinder. The low reactivity fuel ratio up to 90% is used in RCCI engines, and most of the studies showed that combustion phasing is retarded with the increase of low reactivity fuel. This is because the blending of low reactivity fuel increases the ignition delay and consequently retarding the combustion phasing [34].

5.2.4 Fuel Additives

Selection of appropriate fuel is a key aspect of LTC engine design and development. Autoignition characteristics and fuel volatility are the key variables in the fuel selection. In order to easily create premixed charge, fuel needs to have high volatility characteristics. Chemically, single-stage heat release fuels have lower sensitivity to variations in engine load and speed. Lower sensitivity to variations makes the requirements on HCCI control system easier over a wide range of operating conditions. To achieve high fuel conversion efficiency, fuel autoignition temperature is critical parameter for selection of an optimum geometrical compression ratio of the engine [23]. Required fuel in HCCI combustion needs to satisfy the requirements over wide range of engine loads (low to high load). At lower engine loads, low octane fuel is required to facilitate the autoignition, while at higher engine loads relatively higher octane fuel is required to avoid the knocking during combustion. This opposing requirement leads to difficulty in HCCI engine development, and there exists no universal fuel that satisfies the specific requirement of HCCI combustion engine. The optimal fuel selection is influenced by the combustion control strategies used in LTC engine and engine operating conditions. Thus, LTC engines can be operated on any fuel, but adaptation is required either fuel to specific engine design or engine to a specific fuel or even a specific engine operating condition [23].

Autoignition in HCCI combustion is mainly controlled by chemical kinetics, which is sensitive to several parameters including fuel properties, oxygen and EGR concentration, charge temperature and inlet pressure. In naturally aspirated conditions, even though HCCI combustion can be achieved using regular gasoline, a high intake temperature or high amount of hot residual gas is required for autoignition because of low autoignition reactivity of gasoline [1, 48, 49]. Due to high intake temperature requirements, charge densities are much lower, which limits the high load HCCI engine operation. This problem can be significantly reduced by using low octane gasoline in HCCI engine [50]. Furthermore, the combustion phasing can be retarded farther with good stability for a more reactive fuel, which allows higher charge mass fuel–air equivalence ratio (ϕ_m) operation without combustion knock. This leads to significant increase in higher engine load limit of HCCI combustion. Even though low octane gasoline is more suitable for naturally aspirated HCCI engines, these fuels are not readily available in the market. Moreover, the high reactivity of low octane gasoline can turn into a difficulty for intake-boosted operation due to very high reactivity above particular boost level. To avoid the knocking in this condition, large amount of EGR is required, and maximum load can become limited by a lack of oxygen at higher engine load operation [50]. To meet the fuel requirements in HCCI combustion, several fuel additives are added to regular low reactivity fuels. Fuel additives used in HCCI combustion include 2-ethylhexyl nitrate (EHN), di-tertiary butyl peroxide (DTBP), hydrogen (H_2), hydrogen peroxide (H_2O_2), formaldehyde (CH_2O) and ozone (O_3) [49, 51–56]. A study showed that hydrogen addition in DME (dimethyl ether) fuelled HCCI combustion could mitigates low temperature reactions and delay its occurrence by consuming OH radicals. Induction of hydrogen intensifies high temperature reaction of DME, and oxidation reaction process becomes fast due to the active nature of hydrogen in high temperature oxidation. Due to reduced low temperature reactions, high temperature reactions of DME is also retarded by hydrogen addition [53].

The EHN and DTBP are typically used to improve the cetane number (CN) of diesel fuels. These additives are also used to enhance the autoignition of gasoline-like fuels in HCCI combustion [49, 51]. Figure 5.16 illustrates the effect of addition of EHN and DTBP on intake temperature requirement in HCCI combustion engine. Figure 5.16a depicts the intake temperature requirement decreases with increase in concentration of additives to maintain same start of combustion (CA_{10}) position. Typically, high intake temperature ($T_{in} \sim 140$ °C) is required for fuels with low HCCI reactivity, such as regular gasoline and E10 at naturally aspirated engine operating conditions. By adding small amount of EHN (0.15%) or DTBP (0.35%), the tin drops drastically from 144 °C to 95 °C, and it can be further reduced to 60 °C using 0.4% EHN (Fig. 5.16a). The figure also shows that, DTBP has a smaller effect on T_{in} compared with EHN for a particular addition percentage. Furthermore, a nonlinear decrease in T_{in} is found for both EHN and DTBP that means the decrease of T_{in} is greater at low additive concentration (<0.25%) and the effect becomes more moderate as the additive percentage increases [49]. At higher intake pressure reactivity is higher at even lower addition percentage. Figure 5.16b illustrates the dependence of combustion (CA_{50}) on intake temperature for different concentrations of DTBP in 25% DEE-ethanol blends and ethanol. Figure 5.16b depicts that intake temperature requirement reduces at given combustion phasing with increase in DTBP concentration.

Figure 5.17a illustrates the effect of EHN on heat release rate (HRR) at low and intermediate temperature range at constant CA_{10} (366° CA) position. The HRR curves have been normalized by the total amount of detected heat release (THR) in order to eliminate differences between HRR curve caused solely by a difference in the total amount of fuel injected per cycle as T_{in} is varied. The figure depicts that low temperature heat release is significantly enhanced by increased additive

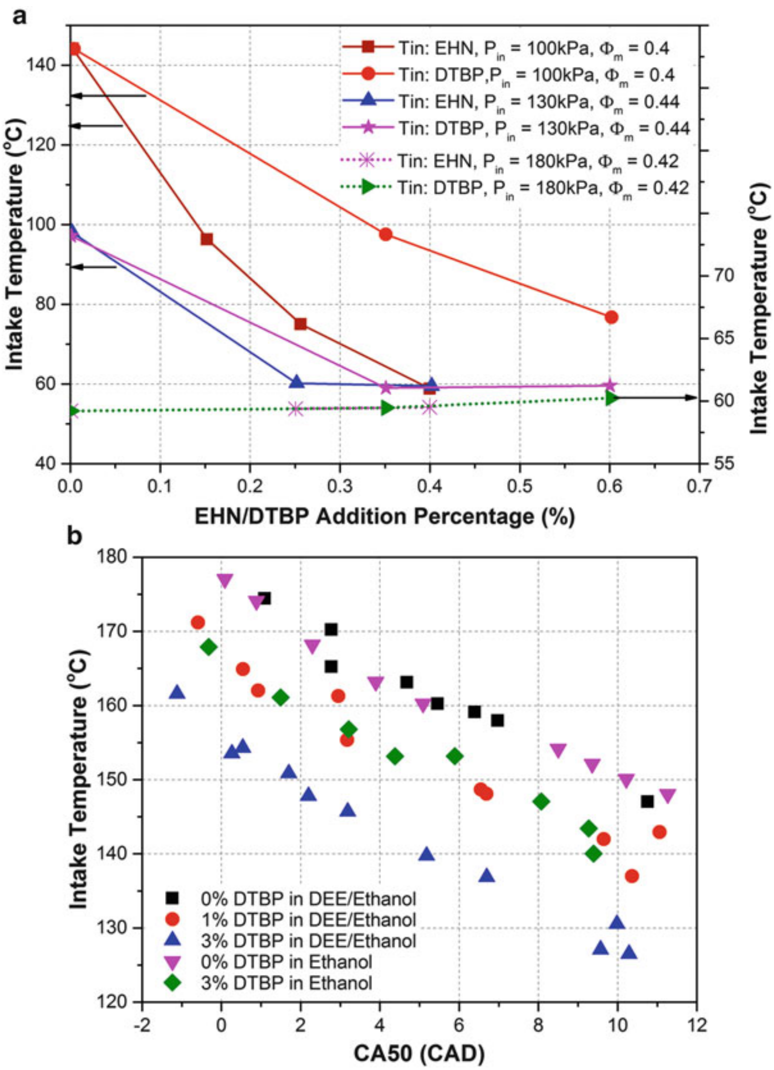

Fig. 5.16 (**a**) Intake temperature requirement as function of additive concentration at different intake pressure in HCCI combustion (Adapted from [49]), and (**b**) effect of DTBP addition to ethanol and ethanol-DEE blends on dependence between intake temperature and combustion phasing in HCCI combustion (Adapted from [51])

concentration, which leads to reduction in intake temperature requirements. Figure 5.17b shows the ϕ-sensitivity of E10 with 0.4% EHN additive. ϕ-sensitivity is an important parameter in applying partial fuel stratification because it allows the autoignition to occur sequentially down the equivalence ratio gradient, which reduces the HRR by increasing the combustion duration [49]. Figure 5.17b shows that addition of EHN significantly increases the ϕ-sensitivity of E10 at even lower

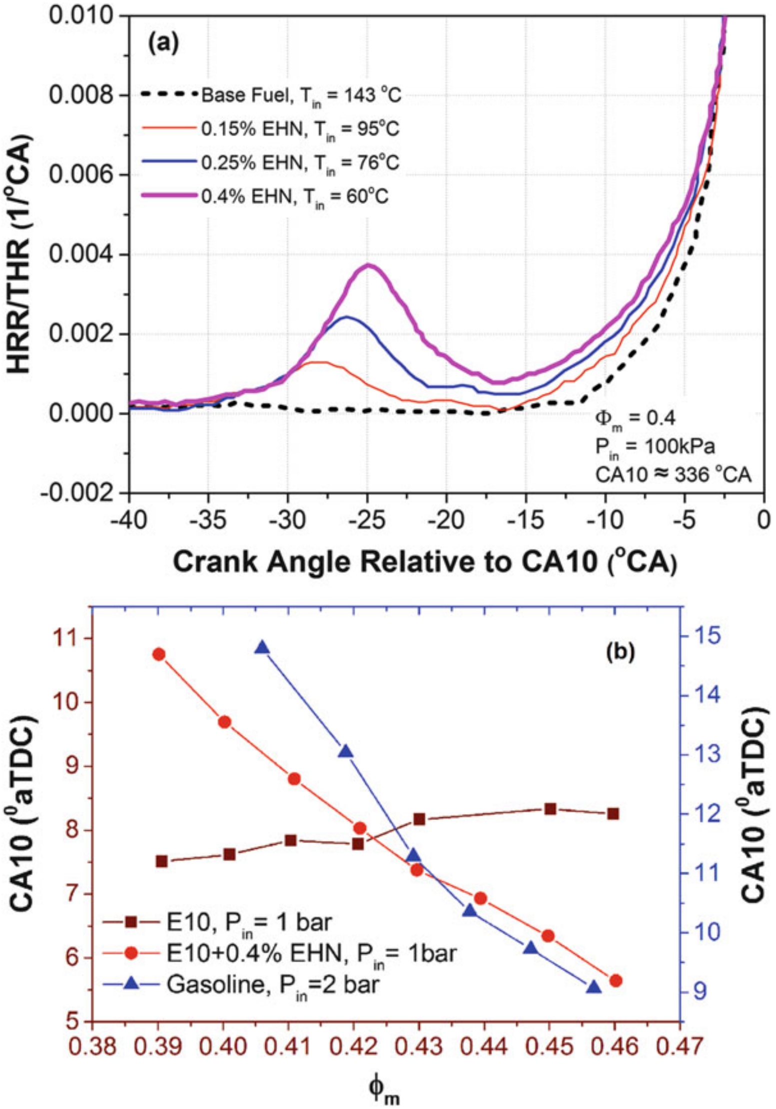

Fig. 5.17 (**a**) Effect of EHN on heat release rate at low and intermediate temperature range, and (**b**) ϕ-sensitivity of CCG-E10 with 0.4% EHN additive in HCCI engine (Adapted from [49, 57])

intake pressure, where E10 is not φ-sensitive without additives. Addition of EHN produces the effect similar to ϕ-sensitivity of gasoline at 2.0 bar intake pressure. It is expected that partial fuel stratification will work well for EHN additive added fuels to improve the thermal efficiency of the engine. Increasing the intake boost pressure also enhances the autoignition. Gasoline having additives can become very reactive as boost level is increased, requiring a large amount of EGR to prevent

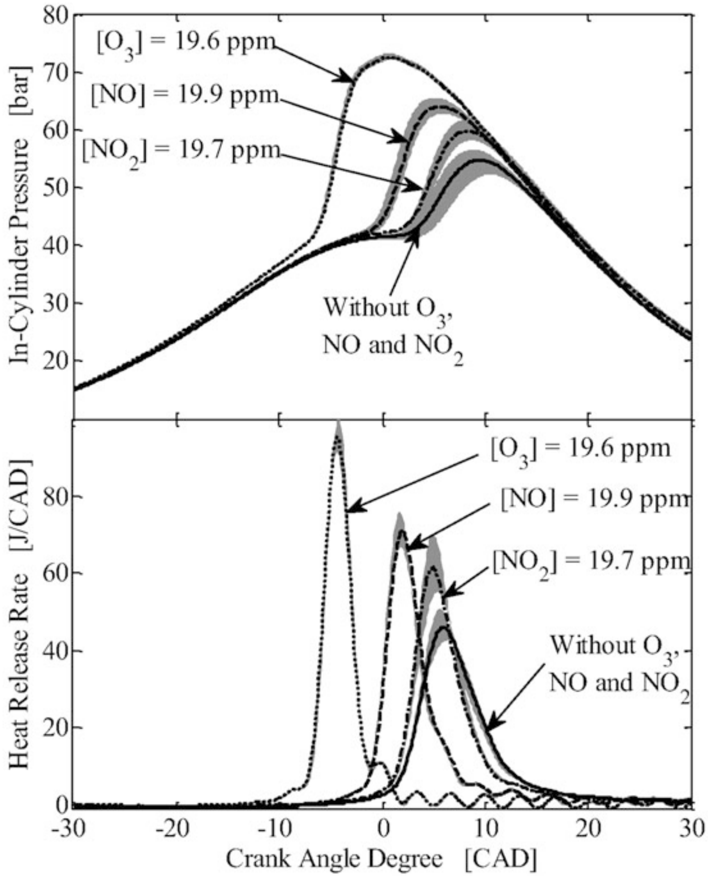

Fig. 5.18 Cylinder pressure and HRR variations with 20 ppm injection of NO, NO_2 and O_3 species in HCCI combustion [58]

overly advanced autoignition timings. These high EGR levels reduce the amount of oxygen in the charge, and the maximum load can become limited by oxygen availability before it reaches the knock/stability limit [49].

The reactivity of the charge can be modified by injecting oxidizing species such as nitric oxide (NO), nitrogen oxide (NO_2) and ozone (O_3) in the intake manifold. Figure 5.18 shows the cylinder pressure and HRR variations by injecting 20 ppm of NO, NO_2 and O_3 species in HCCI combustion. NO and NO_2 are the species that can even present in EGR operated HCCI combustion. These species can advance the combustion phasing of HCCI combustion (see Fig. 6.29a in Chap. 6). Figure 5.18 also depicts that all NO, NO_2 and O_3 species advances the combustion phasing of HCCI combustion and ozone is the most potent oxidizer.

A new strategy to control HCCI combustion is evolved, which is dominated by chemical kinetics, use oxidizing chemical species to initiate fuel oxidation. Ozone

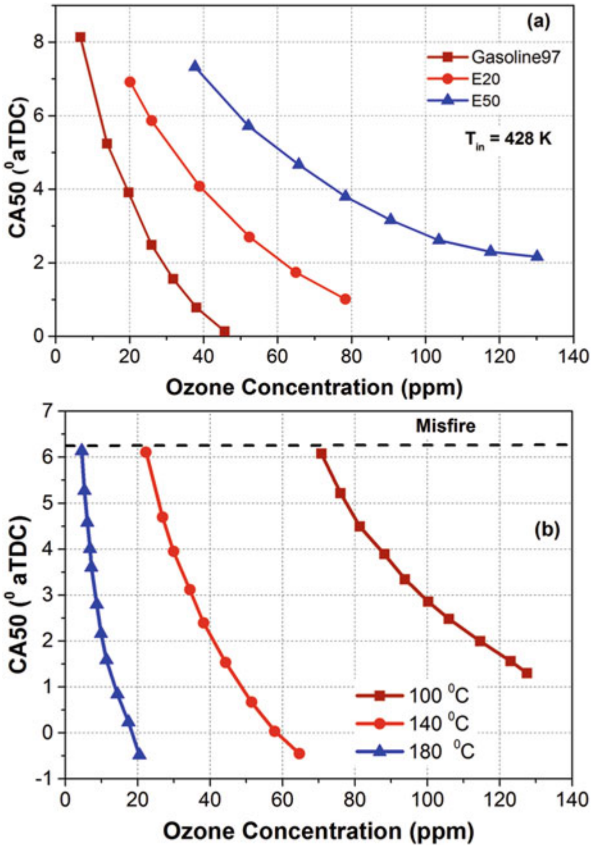

Fig. 5.19 Combustion phasing as function of intake ozone concentration for (**a**) different fuels (Adapted from [55]), and (**b**) different intake temperature using iso-octane (Adapted from [56]) in HCCI combustion

seems to be one of the most promising species. Present development of increasingly smaller ozonizers may lead to their installation on vehicles. This kind of ozonizer in HCCI engine can be used to control combustion phasing and promote HCCI engine application in future vehicles similar to conventional engines [56]. It is proposed to use ozone as oxidizer to modify the molecular structure of the fuel and to control the combustion timing inside the cylinder or to seed the intake of engines with some few ppm of ozone in order to enhance HCCI combustion [56, 59].

Figure 5.19 shows the combustion phasing as function of intake ozone concentration for different fuels and different intake temperature using iso-octane in HCCI combustion. The figure depicts that combustion phasing is advanced with increase of ozone concentration for all studied fuels and intake temperatures. Ethanol blends require higher ozone concentration than gasoline to maintain the same combustion phasing due to lower reactivity (higher octane number) of ethanol. The slope of the curves can be interpreted as the ozone seeding sensitivity, and a steeper slope corresponds to a higher sensitivity. Late combustion phasing at lower intake temperatures and lower ozone concentrations is limited around 7° aTDC or by unstable combustion or misfire (Fig. 5.19).

Figure 5.19b depicts that at 180 °C inlet temperature, less than 20 ppm of ozone is sufficient to vary the CA_{50} by 6 CAD and for the same combustion phasing variation around 120 ppm of ozone is required at 100 °C intake temperature. The decrease in the inlet temperature slows down ozone decomposition because the decomposition of ozone is strongly affected by temperature. Therefore, there are fewer O-atoms formed and oxidation of the fuel by ozone is reduced. This explains the high amount of ozone seeding in the intake manifold is necessary for maintaining the same CA_{50}. The sensitivity of the CA_{50} is reduced at lower intake temperatures [56]. Figure 5.19b also illustrates the intake temperature reduction potential of ozone. The study showed that with sufficient ozone concentration, the cool flame occurs even in the case of iso-octane HCCI combustion [56]. Low temperature heat release is also observed in small quantities for the gasoline and E20 fuels at the lower intake temperatures, and it increases with increase in ozone addition [55]. Increase in low temperature heat release results into lower intake temperature requirement in HCCI combustion.

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