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Progress and recent trends in homogeneous charge compression ignition (HCCI) engines

Mingfa Yao*, Zhaolei Zheng, Haifeng Liu

State Key Laboratory of Engines, Tianjin University, Tianjin 300072, China

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ABSTRACT

HCCI combustion has been drawing the considerable attention due to high efficiency and lower nitrogen oxide (NO_x) and particulate matter (PM) emissions. However, there are still tough challenges in the successful operation of HCCI engines, such as controlling the combustion phasing, extending the operating range, and high unburned hydrocarbon and CO emissions. Massive research throughout the world has led to great progress in the control of HCCI combustion. The first thing paid attention to is that a great deal of fundamental theoretical research has been carried out. First, numerical simulation has become a good observation and a powerful tool to investigate HCCI and to develop control strategies for HCCI because of its greater flexibility and lower cost compared with engine experiments. Five types of models applied to HCCI engine modelling are discussed in the present paper. Second, HCCI can be applied to a variety of fuel types. Combustion phasing and operation range can be controlled by the modification of fuel characteristics. Third, it has been realized that advanced control strategies of fuel/air mixture are more important than simple homogeneous charge in the process of the controlling of HCCI combustion processes. The stratification strategy has the potential to extend the HCCI operation range to higher loads, and low temperature combustion (LTC) diluted by exhaust gas recirculation (EGR) has the potential to extend the operation range to high loads; even to full loads, for diesel engines. Fourth, optical diagnostics has been applied widely to reveal in-cylinder combustion processes. In addition, the key to diesel-fuelled HCCI combustion control is mixture preparation, while EGR is the main path to achieve gasoline-fuelled HCCI combustion. Specific strategies for diesel-fuelled, gasoline-fuelled and other alternative fuelled HCCI combustion are also discussed in the present paper.

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* Corresponding author. Tel.: +86 22 27406842x8014; fax: +86 22 27383362.
E-mail address: y_mingfa@tju.edu.cn (M. Yao).

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1. Introduction

The internal combustion engine is one of the key drivers in modern industrial society. Without the transportation performed by the millions of vehicles, we would not have reached the living standard of today. There are two types of internal combustion engines: spark ignition (SI) and compression ignition (CI). The conventional SI combustion is characterised by a flame propagation process. The onset of combustion in SI engines can be controlled by varying ignition timing from the spark discharge. Because the mixture is premixed and typically stoichiometric ($\lambda = 1$), the emissions of soot are orders of magnitude lower than that in the diesel processes. SI engines nowadays run on a stoichiometric mixture to utilize the catalyst for exhaust after treatment. Using a fixed air/fuel ratio means that the load controlling is possible only by controlling the air mass flow into the combustion chamber. The throttle used for this purpose gives rise to pumping losses and a reduction in efficiency. As a result, the major disadvantage of SI engines is its low efficiency at partial loads. The compression ratio in SI engines is limited by knock and can normally be limited in the range from 8 to 12 contributing to the low efficiency. Conventional diesel combustion, as a typical representation of CI combustion, operates at higher compression ratios (12–24) than SI engines. In this type of engine, the air–fuel mixture auto-ignites as a consequence of piston compression instead of ignition by a spark plug. The processes which occur between the two moments when the liquid fuel leaves the injector nozzles and when the fuel starts to burn, are complex and include droplet formation, collisions, break-up, evaporation and vapour diffusion. The rate of combustion is effectively limited by these processes. A part of the air and fuel will be premixed and burn fast, but for the larger fraction of the fuel, the time scale of evaporation, diffusion, etc. is larger than the chemical time scale. Therefore, the mixture can be divided into high fuel concentration regions and high temperature flame regions. In the high fuel concentration regions, a large amount of soot is formed because of the absence of O_2. Some soot can be oxidized with the increase of in-cylinder temperature. The in-cylinder temperature in a conventional diesel engine is about 2700 K, which leads to a great deal of NO_x emissions. For diesel engines, a trade-off between these two emissions is observed, and their problem is how to break through the compromise between NO_x and PM emissions. after treatment to reduce NO_x and particulates is expensive.

Consequently, the obvious ideal combination would be to find an engine type with high efficiency of diesel engines and very low emissions of gasoline engines with catalytic converters. One such candidate is the process known as homogeneous charge compression ignition, HCCI, on which we shall now focus attention.

1.1. HCCI principle

HCCI is characterised by the fact that the fuel and air are mixed before combustion starts and the mixture auto-ignites as a result of the temperature increase in the compression stroke. Thus HCCI is similar to SI in the sense that both engines use premixed charge

Abbreviations

ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BILE	Backward Illumination Light Extinction
CA50	Crank Angle at 50% Completion of Heat Release
CAI	Controlled Auto-Ignition
CFD	Computed Fluid Dynamic
CI	Compression Ignition
CN	Cetane Number
CNG	Compressed Natural Gas
COV	Coefficient of Variation
CR	Common Rail
DME	Dimethyl Ether
EGR	Exhaust Gas Recirculation
EVO	Exhaust Valve Open
GA	Genetic Algorithm
GDI	Gasoline Direct Injection
HCCI	Homogeneous Charge Compression Ignition
HCDC	Homogeneous Charge Compression Ignition Diesel Combustion
HTHR	High Temperature Heat Release
HTR	High Temperature Reaction
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closure
LES	Lotus Engine Simulation
LIF	Laser Induced Fluorescence
LII	Laser Induced Incandescence
LTC	Low Temperature Combustion
LTHR	Low Temperature Heat Release
LTR	Low Temperature Reaction
MK	Modulated Kinetic
MON	Motor Octane Number
MUIDIC	Multiple Stage Diesel Combustion
NTC	Negative Temperature Coefficient
NVO	Negative Valve Overlap
OI	Octane Index
OKP	Optimized Kinetic Process
PAH	Polycyclic Aromatic Hydrocarbons
PCCI	Premixed Charge Compression Ignition
PLIEF	Planar Laser Induced Exciplex Fluorescence
PLIF	Planar Laser Induced Fluorescence
PM	Particulate Matter
PREDIC	Premixed Lean Diesel Combustion
PRF	Primary Reference Fuel
RCM	Rapid Compression Machine
RON	Research Octane Number
SCCI	Stratified Charge Compression Ignition
SI	Spark Ignition
SMD	Sauter Mean Diameter
SOI	Start of Injection
TCI	Turbulence/Chemistry Interaction
TDC	Top Dead Centre
THC	Total Hydrocarbon
TRF	Toluene Reference Fuel
UHC	Unburned Hydrocarbon
UNIBUS	Uniform Balky Combustion System
VVA	Variable Valve Actuation
VVT	Variable Valve Timing
4VVAS	4-Variable Valve Actuating System

and similar to CI as both rely on auto-ignition to initiate combustion. HCCI combustion of diesel-like fuels displays a peculiar two-stage heat release, as shown in Fig. 1. The first stage of the heat release curve is associated with low temperature kinetic reactions, and the time delay between the first and main heat releases is attributed to the "negative temperature coefficient (NTC) regime" which locates between the two heat release stages [1,2]. In this NTC regime, the overall reaction rate decreases though the in-cylinder temperature increases, which leads to a lower reactivity of the system. Low temperature kinetics has been studied for some time, as this chemistry is responsible for knock in spark-ignition engines [3,4]. Heat release from low temperature reaction relates to octane numbers of fuels. The lower the octane number is, the more obvious the heat release of low temperature reaction. For gasoline-like fuels (high octane numbers), heat release from low temperature reaction (first-stage heat release) is less compared with diesel-like fuels at the same condition. Consequently, heat release from low temperature reaction is too little to obviously observe from the heat release profiles at most conditions for gasoline-like fuels. Research conducted with the use of optical diagnostics has shown that HCCI combustion initiates simultaneously at multiple sites within the combustion chamber and that there is no discernable flame propagation [1,5,6].

1.2. Pioneering research of HCCI combustion

The concept of HCCI was initially investigated for gasoline applications by Onishi et al. [7] in order to increase combustion stability of two-stroke engines. They found that significant reductions in emissions and an improvement in fuel economy could be obtained by creating conditions that led to spontaneous ignition of the in-cylinder charge. Stable HCCI combustion could be achieved between low and high load limits with gasoline at a compression ratio of 7.5:1 over the engine speed range from 1000 to 4000 rpm. Noguchi et al. [8] performed a spectroscopic analysis on HCCI combustion by experimental work on an opposed piston two-stroke engine. From optical investigations they noted that ignition took place at numerous points throughout the cylinder and no discernable flame front was observed during combustion. Using spectroscopic methods to detect the intermediate species, they measured high levels of CH_2O, HO_2, and O radicals within the cylinder before auto-ignition. These species are characteristic of low temperature auto-ignition chemistry of larger paraffinic

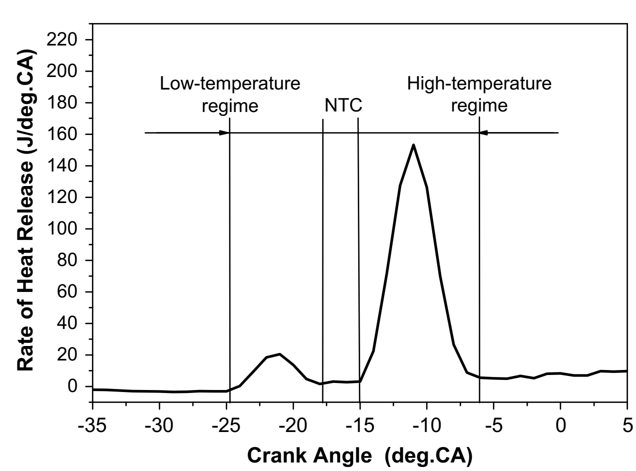

Fig. 1. Typical heat release curve from HCCI combustion of n-heptane fuel.

hydrocarbon fuels. After ignition, they observed high concentrations of CH, H, and OH radicals, which were indicative of high temperature chemistry during the bulk burn. Building on previous work on two-stroke engines, Najt and Foster [9] extended the work to four-stroke engines and attempted to gain additional understanding of the underlying physics of HCCI combustion. They concluded that HCCI auto-ignition is controlled by low temperature (below 1000 K) chemistry and the bulk energy release is controlled by the high temperature (above 1000 K) chemistry dominated by CO oxidation. Combined with the previous work of Onishi [7] and Noguchi [8], it can be concluded that, unlike traditional SI combustion that relies on the flame propagation and diesel combustion that is heavily dependent on the fuel/air mixing, HCCI combustion is a chemical kinetic combustion process controlled by temperature, pressure, and composition of the in-cylinder charge. Correspondingly, they noted that HCCI combustion suffers from a lack of control of the ignition process and a limited operating range. Thring [10] further extended the work of Najt and Foster [9] in four-stroke engines by examining the performance of an HCCI engine operated with a fully-blended gasoline. It was also found that the operating regime was restricted to part load operation, and control of the auto-ignition timing was problematic.

As discussed above, initial efforts with HCCI involved gasoline-fuelled engines, and this technology continues to be strongly pursued today. In many researches, gasoline-fuelled HCCI combustion is also called controlled auto-ignition (CAI) combustion. However, the need to reduce emissions from diesel engines led to investigation into the potential of diesel-fuelled HCCI beginning in the mid-1990s. For diesel fuel, port fuel injection is perhaps the most straightforward approach to obtaining a premixed charge and this approach has been used in some of the earlier investigations of diesel-fuelled HCCI [11,12]. Ryan et al. [11] used port fuel injection to supply diesel fuel into the intake air stream. An intake air heater upstream of the fuel injection point allowed preheating, with engine compression ratios varying between 7.5 and 17:1. This study and the following study on the same engine [12] found: for diesel-fuelled HCCI, very premature ignition and knocking occurred using normal diesel compression ratios; relatively high intake temperatures were required to minimize the accumulation of liquid fuel on surfaces in the intake system; and unburned HC emissions tended to be very high, but NO_x emissions were dramatically reduced.

The results from these pioneering investigations indicate the strong potential of HCCI to improve thermal efficiency of gasoline-fuelled engines and substantially reduce NO_x and soot emissions of diesel-fuelled engines. Furthermore, these results confirmed the dominating role of chemical kinetics in HCCI combustion, which has significancy to following studies. However, they also foresaw some problems of this new combustion mode.

1.3. Challenges of HCCI combustion

Based on the pioneering research of HCCI combustion, obstacles that must be overcome before the potential benefits of HCCI combustion can be fully realized in production applications became clear with more and more studies. This section describes the main difficulties with this combustion mode.

1.3.1. The difficulty in combustion phasing control

One of the principal challenges of HCCI combustion is control of the combustion phasing. Unlike conventional combustion, a direct method for controlling the start of combustion is not available. Instead, the start of combustion is established by the auto-ignition chemistry of the air–fuel mixture. Auto-ignition of a fuel–oxidizer mixture is influenced by the properties of the mixture and by the time–temperature history to which it is exposed. Hence, combustion phasing of HCCI engines is affected by the following factors [13]: auto-ignition properties of the fuel, fuel concentration, residual rate and, possibly, reactivity of the residual, mixture homogeneity, compression ratio, intake temperature, latent heat of vaporization of the fuel, and engine temperature, heat transfer to the engine and other engine-dependent parameters.

1.3.2. High levels of Noise, UHC and CO emissions

The second main challenge for HCCI operation is the potential increase in noise, UHC (unburned hydrocarbon) and CO emissions. As with all homogeneous combustion systems, a significant portion of the in-cylinder fuel is stored in crevices during the compression stroke and escapes combustion. Moreover, the burned gas temperature is too low to consume much of this unburned fuel when it re-enters the cylinder during the expansion stroke. This results in significant increase in both HC and CO emissions relative to conventional combustion. In addition, the peak burned gas temperatures are too low (lower than 1400 K or 1500 K) to complete the CO to CO_2 oxidation at low loads, and the combustion efficiency deteriorates precipitously [14]. This loss of combustion efficiency combined with ignition difficulties limits the effectiveness of HCCI combustion at the lightest loads. At higher loads the rate of pressure rise can become so large that engine noise increases significantly, and if left unchecked, engine damage may occur [15].

1.3.3. Operation range

In addition to the above problems, another fundamental barrier in HCCI development is extending the operating load range whilst maintaining the full HCCI benefit. Extending the operating range is as important as the auto-ignition process. In addition to expanding the HCCI operation to higher load, very light load operation is also limited, because there is insufficient thermal energy to trigger auto-ignition of the mixture late in the compression stroke. Moreover, excess CO and HC emissions in combination with low exhaust gas temperature at near-idle operation makes this combustion mode less appealing from combustion efficiency and exhaust emissions perspectives.

1.3.4. Cold start

Since temperatures are very low at cold start operations and the heat loss from the compressed charge to the cold combustion chamber walls is so high, the HCCI engine will encounter a major difficulty in firing during cold start operation. To overcome this difficulty, the engine may have to be started in a conventional mode and then switched to the HCCI mode after a short warm-up period. Therefore, maintaining a real homogeneous combustion after cold start will also be a real challenge. HCCI operation for cold starts is an area where much more developmental effort is needed. Obviously, achieving a robust HCCI combustion at very light load with full HCCI benefits in fuel efficiency and emissions is as important as extending the HCCI operation to high loads.

1.3.5. Homogeneous mixture preparation

Effective mixture preparation and avoiding fuel/wall interactions is crucial for achieving high fuel efficiency, reducing HC and PM emissions, and preventing oil dilution. Fuel impinging on the surfaces of the combustion chamber has been proven disadvantageous to HC emissions even for moderately volatile fuels such as gasoline [16]. Mixture homogeneity has an effect on auto-ignition reactions that control the HCCI combustion phasing [17], and there is significant evidence that low NO_x emissions can be produced even with some degree of mixture inhomogeneity within the combustion chamber. Homogeneous mixture preparation is most difficult for fuels with reduced volatility such as diesel, which requires elevated intake air temperatures for low-smoke operation when port-injected.

Based on pioneering research, HCCI combustion has been attracting growing attention in recent years due to the potential in significantly reducing NO_x and particulate emissions whilst achieving high thermal efficiency at part loads with the development of electronic control technology, numerical simulations and optical technology. Great progress has been made in the three main research fields of HCCI, which are, fundamental theory, gasoline-fuelled HCCI combustion and diesel-fuelled HCCI combustion. Circumfusing above three aspects, not only models applied to HCCI research and the effects of fuel characteristics on HCCI were discussed, but also the process of extending HCCI concept is focused on with more and more researches. In addition, the general summarization of control strategies for diesel-fuelled and gasoline-fuelled HCCI in the development of HCCI is one of the keys in the following paragraph.

2. Advancements in the investigation of HCCI fundamental theory

2.1. Application of numerical simulation to HCCI combustion

With the development of computers, numerical simulations have become a powerful tool to investigate HCCI and to seek control strategies of HCCI, having both higher flexibility and lower cost compared with engine experiments. As illustrated by the pioneering studies of HCCI combustion, the dominating role of chemical kinetics in HCCI has been established. Circumscribing the central chemical kinetics, there are five categories of models applied to HCCI engine modelling, as discussed in the following sections.

2.1.1. Zero-dimensional single-zone models with detailed chemistry

Zero-dimensional single-zone models with detailed chemistry are the simplest models in HCCI modelling. In the zone, in-cylinder temperature and mixture concentration of fuel and air are both homogeneous. Because of the dominant role of chemical kinetics, it is important to have an understanding of the reaction mechanisms of hydrocarbon fuels in order to correctly interpret experimental data and guide HCCI engine development. The detailed mechanism investigation of HCCI combustion based on zero-dimensional thermo-kinetic model is one of the cores of HCCI simulation research in recent years.

From the advent of HCCI concept, chemical kinetic models received great development. The evaluation of detailed reaction mechanisms has been extended from small fuel molecules (CH_4, CH_3OH, C_2H_2, C_2H_4) [18–20] to higher hydrocarbon fuels such as n-heptane [21] and iso-octane [22]. There are three fundamental "regimes" of hydrocarbon oxidation chemistry, although not all the fuels exhibit characteristics of each of the regimes as shown in Fig. 1. In the following discussion, n-heptane fuel is taken as an example to describe the three combustion regimes of hydrocarbon fuels. After the H-abstraction step from the n-heptane fuel molecule, the heptyl radical can react with molecular oxygen to form an alkylperoxy radical. The low temperature path continues with an isomerization step in which the alkylperoxy radical is transformed into a hydroperoxy alkyl radical. A second oxygen molecule is added to the product of the isomerization step and the oxohydroperoxide radical can then isomerise further and decompose into a relatively stable ketohydroperoxide species and OH radicals. As the temperature increases further, the alkylperoxy radicals decompose back into initial reactants; the formation of olefins and hydroperoxyl radicals is favoured, and the overall reaction rate decreases with the increase of temperature. This is classical NTC behaviour observed in low temperature oxidation of paraffinic fuels. In the high temperature regime, the increasing temperature is high enough so that hydrogen–oxygen branching reactions control the reaction rate. The heat release rate of this stage is dominated by the oxidation process of carbon monoxide to carbon dioxide.

The oxidation mechanisms of hydrocarbon fuels change substantially over the ranges of pressure and temperature encountered in an HCCI engine. Many researchers have studied the detailed chemical kinetics process of hydrocarbon fuels in HCCI combustion by zero-dimensional, single-zone approach. Iida and co-workers [23,24] investigated the characteristics of auto-ignition and combustion in the HCCI engine using elementary reaction calculations for an engine fuelled with natural gas and n-butane. The heat release rate of natural gas HCCI combustion only shows one obvious peak. N-butane is the fuel with the smallest carbon number in the alkane family that shows two-stage auto-ignition, heat release with low temperature reactions (LTR) and high temperature reactions (HTR), similar to higher hydrocarbons. Daisho and his colleagues [25] revealed the effects of initial charge conditions, compression ratios and excess air ratios on ignition and combustion. From their parametric study, they found that HCCI combustion of n-heptane/air mixtures is classified into three types of combustion: complete combustion, only LTR, and misfire, depending on compression ratios and excess air ratios. In addition, the combustion mechanisms of n-heptane were investigated using a zero-dimensional thermodynamic model coupled with a detailed kinetic model by Yao and Zheng [26]. The reaction paths in the LTR, NTC region and HTR of n-heptane HCCI combustion were presented in this study. The results indicated that the HTR can also be separated into two stages. The first stage is mainly the process of the conversion of CH_2O to CO which is controlled by H_2O_2 decomposition. The second stage is mainly the process of the oxidization of CO to CO_2 which is dominated by the reaction $H + O_2 = O + OH$.

In the research of detailed chemical kinetics mechanisms of HCCI combustion fuelled with gasoline and diesel, iso-octane and n-heptane are often substitutes for the two fuels. This is because gasoline and diesel fuels are mixtures of many components and the detailed mechanisms of gasoline and diesel fuels are difficult to describe. However, research on the detailed mechanisms of the two fuels has been making great progress due to the efforts of many researchers. Naik et al. [27] developed a surrogate gasoline reaction mechanism including 5 component fuels of iso-octane, n-heptane, 1-pentene, toluene, and methyl-cyclohexane. The mechanism consists of 1328 species and 5835 reactions. Predictions are in reasonably good agreement with HCCI engine data. Andrae et al. [28] developed a kinetic model for the auto-ignition of toluene reference fuels (TRF) with 2 component fuels of n-heptane and toluene. Important features of the auto-ignition of the mixture proved to be cross-acceleration effects, where hydroperoxy radicals produced during n-heptane oxidation dramatically increased the oxidation rate of toluene compared with when toluene alone was oxidized. Good agreement between experiment and prediction was found when the model was validated against shock tube auto-ignition delay data for gasoline surrogate fuels.

Though the use of chemical kinetics model can reveal the combustion mechanism of HCCI combustion and the time of auto-ignition, they suffer significant shortcomings in predicting the rate of heat release, the combustion duration and emissions due to the simplifying assumption of strict homogeneity throughout the combustion chamber. The in-cylinder temperature distribution is actually non-uniform and the high temperature region in the centre of the chamber is more responsible for auto-ignition. Therefore, using only chemical kinetics has not been successful in simulating combustion by assuming a uniform temperature distribution.

2.1.2. Quasi-dimensional multi-zone with detailed chemistry

Some of the limitations of single-zone HCCI models can be overcome by performing multi-zone modelling instead. Zones in the Multi-zone model mean that computational volume is divided into an arbitrary number of volumes by different researches. Every zone is initialized by different initial temperature and other needed parameters. Mass exchange and heat transfer between zones may be considered. It is necessary to include the crevices and boundary layer (low temperature zones) to obtain good resolution of the HC and CO production for HCCI combustion. A wide variety of multi-zone models has been published in the literature. The main differences between the models can be categorized as follows: the number of zones, the kind of zones (e.g. adiabatic core zones, boundary layers, crevices and mass exchange zones), and the types of interaction that occur between zones (pressure–volume work, heat transfer, mass exchange). Since the zones in the multi-zone models can represent crevices, boundary layers and core zones, the models are also referred to as being quasi-dimensional.

A comprehensive quasi-dimensional model was proposed by Fiveland and Assanis [29,30] with the intent to bridge the gap between the sequential thermo-kinetic models and the zero-dimensional models so as to be able to predict performance and emissions, especially under turbocharged conditions. The model is based on a full-cycle simulation code and includes an adiabatic core, a predictive boundary layer model, and a crevice region as shown in Fig. 2. In particular, the thermal boundary layer is driven by compressible energy considerations, and hence is of varying thickness, and is solved at multiple geometric locations along the piston–liner interface. Interaction of the adiabatic core and the thermal boundary layer is presented in Fig. 3. A full dynamic ring pack model gave reasonably good agreement for unburned hydrocarbons. CO predictions were less satisfactory due to lack of detailed thermal resolution in the near wall regime. In the work of Komninos et al. [31], a multi-zone model was presented for the simulation of HCCI engines. In their model, a phenomenological model was developed to describe mass exchange between zones and the flow of a portion of the in-cylinder mixture in and out of the crevice region. The crevice flow was included in the model because the crevice regions were considered to contribute to unburned HC emissions. More recently, a modified multi-zone chemical kinetic model was developed by Kongsereeparp and Checkel [32] to describe the effects of base fuel/reformed fuel blends on HCCI engine combustion. Their model included imperfectly stirred mixture effects via the virtual enthalpy exchange between hotter internal residual and cooler intake mixtures. No mass transfer was considered between zones in their model.

With suitable calibration, the quasi-dimensional models have shown that they can include key geometric effects without excessive computational times. Prediction of HC and CO emissions is included in these models. The key limitation of the quasi-dimensional models is their inability to predictively describe stratification or inhomogeneities in residual fraction that are likely to exist in practical applications, especially in direct injection systems.

2.1.3. One-dimensional engine cycle with detailed chemistry

The disadvantage of the models mentioned above is that they can only evaluate the processes in the combustion chamber during the intervals the valves are closed. Such models usually start simulations at intake valve closure (IVC) and stop at exhaust valve open (EVO). This means that initial values (average mixture temperature, equivalence ratio and residual gas fraction (RGF) that might be difficult to choose) have to be specified at IVC. To solve this problem, the aforementioned models can be combined with engine cycle simulation codes that can predict the desired values at IVC. These codes are often one-dimensional, predicting results for the complete engine flow system from air intake to exhaust pipe. Since the results are calculated for the complete engine cycle, gas exchange processes are modelled as well. The models presented in the literature differ mainly in terms of the combustion model implemented in the one-dimensional cycle simulation code. It is, for instance, possible to implement single-zone detailed chemical kinetics models or multi-zone detailed chemical kinetics models.

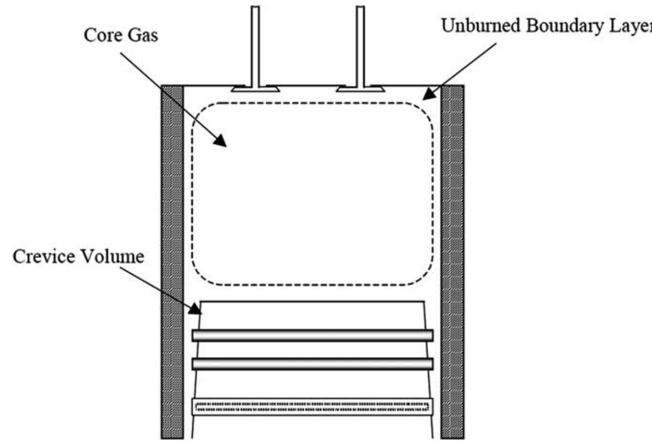

Fig. 2. General layout of the quasi-dimensional thermodynamic simulation showing the interacting adiabatic core, thermal boundary layer, and crevice regions [30].

A one-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model was first developed by Fiveland and Assanis [33,34] to study combustion and performance of HCCI engine. In their model, the one-dimensional quasi-steady flow model is used to model flow through both the intake and exhaust valves during the gas exchange processes. The combustion process in a homogenous charge compression ignition engine is described with a detailed chemical kinetic mechanism. The CHEMKIN libraries [35] have been used to formulate a stiff chemical kinetic solver suitable for integration within a complete engine cycle simulation, featuring models of gas exchange, turbulence and wall heat transfer. Xu et al. [36] have presented a model to study the combustion and gas exchange in an HCCI engine using a single-zone combustion model based on CHEMKIN and an in-house engine simulation code GESIM which is a thermodynamic engine simulation code. In their simulation, characteristics of the

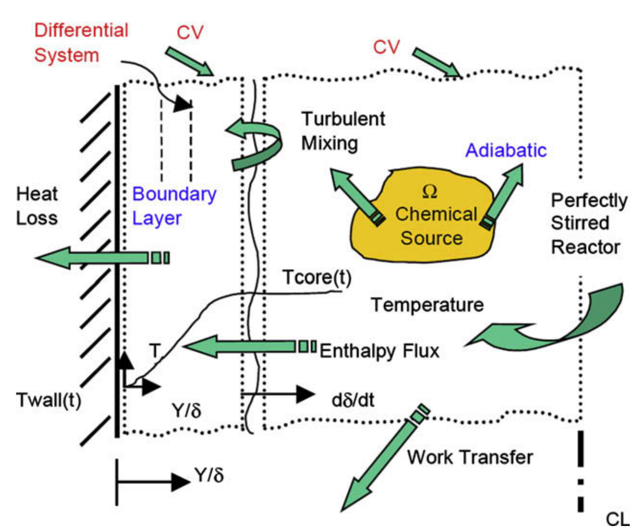

Fig. 3. Interaction of the adiabatic core and the thermal boundary layer [30].

combustion and gas exchange process in the engine were identified. Their study provided further understanding of the processes in such HCCI engines and the results can be used to support the design of prototype experimental engines. For a multi-cylinder engine, gas dynamic effects in the intake and exhaust manifolds have significant impact on the gas exchange process, and therefore simulations of HCCI engines should be better achieved by using gas dynamic simulation codes. Hence, Xu et al. [37] have developed a model combining the CHEMKIN based single-zone chemical kinetic combustion model with the Ricardo 1-D gas dynamics and engine simulation code, WAVE.

Ogink [38] developed a BOOST-SENKIN single-zone model to accurately predict the moment of auto-ignition and the rate of heat release in a gasoline HCCI engine. Milovanovic et al. [39] analysed the influence of the variable valve timing (VVT) strategy on the gas exchange process in an HCCI engine fuelled with standard gasoline fuel by experiment and simulation. The simulation was carried out by combining the Aurora detail chemical kinetics code from the CHEMKIN III combustion package with the one-dimensional fluid dynamic Lotus Engine Simulation (LES) code. Ogink and Golovitchev [40] further extended their model to BOOST-SENKIN multi-zone detailed chemical kinetics model to accurately predict fuel consumption, emissions and indicated mean effective pressure (IMEP) for a wide range of experimental operating conditions. The multi-zone model presented in their research has nine zones: a constant volume zone representing crevices, a zone representing the quench layer, a core zone that can exchange mass with the quench layer, and six core zones of constant mass. Narayanaswamy and Rutland [41] developed a model to understand early direct injection diesel HCCI processes. GT-Power, a commercial one-dimensional engine cycle code, was coupled with an external cylinder model which incorporates submodels for fuel injection, vaporization, multi-zone detailed chemistry calculations (CHEMKIN), heat transfer, energy conservation and species conservation. Schematic representation of coupling between GT-Power and user-defined models is shown in Fig. 4a. Fig. 4b gives schematic distribution of zones in the cylinder in the multi-zone detailed chemistry submodel.

The specific information of references belonging to above three types of models is summarized in Table 1, such as types of models, codes, fuels, etc.

2.1.4. Multi-dimensional CFD with multi-zone detailed chemistry

As described earlier, multi-zone models are more sophisticated than single-zone counterparts because they can represent the inhomogeneity present in the cylinder prior to combustion. However, one difficulty is that realistic temperature distributions, equivalence ratios and RGF need to be specified over the various zones at IVC.

In order to obtain some of the zonal resolution afforded by computed fluid dynamic (CFD) models and yet reduce the computational time required by detailed kinetics calculation, a segregated, sequential multi-zone modelling approach has been pioneered by Aceves et al. [42–44]. This is a hybrid procedure that uses a 3D-CFD code (KIVA) and a Quasi-dimensional multi-zone detailed chemical kinetics code. KIVA is a transient, three-dimensional, multiphase, multicomponent code for the analysis of chemically reacting flows with sprays has been under development at the Los Alamos National Laboratory for the past several years [45]. It is an open source, and you also can get the code from Ref. [45]. In this approach, the fluid mechanics code is run to evaluate the temperature distribution inside the cylinder without combustion between the zones, and the information is then fed into the chemical kinetics code. Every zone in multi-zone chemical kinetics code is initialized by different initial temperature and other

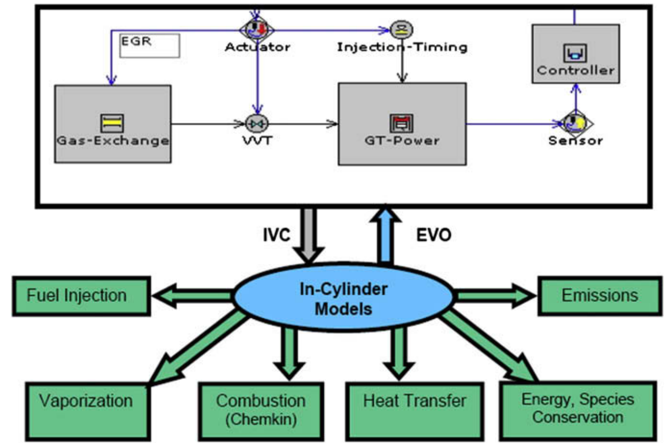

Fig. 4. Schematic representation of coupling between GT-Power and user-defined models [41].

needed parameters obtained from CFD code. This two-step procedure takes full account of the effects of temperature gradients inside the cylinder, with a much-reduced computational intensity that makes it accessible to current computers. Ten zones have been selected for this analysis as shown in Fig. 5 [42]. The initial mass and temperature of each zone are given according to CFD calculation results. Detailed combustion kinetics calculations were then carried out in each mass temperature group, with the groups interacting with each other only by P-dV work and subject to the constraints of the time-varying chamber volume. Both species and heat diffusion between zones was thought to be unimportant due to the rapid combustion time and was not considered. The model succeeded in resolving the low temperature regions of the chamber, along the wall and in the ring crevice, and showed that these zones are responsible for combustion inefficiency, UHC, and CO emissions. However, their further research [46] indicated that CO emissions are extremely sensitive to zone resolution, and 10 zones are typically not enough to provide appropriate resolution. In order to more accurately predict CO emissions, Aceves et al. [47] developed a 40-zone model to analyse the effect of piston crevice geometry on HCCI combustion and emissions. The 40 zones mean that the computational volume is divided into 40 regions and every zone has different initial temperature and mass fraction. Using 40 zones may still not provide the necessary resolution in some cases, but it does increase the likelihood of capturing the very small temperature range from which most CO emissions originate. Three different pistons of varying crevice size were analysed in this study. Crevice sizes were 0.26, 1.3 and 2.1 mm, whilst a constant compression ratio was maintained. The results show that the multi-zone model can predict pressure traces and heat release rates with good accuracy. Combustion efficiency is also predicted with good accuracy for all cases. CO emissions are underpredicted, but the results are better than those obtained in previous publications. HC emissions are well predicted. The improvement is attributed to the use of a 40-zone model. The model can successfully predict the effect of crevice geometry on HCCI combustion, and therefore it has applicability in the design of HCCI engines with optimum characteristics for high efficiency, low emissions and low peak cylinder pressure.

Nevertheless, CFD calculations were limited to the closed valve portion of the engine cycle and the chamber was treated only in two dimensions in these above-mentioned models. One approach to solve this problem is to perform a multi-dimensional CFD simulation prior to the multi-zone calculations, and to use the parameter distributions obtained from the CFD results as input for

Table 1
Zero-dimensional single-zone model, Quasi-dimensional multi-zone model and One-dimensional cycle simulation coupled with chemical kinetics model.

Types of models	Codes	Fuels	Focuses	Experiment validating
Zero-dimensional single-zone	Chemkin	Natural gas [23] n-butane [24]	Auto-ignition and combustion in the HCCI engine [23,24]	HCCI engine (pressure, temperature and heat release rate profiles)
Zero-dimensional single-zone	Chemkin	n-Heptane	Effects of initial charge conditions, compression ratios and excess air ratios on ignition and combustion [25]	HCCI engine (cylinder pressure)
Zero-dimensional single-zone	Chemkin	n-Heptane	The chemical reaction kinetics of n-heptane for HCCI combustion process [26]	No experiment validating because of the use of reaction mechanism which has been widely validated by other researches
Zero-dimensional single-zone	Chemkin	Iso-octane, n-heptane, 1-pentene, toluene, methyl-cyclohexane	Developing a surrogate gasoline reaction mechanism [27]	HCCI engine (crank angle for 10% burn, heat release rate etc.) shock tube
Zero-dimensional single-zone	Chemkin	n-Heptane and toluene	Developing a kinetic model for the auto-ignition of toluene reference fuels [28]	Shock tube
Quasi-dimensional multi-zone	Chemkin	Natural gas	To bridge the gap between the sequential thermo-kinetic models and the zero-dimensional models so as to be able to predict performance and emissions [29,30]	HCCI engine (pressure, location of 10% and 50% burned, turbocharger efficiency, UHC and CO emissions, etc.)
Quasi-dimensional multi-zone	Chemkin	Iso-octane	A phenomenological model was developed to describe mass exchange between zones and the flow of the in-cylinder mixture in and out of the crevice region [31]	HCCI engine (in-cylinder pressure and heat release rate.)
Quasi-dimensional multi-zone	Chemkin	Natural gas n-heptane	The effects of base fuel/reformed fuel blends on HCCI engine combustion [32]	HCCI engine (in-cylinder pressure and ignition timing, etc.)
One-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model	Chemkin; full-cycle simulation code	Hydrogen natural gas [33], methane and ethane methane and propane, methane and butane [34]	The CHEMKIN libraries are integrated within a complete engine cycle simulation to investigate combustion and performance of HCCI engine [33,34]	HCCI engine (cylinder pressure during the valves close and gas exchange) [34]
One-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model	Chemikin and GESIM	Iso-octane and n-heptane	Investigate the combustion and gas exchange in an HCCI engine [36]	The results of the combined modelling were compared to the experimental data available within the company of Jaguar Cars with a reasonable agreement.
One-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model	Chemikin and WAVE	Iso-octane and n-heptane	Simulations of HCCI engines using gas dynamic simulation codes [37]	The results of the combined modelling were compared to the experimental data available within the company of Jaguar Cars with a reasonable agreement.
One-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model	BOOST and SENKIN	Iso-octane, n-heptane and toluene	Predict the moment of auto-ignition and the rate of heat release in a gasoline HCCI engine [38]	HCCI engine (cylinder pressure in the full cycle)
One-dimensional cycle simulation coupled with single-zone detailed chemical kinetics model	Chemikin and Lotus engine simulation (LES)	Iso-octane and n-heptane	influence of the VVT strategy on the gas exchange process [39]	HCCI engine (cylinder pressure in the full cycle, exhaust gas temperature, IMEP, BMEP and BSFC)
One-dimensional cycle simulation coupled with multi-zone detailed chemical kinetics	BOOST and SENKIN	Iso-octane n-heptane and toluene	To accurately predict fuel consumption, emissions and IMEP for a wide range of experimental operating conditions [40]	HCCI engine (cylinder pressure, heat release rate and HC, CO, CO_2 emissions)
One-dimensional cycle simulation coupled with multi-zone detailed chemical kinetics	GT-Power and Chemikin	n-Heptane	To understand early direct injection diesel HCCI processes [41]	HCCI engine (cylinder pressure at different injection timings and EGR rates, heat release rate)

the zones in the multi-zone model. The best method is to represent the engine geometry by a 3-D mesh and to include the gas exchange process in the CFD calculations as well, because this allows the mixture inhomogeneity present in the cylinder during the compression stroke to be captured.

Babajimopoulos et al. [48] have extended the model proposed by Aceves et al. to study the effects of valve events and gas exchange processes in the framework of a full-cycle HCCI engine simulation. The multi-dimensional fluid mechanisms code KIVA-3V was used in 3-D to simulate the exhaust, intake, and compression strokes up to a transition point, whilst a multi-zone, thermo-kinetic code computes the combustion event. After validation by comparison with a natural gas Caterpillar engine, the model was used to explore the effects of variable valve actuation (VVA). The model was able to identify not only large variations in temperature, but also significant non-homogeneities in residual content throughout the chamber at the beginning of combustion. Flowers et al. [49] improved the multi-zone model proposed by Babajimopoulos et al. by including mixing effects to investigate how the handling of mixing and heat transfer in a multi-zone kinetic solver affects the prediction of CO and UHC emissions for HCCI engine combustion simulation. The fluid mechanics is solved with high spatial and temporal resolution (40,000 cells). The chemistry is simulated with high temporal resolution, but low spatial resolution (20 computational zones). The results show that CO and UHC emissions may be greatly influenced by the mixing and heat transfer during expansion. Babajimopoulos et al. [50] also extended previous modelling efforts to include the effect of RGF distribution on the onset of ignition and the rate of combustion using a multi-dimensional fluid mechanics code (KIVA-3V) sequentially with a multi-zone code with detailed chemical kinetics. The results indicate that for different valve strategies and the same amount of

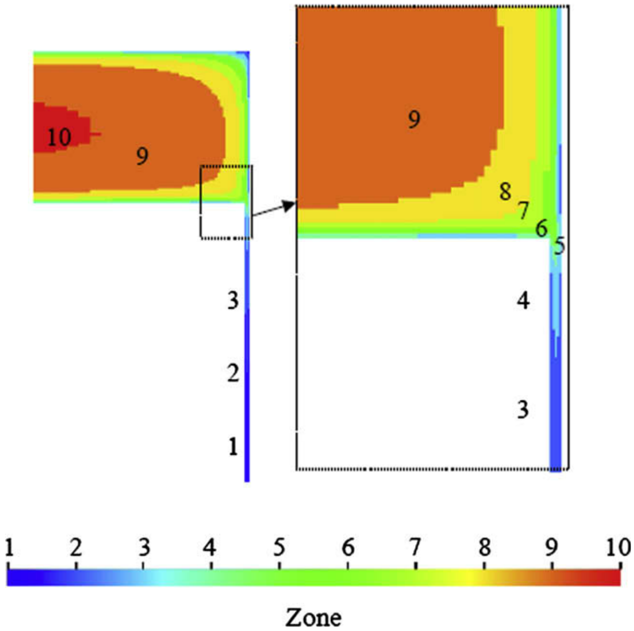

Fig. 5. Geometrical distribution of the 10 zones inside the cylinder [42].

internal EGR, the degree of temperature and composition stratification in the cylinder can be quite different.

Aceves et al. [51] conducted a detailed numerical analysis of HCCI engine operation at low loads to investigate the sources of HC and CO emissions and the associated combustion inefficiencies. A detailed iso-octane mechanism was coupled with 40 zones and 51,000 cells were used in the KIVA-3V computation in this research. The computational results agreed very well with experimental results. The computational model showed where the pollutants originate within the combustion chamber, thereby explaining the changes in the HC and CO emissions as a function of equivalence ratio. Up to now, research work by Aceves et al. has been focused on developing and validating multi-zone models for HCCI combustion. The results have shown considerable success in predicting combustion and emissions over multiple geometries, fuels, and operating conditions. In the following work [52], the multi-zone model was applied to study the effect of turbulence on HCCI combustion according to previous experiments in which an HCCI engine was run with two different piston geometries: a low turbulence flat-top piston (the disc geometry) and a high turbulence piston with a square bowl (the square geometry). The detailed iso-octane mechanism was also coupled with 40 zones and 54,000 cells were used in the KIVA-3V computation. The multi-zone model, which made the basic assumption that HCCI combustion is controlled by chemical kinetics, was capable of explaining the experimental results obtained for different levels of turbulence, without considering a direct interaction between turbulence and combustion. A direct connection between turbulence and HCCI combustion may still exist, but it seems to play a relatively minor role in determining burn duration at the conditions analysed in this research.

2.1.5. *Multi-dimensional CFD with detailed chemistry*

Multi-dimensional CFD models have the highest potential for predicting realistic results when the geometry of the combustion chamber is resolved in full detail, in combination with a detailed chemistry approach to model combustion. Of course, the required computational resources can become enormous depending on the CFD mesh resolution and the size of the reaction mechanism implemented.

Agarwal and Assanis [53–55] reported on the coupling of a detailed chemical kinetic mechanism for natural gas ignition (22 species and 104 elementary reactions) with the multi-dimensional reacting flow code KIVA-3V and explored the auto-ignition of natural gas injected into a quiescent chamber under diesel-like conditions. Full kinetics was used up to the ignition point. Kong et al. [56–58] proposed a similar approach up to the point of ignition, while after ignition they introduced a reaction rate incorporating the effects of both chemical kinetics and turbulent mixing over characteristic timescales. The turbulent timescale was defined as the characteristic time for eddy break-up, whilst the kinetic timescale was estimated as the time needed for a species to reach the equilibrium state under perfect-mixing conditions. In addition, with a numerical model, Kong et al. [59] investigated the effects of flow turbulence on premixed iso-octane HCCI engine combustion. The numerical model is based on the KIVA code which is modified to use CHEMKIN as the chemistry solver. It was found that turbulence has significant effects on HCCI combustion. In their engine setup, the main effect of turbulence is to affect the wall heat transfer, and hence to change the mixture temperature which, in turn, influences the ignition timing and combustion duration. The results imply that it is preferable to incorporate detailed chemistry in CFD codes for HCCI combustion simulations so that the effect of turbulence on wall heat transfer can be better simulated. On the other hand, it was also found that the onset of combustion is very sensitive to the initial conditions so that an accurate estimate of initial mixture conditions is essential for combustion simulations. The effects of inhomogeneous distributed EGR gas inside the cylinder can also be investigated using multi-dimensional CFD with detailed chemistry models. Tominaga et al. [60] have done this study for HCCI combustion fuelled with natural gas. The results showed that when EGR gas is distributed densely in the lower temperature zone (i.e. near the outer wall of the cylinder), the EGR effects are reduced and, hence, combustion is enhanced compared with cases in which the EGR gas distribution is homogeneous. Conversely, when EGR gas is densely distributed in a higher temperature zone (i.e. near the cylinder centre), the EGR effects are more pronounced and combustion is further moderated than in the case of homogeneous EGR.

Surprisingly, it is not necessary to know the detailed variation histories of every species in HCCI research. Only overall variation histories of the reaction system and certain details of some key species require consideration. Thus the reduced chemical kinetics models constructed by skeletal form have obvious superiority when coupled with the multi-dimensional CFD model, and this is the reason that the reduced or skeletal mechanisms have received great attention. Both the Cox and Cole [61], and Hu and Keck [62] models can generally reproduce the overall trend for ignition delay of specific hydrocarbons of interest in the low/intermediate temperature regime. Li et al. [63] have modelled the auto-ignition of primary reference fuels (PRF) and their mixtures at engine conditions. Griffiths et al. [64] presented a unified reaction model for alkane combustion based on their long-term investigation of the kinetic structures of alkyl radicals. Tanaka et al. [65] added global breakdown reactions, which convert intermediate products formed during the ignition stage to CO and HO_2, as well as H_2 and CO oxidation reactions to the Hu and Keck mechanism. This mechanism could describe the entire two-stage ignition process. Similar to Tanaka et al., Zheng et al. [66] suggested that the large molecules produced from the LTR should be broken up into smaller C1–C3 molecules, which link the pre-ignition and HTR regimes. Su and Huang [67] developed a reduced chemical kinetic model of n-heptane, which is based on the two reduced models of Li et al. and Griffiths et al. as well as a genetic algorithm (GA) optimization methodology. The model combines the chemistry for both low/intermediate and high temperature regimes, and is able to reproduce the NTC dependence on temperature. Jia and Xie [68] developed a skeletal mechanism of iso-octane in the prediction of ignition delay, heat release rate, fuel consumption, CO formation and production of other species with as few species as possible.

Ohashi et al. [69] employed CFD codes (KIVA and VECTIS) and chemical kinetic models for iso-octane and PRF mixtures that can be utilized to investigate HCCI engine operations on gasoline. First, they reduced and optimized the chemical mechanisms of iso-octane and PRF using the GA method. Second, they tested reduced and optimized reaction mechanisms and three-dimensional CFD codes for real engine operation. In iso-octane, the tendency of calculated ignition delay for various air/fuel ratio corresponded to those measured with conventional valve timing. Gasoline ignition timing measurements also corresponded to those calculated with PRF models using equivalent octane values. The heat released during calculations is slightly different from that measured experimentally.

A multi-dimensional model (the coupled model of Star-CD and Chemikin) was adopted to investigate the combustion and emissions formation of dimethyl ether (DME) and methanol dual fuel HCCI engine by Yao et al. [70]. A DME/methanol reduced mechanism consisting of 27 species and 35 reactions was used to simulate the fuel chemistry. The simulation results indicate that both LTR and HTR take place in some specific locations in the cylinder and then propagate to the entire cylinder. The quasi-low temperature reaction occurs in the core zone first, and the HTR starts from the core zone adjacent to the combustion chamber axis. The main components of UHC emissions are the unburned fuels (DME and methanol) and CH_2O. The unburned fuels mainly reside in the piston-ring crevice region whilst CH_2O dwells mainly in the region next to the cylinder liner wall. The majority of CO emissions are located in the region near the top surface of the piston.

Finally, it must be noticed that there are still other CFD codes were used in the numerical simulation of HCCI combustion, such as CFX [71], FIRE [72,73], IFP-C3D [74,75], Star-CD [70,76], and VECTIS [69,77]. It must notice that all these CFD codes are from commercial software, which is not open to the users and need a license. However, the KIVA is an open source, which results in the convenience of the submodel development.

The specific information of references belonging to two types of multi-dimensional CFD models is summarized in Table 2.

2.2. Effects of fuels, additives and fuel modification on HCCI chemical kinetics process

2.2.1. Effects of fuel characteristics

Due to the limitation caused by combustion mode and engine configuration, conventional combustion engines must use specific fuels. For spark-ignition engines, gasoline which has good volatility and anti-knock characteristic is appropriate; whilst for compression–ignition engines, diesel which has higher viscosity and lower resistance to auto-ignition is more appropriate. However, the HCCI combustion process can accept a variety of fuel types. Since ignition occurs in an HCCI engine by auto-ignition of the fuel/air mixture, the choice of fuel will have a significant impact on both engine design and control strategies. Both fuel volatility and auto-ignition characteristics are important. As discussed by Epping et al. [78], the fuel must have high volatility in order to easily form a homogeneous charge. Chemically, fuels with single-stage ignition are less sensitive to changes in load and speed. This can ease the requirements on an HCCI control system over a wide range of engine operating conditions. Christensen et al. [79] studied the relationship between the fuel's octane number, the inlet air temperature and the compression ratio needed to get auto-ignition close to TDC by mixing iso-octane with octane number 100 and n-heptane with octane number 0. The test results show that almost any liquid fuel can be used in an HCCI engine using a variable compression ratio.

In an effort to develop a better understanding of the fuel and engine parameters that govern the start of reaction in an HCCI engine, Ryan and Matheaus [80] performed engine and constant volume combustion bomb experiments. The primary fuel properties relate to distillation and ignition characteristics. Engine testing provided preliminary guidance on the distillation requirements and an indication of the important ignition requirements. The constant volume combustion bomb experiments were performed in an effort to ascertain a new fuel property that will serve to characterize the ignition property of a fuel in an HCCI engine. The influence of the octane number of the PRF on combustion, performance and emissions characteristics of an HCCI engine was investigated by Yao et al. [81]. The test results showed that HCCI combustion can be controlled and the HCCI operating range can be extended by burning different octane number fuel at different engine modes, i.e. burning low/high octane number fuel at low/high load mode respectively. There exists an optimum octane number that achieves the highest indicated thermal efficiency at different engine load. The effects of fuels on HCCI combustion were further investigated by Yao et al. [82]. The fuels include gasoline, PRF and the mixture of PRF and ethanol at different operating conditions. The test results showed that fuel chemistries have different effects on the combustion progress at various operating conditions. It was found that CA50 (crank angle at 50% completion of heat release) shows no correlation with either research octane number (RON) or motor octane number (MON) at the same operating conditions, but correlates well with the octane index (OI) at all conditions. RON is a measure of how resistant the fuel is to premature knocking. It is measured relative to a mixture of octane and n- heptane. MON is a better measure of how the fuel behaves when under load. Its definition is also based on the mixture of iso-octane and n-heptane that has the same performance. OI is correlated with RON and MON and defined as $OI = (1 - K)RON + KMON = RON - KS$, where K is a constant depending upon the engine operating conditions and S is the fuel sensitivity [83]. The higher the OI, the more resistance there is to auto-ignition and the latter is the heat release in the HCCI combustion. The operating range is also correlated with the OI. The higher the OI, the higher IMEP can reach. The HC and CO emissions increase with the increase of OI, but the effect of OI on NO_x emissions is not obvious. However, the OI does not show correlation with auto-ignition and emissions when the mixtures of PRF and ethanol are used at the same operating conditions. Amann et al. [84] tested four fuels in the gasoline boiling range in a constant volume combustion bomb and a variable compression ratio HCCI engine. The overall goal of these experiments was to determine the fuel that provides the widest possible operating range in the engine. The results showed that there may be some improvement if the ignition characteristics were slightly reduced.

The effects of cetane number (CN) on HCCI auto-ignition, performance, and emissions were also investigated by some researchers [85–87]. Szybist et al. [85] found that high CN fuels exhibited a strong low temperature heat release (LTHR) event and no LTHR was detected for fuels with $CN \leq 34$. Furthermore, at advanced combustion phasing, low CN fuels had significantly higher pressure rise rates and higher NO_x emissions than the high CN fuels. Risberg [86] found that the cetane number can describe the auto-ignition quality of diesel-like fuels in HCCI combustion well. Li et al. [87] found that decreasing cetane number in fuels

Table 2
Multi-dimensional CFD with chemical kinetics models.

Types of models	Codes	Fuels	Focuses	Experiment validating
Two-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA and HCT	Iso-octane	Resolving the low temperature regions of the chamber, along the wall and in the ring crevice [42–44]; CO emissions are extremely sensitive to zone resolution, and 10 zones are not enough to provide appropriate resolution [46]	HCCI engine (cylinder pressure, apparent heat release rate, combustion efficiency and HC, CO emissions at different conditions)
Two-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA and HCT	Iso-octane	Developing a 40-zone model to analyse the effect of piston crevice geometry on HCCI combustion and emissions [47]	HCCI engine (cylinder pressure, apparent heat release rate, combustion efficiency and HC, CO emissions at different conditions)
Three-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA and HCT	Natural gas	CFD simulate the exhaust, intake, and compression strokes up to a transition point, whilst a multi-zone, thermo-kinetic code computes the combustion event [48]	HCCI experiment (cylinder pressure at different conditions)
Three-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA and HCT	Iso-octane	Investigate how the handling of mixing and heat transfer in a multi-zone kinetic solver affects the prediction of CO and UHC emissions for HCCI engine [49]	HCCI engine (cylinder pressure, IMEP, peak cylinder pressure, burn duration and HC, CO emissions)
Three-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA and HCT	Natural gas	Effect of RGF distribution on the onset of ignition and the rate of combustion [50]	Refer to the previous study (reference 48)
Three-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA-3 V and HCT	Iso-octane	Analysis the HCCI engine operation at low loads to investigate the sources of HC and CO emissions and the associated combustion inefficiencies [51]	HCCI engine (cylinder pressure, apparent heat release rate, combustion efficiency and HC, CO, OHC, CO_2 emissions at different conditions)
Three-dimensional CFD with multi-zone detailed chemistry kinetics	KIVA-3 V and HCT	Iso-octane	Effect of turbulence on HCCI combustion with two different piston geometries: a low turbulence flat-top piston and a high turbulence piston with a square bowl [52]	HCCI engine (turbulence intensity, cylinder pressure, heat release rate, combustion efficiency and HC, CO emissions at different conditions)
Three-dimensional CFD with detailed chemistry kinetics	KIVA-3 V and Chemkin	Natural gas	the auto-ignition of natural gas injected into a quiescent chamber under diesel-like conditions [53–55]	Diesel-like engine (ignition delay and peak pressure trends)
Three-dimensional CFD with detailed chemistry kinetics	KIVA-3 V and Chemikin	Natural gas, ethanol, iso-octane [57], iso-octane [56,58,59]	Introduced a reaction rate incorporating the effects of both chemical kinetics and turbulent mixing over characteristic timescales [56–58] Turbulence has significant effects on HCCI combustion [59]	HCCI engine (turbulence intensity, cylinder pressure, heat release rate, NO_x emissions at different conditions)
Three-dimensional CFD with detailed chemistry kinetics	CFD and Chemkin (do not give specific CFD code)	Natural gas	Effects of inhomogeneous distributed EGR gas inside the cylinder [60]	HCCI engine (cylinder pressure)
Three-dimensional CFD with reduced chemistry kinetics	KIVA and VECTIS	Iso-octane and PRF	Tested reduced and optimized reaction mechanisms (iso-octane and PRF) and three-dimensional CFD codes for real engine operation [69]	Shock tube (ignition delay), HCCI engine (cylinder pressure with and without EGR, heat release rate and NO_x emissions)
Three-dimensional CFD with reduced chemistry kinetics	Star-CD and Chemikin	DME/methanol	To investigate the combustion and emissions formation of dimethyl ether (DME) and methanol dual fuel HCCI engine [70]	HCCI engine (cylinder pressure)

significantly reduces smoke emission due to an extension in ignition delay and the subsequent improvement in mixture formation. Smokeless combustion, ultra-low NO_x, and efficient operating range with regard to EGR and IMEP are significantly extended by decreasing fuel cetane number.

The primary conclusion from these experiments is that the best test fuel based on overall observations was medium octane fuel. In addition, the presence of aromatic compounds appears to control the reaction rates during HTR. Therefore, it appears that the timing of the main reaction and reaction rate can be controlled by optimizing the fuel composition.

2.2.2. Effects of additives and fuel modification

Some chemical components have the ability to inhibit or promote the heat release process of auto-ignition. Therefore, HCCI auto-ignition can be controlled by modifying the fuel so that it is more chemically reactive or inhibitive by adding an ignition promoter or inhibitor. These effects can go back to the Leppard [88] and Westbrook [89] who used numerical additives to influence auto-ignition causing engine knock, which is nearly identical to the kinetics of HCCI ignition.

Ricklin et al. [90] reported that for a natural gas-fuelled HCCI engine, even a small amount of NO_x present in a trapped charge may be important in the HCCI operation. The large reactivity difference between compressed natural gas (CNG) and DME can result in a wide range of HCCI timing control [91]. Aceves et al. [92] gave a numerical evaluation of fuels and additives for HCCI combustion. Their research included a ranking of many potential additives. Additives were ranked according to their ability to advance HCCI ignition. Several additives were identified for advancing combustion by almost 11 crank angle degrees when added to the intake mixture at a concentration of 10 ppm. This is a big effect, equivalent to the increase of the intake temperature by more than 30 K.

In order to further investigate the effects of additives and fuel modification, Shibata et al. [93] investigated the effects of fuel chemical composition on LTHR and HTHR (high temperature heat release). In their research, twelve fuels were tested. The base fuel is a mixture of equal parts of 11 pure hydrocarbons and another 11 test fuels include 15vol% of one component and 8.5vol% of each of the remaining ten components. The results indicated that the n-paraffins, such as n-heptane, hasten the HTHR reaction and aromatics (except benzene), naphthenes and olefins have a mechanism that reduces the LTHR of their mixture counterparts. The structure of hydrocarbons strongly influences the tendency to exhibit LTHR. The similar results were also found by Farrell and Bunting [94]. They also found that all-paraffinic (normal, iso, and cycloparaffins) fuel exhibited distinctly earlier combustion phasing, increased rate of cylinder pressure rise, and increased rate of maximum heat release compared to the indolene reference fuel. Conversely, olefin-containing fuels exhibited retarded combustion phasing. It suggests that the effects on combustion phasing and engine operability range may need to be considered in the practical implementation of HCCI for fuels with large compositional variations. Shudo and Ono [95] proposed an HCCI system fuelled with a mixture of DME and methanol-reformed gas, both of which are produced from an on-board methanol reformer utilizing exhaust heat. A high system thermal efficiency over a wide range of operating equivalence ratio has been demonstrated. Methanol-reformed gas effectively controls the timing of the second stage of heat release by high temperature reactions in the DME HCCI operation. This allows an expansion of the operable range of equivalence ratio and engine load. Furthermore, it is also found that both hydrogen and CO in the reformed gas have the positive effect of retarding the onset of the second-stage heat release. DME and methanol dual fuel combustion mode was also investigated by Yao et al. [96]. Both DME and methanol were injected into the intake manifold. HCCI combustion can be obtained in dual fuel mode, resulting in ultra-low NO_x emissions and high thermal efficiency. Like the research of Shudo et al., the major advantage of this combustion system is that the ignition timing and combustion rate can be controlled by adjusting the relative proportions of DME and methanol, and both the low and high loads in the operating range of HCCI engine can be extended through this system. In addition, other fuels have also been used in the HCCI engines, such as ethanol [97,98], liquefied petroleum gas [99,100], and biodiesel [101,102], and diethyl ether (DEE) [103].

EGR is also a specific strategy of fuel modification, and the addition of EGR into intake is the most practical means of controlling charge temperature in an HCCI engine. It has been well confirmed that hot EGR enhances combustion in four-stroke HCCI engines mainly due to the high temperature of the resulting intake mixture, rather than the existence of active radicals. In addition to the thermal effects, the inserted gases contained in the EGR can be used to control the heat release rate due to its impact on chemical reaction rates, which can delay the auto-ignition timing, reduce the heat release rate, and thus lower peak cylinder pressure. External EGR works well due to its simplicity. But its thermal effect is limited due to the heat loss in the EGR system and slow response during transient operation.

The effects of cooled EGR on combustion, performance and emission characteristics in HCCI operation region were also investigated by Yao et al. [104,105]. The results indicate that the EGR rate can broaden the HCCI operating region, but it has little effect on the maximum load of the HCCI engine fuelled with DME/methanol. In contrast, for HCCI combustion fuelled with PRFs, EGR can delay the ignition timing, slow down the combustion reaction rate, reduce the pressure and average temperature in cylinder and extend the operation region to higher loads. The optimum indicated thermal efficiency of different octane number fuels appears in the region of high EGR rate and large fuel/air equivalence ratio (according to the different octane numbers, φ~1.3–2), which is close to the boundary of knocking region. The Ryan [106] reviewed the effect of the EGR on the HCCI combustion processes. The results show that increasing EGR will retard the combustion timing. The EGR also tends to moderate the heat release rate, and is regularly used during high load operation.

2.3. Extension of HCCI concept – stratification combustion

A great deal of HCCI research has revealed that the process involved in mixing fuel and air has important effects on the HCCI combustion process. With HCCI, the start of combustion is dictated by auto-ignition chemical kinetics. Thus controlling the combustion phasing requires tuning of the auto-ignition kinetics, which is affected by the charge composition and the pressure and temperature histories of the reactants during the compression process. In fact, it is impossible to get an absolutely homogeneous mixture in the operation of practical HCCI engines. A little inhomogeneity in fuel concentration and temperature appearing in mixing can produce significant effects on the auto-ignition and combustion process. Therefore, solving the HCCI control problems has led to the investigation of various control strategies that may move away from homogeneous mixtures.

With homogeneous charge, 10% of the fuel can exit the cylinder unburned [107]. Consequently, one could assume that this amount of the fuel does not contribute to pressure rise. If one imagines that all of the fuel was supplied via direct injection to the cylinder with less fuel in the quenching zones, the operational equivalence ratio in the combustion zone would rise relative to the homogeneous charge operation. Thus it would be possible to reduce the amount of the fuel residing in the quenching zones using charge stratification. Consequently, the fuel economy could be improved and the HCCI operating range can be expanded. Richter and co-workers [108] performed engine imaging experiments to assess the magnitude and role of inhomogeneities in HCCI operation. They concluded that charge inhomogeneities were potentially significant and play an important role in the combustion process.

In the limit of homogeneous reactants and adiabatic combustion, ignition timing and pollutant emissions would be governed solely by chemical kinetics. As one moves away from this idealization, turbulence and turbulence/chemistry interactions (TCI) play increasingly important roles. Both experimental [109] and computational [59,110,111] studies have shown that turbulence and TCI can influence ignition timing and emissions for practical HCCI engines. It can be concluded from the above analysis that the auto-ignition and combustion processes are related to turbulence and inhomogeneity of charge and temperature in HCCI combustion. That is, auto-ignition and combustion processes can be effectively controlled by controlling turbulence and inhomogeneity of charge and temperature.

In the research of Thirouard et al. [112], mixture inhomogeneity was investigated on a single cylinder gasoline engine equipped with optical access. Single cycle observations of the fuel distribution and the combustion process shows that the location of the first auto-ignition zones is strongly correlated with the position of the fuel-rich areas. The results also show that the potential for air/fuel mixture quality management offered by multiple direct injections can be used to control the phasing of auto-ignition in CAI combustion, and then used to extend the operating range and improve fuel economy. Dec and Sjöberg [113] concluded that sufficient sensitivity is observed only for fuels that exhibit obvious low temperature heat release (like diesel fuel) in reducing pressure rise rate and prolonging combustion duration. In contrast, the timing of the hot ignition is relatively insensitive to the local equivalence ratio for fuels that the low temperature heat release is not distinct (like typical gasoline). Their recent research [114] explores the potential of partial fuel stratification to smooth HCCI heat release rates at high load. The results indicate that partial fuel stratification has potential to increase the high load limits for HCCI operation. Multi-zone chemical kinetics modelling indicates that partial fuel stratification enabled an increase of IMEPg (gross indicated mean effective pressure from computed results) from 537 to 597 kPa. With the introduction of partial fuel stratification there is a risk of creating regions with too high fuel concentration and combustion temperatures, which could lead to unacceptable NO_x emissions. The effects of charge stratification on combustion and emissions have also been investigated using coupled multi-dimensional CFD and a reduced n-heptane chemical kinetics model by Yao and Zheng [76]. Seven different kinds of imposed stratification have been introduced according that the position of maximal local equivalence ratio appears along the radial direction of the cylinder at IVC. In the first type of imposed stratification, the concentration of fuel–air is the strongest in the centre of the cylinder and leanest along the cylinder wall; in the last type of imposed stratification, the concentration of fuel–air is the leanest in the centre of the cylinder and strongest near the cylinder wall (the concentration of fuel–air along the cylinder wall was assumed as 0 in order to accord with the operation condition of practical engine); in other types of imposed stratification, the concentration of fuel–air increases first then decrease and the positions of maximal local equivalence ratio are different in every type of imposed stratification. It is by different positions of maximal local equivalence ratio, the auto-ignition process and emission formation are affected. Consequently, the combustion process is controlled. The results show that the charge stratification results in stratification of the in-cylinder temperature. The former four kinds of stratification, for which the maximal local equivalence ratios are located between the cylinder centre and half the cylinder radius, advance ignition timing, reduce the pressure rise rate and retard combustion phasing. But three kinds of stratification, for which the maximal local equivalence ratios appear between half the cylinder radius and the cylinder wall, have little effect on the cylinder pressure. All seven kinds of stratification can reduce unburned fuel emissions; the four kinds of stratification increase the CO and NO_x emissions, whilst the last two kinds of stratification can reduce unburned fuel, formaldehyde and CO emissions and maintain low NO_x emissions simultaneously.

It can be concluded from the above research that utilizing the inhomogeneity is an important path to achieve clean and high efficiency combustion in engines. Consequently, advanced control strategy of fuel/air mixture is more important than simple "homogeneous charge" in the control of HCCI combustion processes. Control of charge thermal stratification, concentration and components is the key to achieve clean and high efficiency combustion in engines. Charge stratification is mainly controlled by advanced fuel injection techniques and turbulence control such as in the MULINBUMP (Multi-In-jection and BUMP Combustion Chamber) compound combustion system proposed by Su and the CSI combustion system proposed by AVL which will be introduced later.

Though stratification has the potential to extend the HCCI operation range to higher loads, it is not enough for diesel engines because high loads, and even full loads, are often required for this type of engine. The rapid increase of NO_x emissions for stratified HCCI combustion limits the extension of the operating range to high loads. A new combustion mode is needed for clean and high efficiency combustion of diesel engines at high loads. It is well known that the main challenge for diesel engines is reducing the emissions of NO_x and PM simultaneously. In fact, the amount of both soot and NO_x emissions from a diesel engine is dependent on local temperature and mixture in the cylinder. The effect of temperature on soot formation has been investigated in Ref. [115]. At high flame temperatures, polycyclic aromatic hydrocarbons (PAH), which are considered soot precursors, are oxidized, instead of forming species that transform into soot. The maximum soot concentration can be found at intermediate flame temperatures, which are ideal for both the formation of PAH and their transformation to soot particles. In low temperature flame, the oxidation of PAH is less effective and the production of PAH is higher than at intermediate temperatures. The temperature, however, is too low to induce the coagulation of PAH into tar and the subsequent transformation of tar into soot. In addition, the researches of soot formation and oxidation in combustion also can be founded in Bockhorn [116] and Tree and Svensson's work [117].

Unlike soot, NO_x is formed at high temperatures due to the high activation energy of the $O + N_2 = NO + N$ reaction in the Zeldovich mechanism [118]. Alriksson and Denbratt [119] calculated the $\varphi - T$ map for soot and NO concentrations using the Senkin code with a surrogate diesel fuel consisting of n-heptane and toluene for a homogenous mixture. The results suggest that the local combustion temperature should be kept below approximately 2200 K to avoid high NO concentrations for low equivalence ratios. At high equivalence ratios, it becomes necessary to further decrease the maximum allowable temperature to avoid soot formation. Both the NO and soot formation areas are completely avoided regardless of the equivalence ratio if the temperature is kept below approximately 1650 K. This concept is referred to as low temperature combustion (LTC). In addition, many HCCI researches indicate that the oxidation rate from CO to CO_2 becomes very low if the temperature is below 1400 K [120]. The ideal combustion region is shown in Fig. 6.

In conventional diesel combustion engines, one way to reduce the temperature during combustion is to use EGR. This results in a reduced local flame temperature during combustion. Unfortunately, the reductions in oxygen flow rate and local flame temperature cause an increase in particulate emissions. Smokeless high equivalence ratio combustion has been investigated by Sasaki et al. [121] in a direct injection, 2 l, 4-cylinder passenger car engine with a compression ratio of 18.6. They reduced the combustion temperature to suppress soot formation using large amounts of cooled EGR. Smoke emissions increased with decreasing air/fuel ratios until a critical point was reached, below which smoke emissions decreased rapidly. Increasing the EGR caused increases in HC and CO emissions as well as in fuel consumption. However, these were only considered serious concerns in the fuel-rich region. Therefore, LTC utilizes high levels of dilution to reduce overall combustion temperatures and to lengthen ignition delay. This increased ignition delay provides time for fuel evaporation and reduces inhomogeneities in the reactant mixture, thus reducing NO_x formation from local temperature spikes and soot formation from locally rich mixtures.

The possibilities of using low temperature combustion in order to reduce both soot and NO_x emissions have also been studied by Alriksson et al. [122]. At 25% engine load, the use of very high EGR levels resulted in low soot and NO_x emissions. LTR was detected in the rate of heat release analysis at all operational conditions with sufficiently large amounts of EGR, which gave rise to longer ignition dwells (end of injection to start of combustion) and low soot formation. The LTR seems to have a significant role in prolonging the ignition delay before high temperature combustion and thus in extending the time available for mixing. However, at these operating conditions, fuel consumption and emissions of CO and HC were increased. A long ignition dwell is important because injecting into a flame must be avoided in order to achieve low temperature combustion. In the follow-up research, they investigated possibilities to extend the range of engine loads in which soot and NO_x emissions can be minimized by using LTC in conjunction with

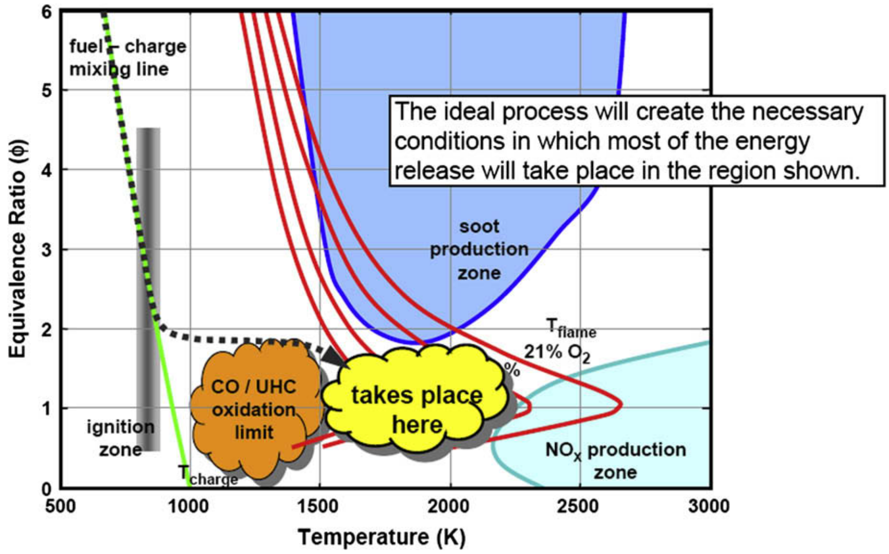

Fig. 6. Ideal combustion region which escapes soot, NO_x and CO/UHC production zones [120].

high levels of EGR [119]. The research engine was a 2-l direct injected, supercharged, heavy duty, single cylinder diesel engine and the amount of EGR was increased by adjusting the exhaust back pressure whilst keeping the charge air pressure constant. The results show that very low levels of both soot and NO_x emissions can be achieved at engine loads up to 50% by reducing the compression ratio to 14 and applying high levels of EGR (up to approximately 60%).

It can be concluded from these pioneering researches that the EGR-diluted LTC mode can reduce NO_x and soot emissions simultaneously. It is not difficult to imagine that LTC has the potential to the extend operation range to high loads by proper control strategies, such as high injection pressure, to improve mixing of fuel and air and high boost to ensure sufficient air intake.

In reference to the above discussions on inhomogeneous combustion and LTC, the meaning of HCCI has been greatly enriched with deeper and deeper research, and hence the broader subject area is often referred to as "generalized HCCI combustion" with many combustion concepts appearing under this heading, such as SCCI (Stratification Charge Compression Ignition), PCCI (Premixed Charge Compression Ignition) and LTC, etc.

2.4. The development on the application of optical diagnostics

Optical diagnostics has been used in the study of HCCI since its discovery in 1979. Noguchi et al. [8] set up a spectroscopic system to investigate the initiation and development of this new combustion process. Subsequently, with the development of HCCI, more optical diagnostic have been applied to study this new combustion process. In this section, simple theoretical descriptions are used to introduce the principle of the optical technique that has been used in HCCI diagnostics. Experimental considerations are given for the application of these techniques to HCCI or other engines. Much attention is devoted to providing readers with appropriate references to the optical diagnostic technical literature, which will be of further assistance to readers interested in a particular application of each optical technique in HCCI combustion.

2.4.1. Optical diagnostics for in-cylinder mixture formation

For HCCI combustion by port fuel injection, optical diagnostics is often used to investigate the inhomogeneity in fuel distribution and temperature in the pre-mixture and their effects on the ignition and combustion process. Richter et al. [108] investigated the imaging of fuel and OH using planar laser induced fluorescence (PLIF) in an HCCI engine. Two different premixing procedures were used to obtain different degrees of homogeneity of the fuel/air charge. In the first setup, a standard port injection was used to create the charge. In the second setup, an additional preheated mixing tank of 20 l was used to prepare a more homogenous charge. The PLIF confirmed that the fuel preparation strategy influences the fuel/air homogeneity at the time of combustion. It was found that the charge inhomogeneity has a modest effect on the spatial variations of the combustion process. Iida et al. [123,124] investigated the effect of charge inhomogeneity in fuel distribution on the HCCI combustion process. Two-dimensional images of the chemiluminescence were captured by using a framing camera with an optically accessible engine. DME was used as a test fuel. By changing a device for mixing air and fuel in the intake manifold, inhomogeneity in fuel distribution in the pre-mixture was varied. Chemiluminescence images with different mixing process are shown in Fig. 7.

In fact, more optical diagnostics for observing in-cylinder mixture formation has been focused on the direct injection of fuel into the cylinder. As mentioned above, many recent studies suggest some more practical ways, such as SCCI, PCCI, and LTC for increasing the high load limits for HCCI. All of these need this direct injection.

Kashdan et al. [125] investigated the in-cylinder mixture distribution in an optically accessible, direct injection HCCI diesel engine. Planar laser induced exciplex fluorescence (PLIEF) imaging was used in this study, which allowed qualitative visualization of the mixture (liquid and vapour phase) distribution within the piston bowl through the use of exciplex-forming dopants. In this study, the start of injection (SOI) was -40°ATDC, liquid fuel typically appears 2° crank angles later. The liquid fuel impinges on the piston face whilst the corresponding vapour phase images acquired at -33°ATDC show a certain degree of fuel stratification with a fuel-rich region appearing at -30°ATDC as a result of fuel impingement. Such trends would be further exacerbated for even later SOI timings closer to TDC.

Musculus [126] investigated the in-cylinder spray, mixing processes for low load, LTC conditions with early fuel injection in an optical heavy-duty diesel engine. Laser-elastic/Mie scattering

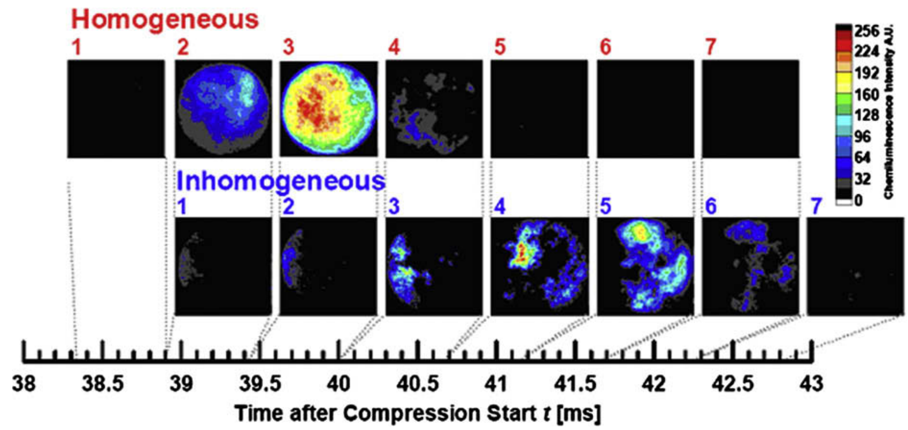

Fig. 7. Chemiluminescence images with different mixing process ($\varphi = 0.36$; 8 images per 1cycle can be taken at a maximum with framing streak camera; the imaging intensifier open time period is 0.55 ms that is proper for about 2degree at the 600 rpm) [124].

showed liquid fuel penetration and fuel fluorescence indicating the leading edge of the vapour jet. Instantaneous images of liquid fuel elastic-scattering and fuel fluorescence are shown in Fig. 8, with the naturally aspirated condition on the left, and the low-boost condition on the right. The elastically scattered light from the liquid fuel is false-colored in blue, and the fuel fluorescence is false-colored green. The maximum extent of the liquid fuel ranges between 45 and 50 mm for the naturally aspirated condition, and 40–45 mm for the low-boost condition. Note that some weak scattering is also visible downstream of the strong scattering in some images. Based on observations from this study, the most likely source of weak scattering in the head of the jet is small liquid fuel droplets. The small droplets of the heavy ends of the fuel do not vaporize until cool flame heat release, for low-density, low temperature conditions of the current study. However, the liquid fuel typically penetrates only about 25 mm (the "liquid length") under conventional diesel conditions, with regions downstream of 25 mm having only vaporized fuel [127,128].

Fang et al. [129] investigated the liquid spray evolution process using Mie scattering in a High speed direct injection (HSDI) diesel engine with a narrow angle injector (70 degree). Keeping the IMEP constant, the main injection timings are -40, -60 and -80° ATDC respectively. With later injection timing at -40° ATDC, the liquid spray tip impinges on the bowl wall, in the corner at the outer region of the conic extrusion in the bowl, due to shorter fuel penetration. But for -60° ATDC, due to cooler air and lower density, spray penetrates deeper. The impinging point is close to the bowl lip. For -80° ATDC, the liquid spray will impinge on the piston top. Some fuel will develop downward along the bowl curvature and some would bounce back and collide with the cylinder liner. The fuel liner collision will lead to fuel leakage to the crankcase without combustion and results in reduced fuel economy and lubricating oil dilution. Fuel impingement results in fuel film deposition with pool fires found in the later crank angles. Pool fire leads to incomplete combustion with soot formation. Finally, the Table 3 showed the studies on the mixture formation of HCCI engines, which includes the diagnosis methods, excited fuels, detection system, and the application focuses.

In summary, it can be concluded that the early pre-TDC injection strategy helps to form a more uniform air–fuel mixture before ignition than the conventional diesel combustion. However, due to the lower air temperature and density, fuel impingement on the bowl wall will occur with an early injection strategy. To minimize the over-penetration and wall wetting that can occur when fuel is injected well before TDC, the optimization of injector (such as a narrow angle injector), chamber shape (such as a BUMP chamber), and air management (such as the EGR ratio, higher boost pressure) is also very important for the new combustion models. Detailed control strategies of improving mixing rates of fuel and air will be discussed in next section.

2.4.2. Optical diagnostics for combustion process

2.4.2.1. Chemiluminescence imaging and spectral analysis. As stated in the book by Glassman [131], chemiluminescence often starts from low temperature combustion due to the relaxation of excited combustion radicals to their ground states, which indicates the start of exothermic chemical reactions and heat release. Generally speaking, natural flame emissions from conventional diesel combustion involve two aspects: chemiluminescence and soot luminosity. For diesel combustion, chemiluminescence often comes from the visible and near ultraviolet bands due to OH, CH, CH_2O and C_2 radicals [132]. Although chemiluminescence occurs just after the SOI, the signal level is quite weak. An intensified charge coupled device (ICCD) camera is always used to capture these early non-luminous flames. It has also been noticed that chemiluminescence exists for the whole combustion process, but it is overwhelmed by strong radiation from the luminous flame after soot is generated within it. The soot luminosity in the gasoline direct injection (GDI) engine is also very strong and effectively drowns out chemiluminescence from other interesting species produced in the combustion process. Furthermore, this soot signal is not a good measure for tracking the actual flame front because it only represents areas that are burning rich.

A similar problem can also be found in spectral analysis, which has been used for many years for in-cylinder diagnostics. However, due to strong black body radiation from soot particles, the signal-to-noise ratios are usually too low to detect specific species. Most research involved with spectral analysis has been applied to the study of SI engines or low soot fuels, e.g. DME. But for HCCI combustion, they can permit very low soot emissions. Therefore, the spectral analysis will be more suitable in research on HCCI combustion. In this section, chemiluminescence imaging and spectral analysis will be introduced. For soot luminosity, optical diagnostics will be introduced in the next section.

Kawahara et al. [133] investigated HCCI combustion with dimethyl ether in a single cylinder engine using chemiluminescence spectral analysis. Results show that emitted light from HCHO appears at LTR in accordance with Emeléous' cool flame bands. During the main heat release, the CO–O recombination spectrum was strong. There was a strong correlation between the

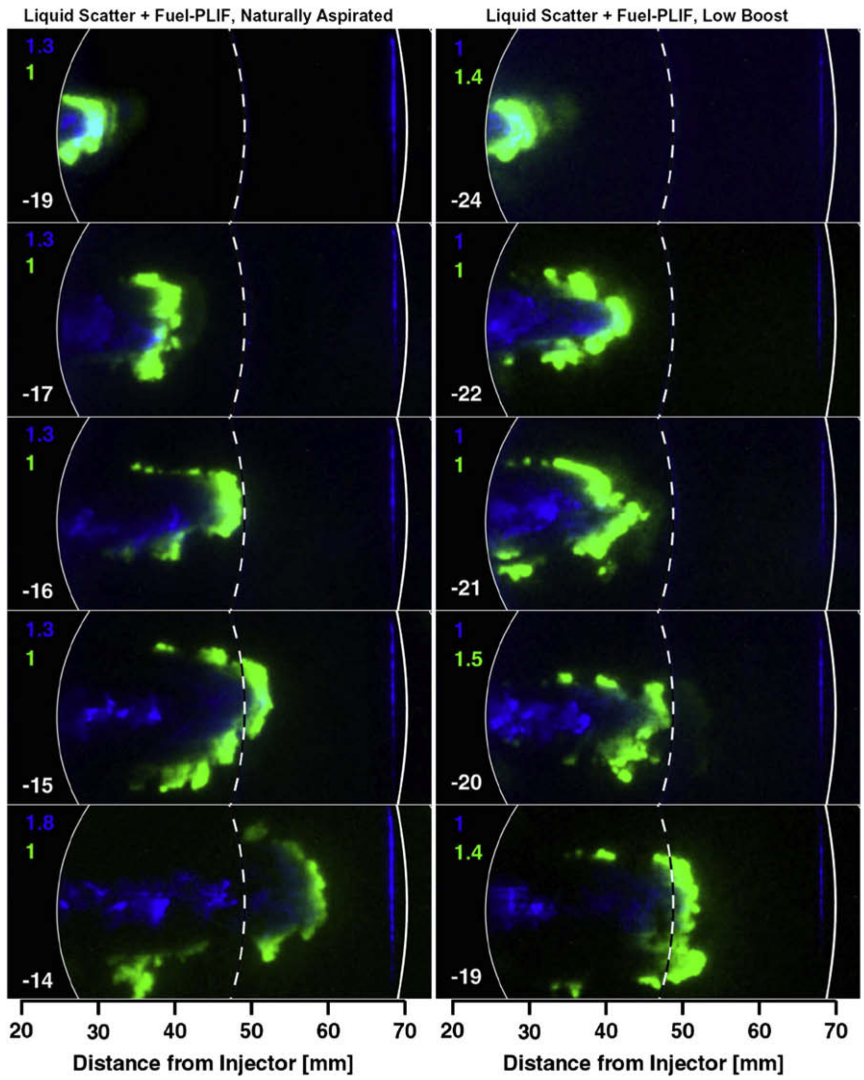

Fig. 8. Liquid fuel (blue/or gray without color) and vapor fuel perimeter (green/or white without color) for naturally aspirated (left) and low-boost (right) conditions (the dashed is the edge of piston bowl-rim) [126].

rate of heat release (ROHR) and the CO–O recombination spectrum. Augusta et al. [134] investigated the effects of different engine parameters on the chemiluminescence characteristic of HCCI combustion using a spectroscopic diagnostic system. It was determined that different engine parameters affect the ignition timing of HCCI combustion without altering the reaction pathways of the fuel after the combustion has started. A strong correlation was found between the chemiluminescence light intensity and the

Table 3
Optical diagnostics for in-cylinder mixture formation.

Diagnosis methods	Excited fuels	Detection system spectral filters	Application and reference
Fuel-PLIF (Nd:YAG laser/532 nm; dye laser/289 nm)	n-Heptane (65%);acetone(35%)	ICCD camera; a cut-off filter and a WG 305 color filter	Imaging of fuel with different premixing procedures in a HCCI engine using port injection [108]
Chemiluminescence imaging	DME	Intensifier in framing streak camera	Effect of charge inhomogeneity in fuel distribution on the HCCI combustion process using port injection [123,124]
PLIEF imaging (Nd:YAG laser/355 nm)	n-Decane; 1-methyl –naphthalene; tetramethyl –phenyldiamine (TMPD)	ICCD camera; A dichroic mirror	Visualization of the mixture (liquid and vapor phase) distribution in a direct injection HCCI diesel engine with the high pressure common-rail (CR) injection [125]
Laser-elastic/Mie scattering (Nd:YAG laser/532 nm); Fuel-PLIF (optical parametric oscillator (OPO); and Nd:YAG laser/ 355 nm)	Diesel	ICCD camera; Mie scattering: a 532-nm center wavelength bandpass filter (10-nm spectral width)	The in-cylinder spray, mixing processes in low temperature combustion conditions (12.7% charge oxygen) with SOI = −22° ATDC in an optical heavy duty diesel engine [126]
Mie scattering/Copper vapor laser using an optic fiber LIEF-Fuel (Nd:YAG laser 355 nm)	Diesel for Mie tetradecane/ 90%; naphthalene/9%; TMPD/1%	High speed digital video camera; ICCD camera	Visualization of the spray evolution process (liquid and vapor phase) in a direct injection LTC diesel engine with a narrow angle injector (70 degree) [129]
Mie scattering Fuel–PLIF (Nd:YAG laser 266 nm)	PRF50/90%; 3-pentanone/10%	High speed camera. ICCD camera	Visualization of the spray and mixture distribution using two high pressure GDI-type injectors in an automotive HCCI engine during low load, stratified operation [130]

rate of heat release. Mancaruso et al. [135] investigated the auto-ignition and combustion of HCCI in a diesel engine. By means of the common-rail system, the quantity of fuel was split into five injections per cycle. UV–visible imaging and spectra showed that the presence of HCO and OH were homogenously distributed in the chamber. Since the process is widely dominated by the presence of OH radical in the chamber, it seems that this radical contributes to the reduction of PM in the cylinder. OH could be a suitable tool to identify the start of high temperature combustion and phase the rate of heat release. Dec et al. [136] investigated the naturally occurring charge stratification in an HCCI engine using chemiluminescence imaging. The images showed that combustion is not homogeneous but has a strong turbulent structure even when fuel and air are fully premixed prior to intake. The inhomogeneities are caused primarily by thermal stratification due to heat transfer during compression, combined with turbulent transport. Therefore, chemiluminescence imaging shows that all real HCCI engines have some naturally occurring charge stratification even when fuel and air are well mixed. This stratification is critical to the high load operating limit of these engines.

Persson et al. [137] investigated the early flame development in spark assisted HCCI combustion using high speed chemiluminescence imaging. The results show that even for large negative valve overlap (NVO, resulting from the early close of the exhaust valve and late open of the intake valve to modulate residual gas fraction), and thus high residual fractions, it is a growing SI flame that interacts with, and governs the subsequent HCCI combustion. Using spark timing, it is possible to phase the combustion timing even when the major part of the released heat is from HCCI combustion. Berntsson and Denbratt [138] investigated the effect of charge stratification on the combustion process and emissions from HCCI combustion. Port injection was used for the main fuel supply to create a homogeneous charge, and a stratified charge was obtained by a special injector which is designed as an air-assisted, spray-guided direct injection device. Direct imaging of the combustion shows that the initial sequence of main heat release is staged and prolonged with stratification. The charge is initially ignited near the centre of the cylinder because the cold cylinder wall introduces a temperature gradient with lower temperature near the wall. The heat release rate analysis and imaging show that the duration of combustion is increased due to staged combustion caused by the local variation of equivalence ratio prolonging heat release. Kook and Bae [139] investigated a diesel-fuelled PCCI combustion technique using a two-stage injection strategy in a single cylinder optical engine equipped with a common-rail fuel system. The results show a luminous flame only in heterogeneous combustion regions of the second injection. Finally, they concluded that the main injection timing should be advanced earlier than $-100°$ ATDC for homogeneous and non-luminous combustion. PCCI combustion showed reduced NO_x by more than 90% but increased fuel consumption and HC, CO emissions compared with direct injection diesel engine. Kanda et al. [140] investigated PCCI combustion with early injection in a common-rail diesel engine and highly efficient EGR cooling. The in-cylinder visualization results at different injection timings are shown in Fig. 9. When the injection timing was advanced (at $-35°$ and $-45°$ATDC), the main diffusive combustion featured non-luminous flames. After the main combustion, luminous flames were observed in the neighbourhood of the piston bowl wall. After long ignition delay, a large portion of the fuel formed a premixed mixture and burnt with no soot, but some portion formed a rich fuel–air mixture on the fuel wall-film and burnt with soot.

The research on charge or thermal stratification through some active methods has shown that the stratification can reduce maximum rates of heat release and its phasing capabilities and thus may provide scope to increase the HCCI operating range. It was also been found that ignition of the premixed gas could be controlled by a second injection when the earlier injection maintained LTR. Therefore, charge and thermal stratification are effective methods to control the HCCI combustion.

Chemiluminescence analysis can also be used to investigate the combustion process of LTC as investigated by Kook et al. [141]. In their research, fresh air was diluted with additional N_2 and CO_2, simulating 0–65% EGR in an engine. Images of natural combustion luminosity exhibit no soot luminosity until late in the premixed burn period. The overall soot luminosity is decreased and becomes more uniform with increasing dilution. They concluded that a mixing-controlled combustion phase remains important under highly dilute operating conditions to achieve high fuel efficiency and low CO emissions. Upatnieks and Mueller [142,143] investigated the influence of nitrogen dilution and charge-gas temperature on in-cylinder combustion processes and engine-out NO_x and smoke emissions. Engine-out measurements of NO_x and smoke emissions and in-cylinder images of natural luminosity were obtained for charge-gas oxygen concentrations from 9% to 21%. The results show that soot incandescence can be negligible for fuel-rich local mixture stoichiometries that would result in intense soot incandescence under undiluted operating conditions, as shown in Fig. 10 [144]. Fig. 10 shows that the flame colour of LTC is blue due to very low flame temperatures. Finally, the two results all demonstrate that low temperature combustion with near-zero engine-out smoke and NO_x emissions can be achieved using a traditional direct injection strategy.

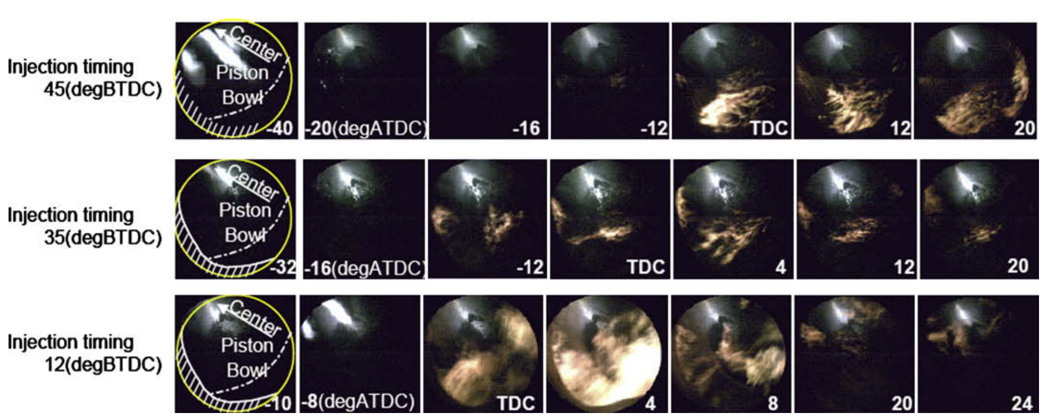

Fig. 9. Effect of injection timing on flame behavior (Engine speed = 1500 rpm; injection quantity = 19 mm^3/st; injection pressure = 70Mpa; without EGR) [140].

Using a high speed digital video camera, Fang et al. [145] investigated combustion processes in an HSDI diesel engine employing different injection strategies. For early pre-TDC injection strategies, a two-stage low temperature combustion heat release rate pattern was observed including HCCI combustion and fuel film combustion (pool fire). For multiple injections, very high soot combustion left over from the first injection was observed with pool fires during the second injection process. By further advancing the first injection and retarding the second injection, the pool fires from the first injection can be eliminated. From Fang's PhD thesis [129], EGR was used to realize LTC, leading him to conclude that simultaneous reduction of soot and NO_x can be achieved by increasing the EGR rate and retarding the injection timing. For clean combustion modes, such as PCCI or LTC, longer ignition delays are observed, due to a lower ambient air temperature, and the air–fuel mixture starts combusting with a low flame temperature which inhibits the soot formation process at the beginning and also extends the time for air–fuel mixing without soot formation. Therefore, the clean combustion mode shows premixed combustion dominant heat release patterns. Finally, as the Table 3, Table 4 showed the chemiluminescence imaging and spectral analysis of the HCCI engines, which also includes the diagnosis methods, excited fuels, detection system, and the application focuses.

2.4.2.2. Planar laser induced fluorescence (PLIF) diagnostics. Laser induced fluorescence (LIF) is the emission of light from an atom or molecule following excitation by a laser beam. The early development of LIF methods was driven by single-point measurements, but the application of LIF to IC engines has been dominated by multi-point planar imaging. PLIF may be viewed as a modern form of flow visualization. In common with methods such as schlieren and shadowgraph, PLIF is extremely useful for qualitative or semi-quantitative characterization of complex flow fields, providing spatially resolved information in a plane rather than integrated over a line-of-sight. The wide application of PLIF in IC engines is due to the strength of the fluorescence process, relative to its primary competitors, Rayleigh and Raman scattering, and to its species specificity [152]. In addition, PLIF imaging can reveal the presence of the intermediate species formaldehyde, allowing the temporal and spatial detection of auto-ignition precursors prior to the signal observed by chemiluminescence in the early stages of the cool flame.

In HCCI combustion of hydrocarbons, formaldehyde (HCHO) and OH radical are important intermediate species and active radical as seen in chemical kinetics analysis. Therefore, the ignition and combustion processes can be studied by visualizing their distributions.

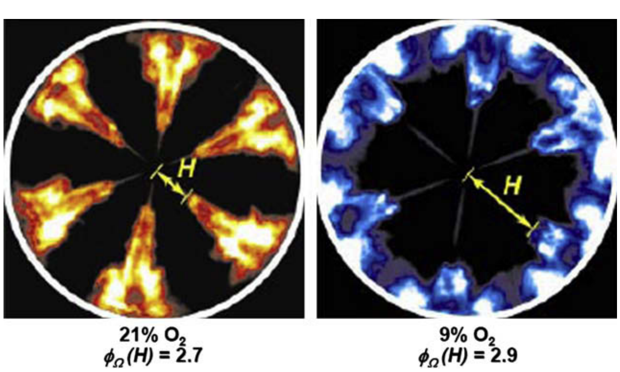

Fig. 10. The imaging of conventional diesel and LTC (Engine speed = 1200 rpm, nitrogen dilution to simulate the EGR, using the DGE as fuel) [144].

Simultaneous OH and formaldehyde LIF measurements have been performed on an HCCI engine with a port fuel injection system by Collin et al. [153]. A blend of iso-octane and n-heptane was used as fuel. At the start of the LTR, clouds of formaldehyde are detected. More formaldehyde is detected as the LTR progressed. After the LTR ends, formaldehyde fills the entire measured region. The PLIF signal of OH is detected only in regions where formaldehyde is absent. Maximum OH concentration is detected when most of the fuel is consumed and thus close to the peak in-cylinder temperature. These results can be seen in Figs. 11 and 12. Särner et al. [154] investigated the simultaneous LIF imaging of formaldehyde and a fuel tracer in a direct injection HCCI engine, using a mixture of n-heptane and iso-octane as fuel and toluene as a fluorescent tracer. Images from both early and late injection and at all crank angle degrees show close spatial resemblance between toluene signal area and formaldehyde signal area. The work shows that formaldehyde is a feasible alternative to traditional fuel tracers for visualizing fuels featuring LTR in HCCI combustion. Zhao et al. [155] investigated the HCCI/CAI combustion process in a single cylinder optical engine. The auto-ignition and combustion processes of PRFs were studied using the two-dimensional PLIF technique as well as heat release analyses. The formaldehyde formed during the low temperature reactions of HCCI/CAI combustion is also visualized by a PLIF system. The formation of formaldehyde is more affected by charge temperature than by fuel concentration, but its subsequent burning or the start of main heat release combustion takes place in those areas where both the fuel concentration and charge temperature are sufficiently high. The research shows that the OH and HCHO are complementary markers of the HCCI combustion process.

For both HCCI with stratification and LTC combustion processes, PLIF diagnosis is also widely applied. Seyfried et al. [156] utilized high speed laser diagnostics for single-cycle-resolved studies of the fuel distribution in the combustion chamber of a truck-size HCCI engine. Two different piston geometries, one flat and one with a square bowl, were used in this research. The results for the flat piston geometry reveal multiple ignition kernels and a gradual oxidation of the bulk charge through distributed reactions. For the square bowl piston, there are, instead, very few isolated ignition kernels. Furthermore, the progression of subsequent combustion shows remarkable similarities to conventional flame front propagation. Finally, they thought that the temperature stratification should be the main reason for the extended burn duration of the square bowl chamber. Hildingsson et al. [157] investigated simultaneous formaldehyde and OH-PLIF in an optical light duty diesel engine running with different injection strategies: UNIBUS (Uniform Balky Combustion System, detailed information and references about UNIBUS is presented in section 3.1.2), direct injection HCCI and port injection HCCI fuelled with n-heptane. The results show a clear difference in OH and formaldehyde formation among UNIBUS, direct injection HCCI and port injection HCCI. The formaldehyde formation always starts at the beginning of heat release for LTR in all combustion strategies. The rise of formaldehyde signal level at LTR is steepest for the port injection HCCI and slowest for UNIBUS because port injection provides a more homogeneous charge than UNIBUS and direct injection HCCI. Formaldehyde presence in UNIBUS is longer than in port injection HCCI. Berntsson et al. [158] investigated using OH-PLIF in the NVO of a spark assisted HCCI combustion engine in an optical single cylinder engine. Data acquired from corresponding optical analysis showed the occurrence of OH radicals (and thus high temperature reactions) during the NVO in all tested operating conditions. The results also indicate that the extent of the high temperature reactions is influenced by both the amount of the pilot injection and the total amount of fuel because decreasing the relative amount of the

Table 4
Optical diagnostics for combustion process using chemiluminescence imaging and spectral analysis.

Diagnosis methods	Excited fuels	Detection system spectral filters	Application and reference
Chemiluminescence spectra and imaging	PRF35	ICCD and streak camera; center wavelength 308 nm and 431 nm (FWHM: 10 nm) for OH and CH; center wavelength 560 nm, FWHM: 9 nm for C_2	HCCI combustion process and the chemical intermediate products (OH, CH, C_2) using port injection [146]
Chemiluminescence spectra and imaging	DME	High speed video camera with an image intensifier; spectrometer, quartz fiber, and ICCD	HCCI combustion process and the chemical intermediate products (OH, CH, C_2) using port injection [133]
Chemiluminescence spectra	Iso-octane	Imaging spectrograph and the fiber	Effects of different engine operating parameters on the chemiluminescence spectra of HCCI combustion [134]
UV-visible imaging and spectra	Diesel	CCD camera spectrograph-ICCD	The auto-ignition and combustion of HCCI in a diesel engine with high pressure CR injection system [135]
Chemiluminescence imaging	PRF50	ICCD camera	The effects of speed, load, swirl and injection strategy on the boundary layer of HCCI combustion [147]
Chemiluminescence imaging	Iso-octane	ICCD cameras; high speed intensified CMOS video camera	The naturally occurring charge stratification in an HCCI engine [136]
High speed chemiluminescence imaging	Ethanol 90%; acetone 10%	High speed camera with a image intensifier	How geometry generated turbulence affects the ROHR in an HCCI engine [148]
High speed chemiluminescence imaging	Ethanol 40%; n-heptane 60%	High speed framing camera with a image intensifier	Early flame development in spark assisted HCCI combustion [137]
Chemiluminescence imaging	PRF50	High speed CCD camera	Effects of charge stratification on combustion process and emissions of an HCCI engine using pressurized air to inject in the cylinder and break up the fuel spray for obtaining a stratified charge [138]
Natural luminosity imaging	Diesel	High speed video camera	Diesel-fueled PCCI combustion technique using a two-stage injection strategy equipped with a CR fuel system [139]
Chemiluminescence imaging	n-Heptane	ICCD camera	Possibility of controlling HCCI combustion with charge stratification using both inlet air heating and residual gas trapping (Internal EGR) [149]
Chemiluminescence imaging	SP95 (Euro95)	High speed intensified CCD camera	Effects of mixture quality on CAI combustion with separated intake ports and the natural symmetric tumble motion; NVO and burned gas re-breathing; direct injection [112]
Natural luminosity imaging	Diesel	Endoscope	PCCI combustion luminous flames imaging with early injection in a CR diesel engine and highly EGR cooling [140]
Natural luminosity imaging	Diesel	High speed video camera	Effects of EGR rate, injection timing, and swirl ratio on the combustion and fuel conversation efficiency of LTC in a high speed diesel engine [150,141]
Natural luminosity imaging and spectrum	Diesel and H_2	High speed framing drum-camera; monochromator and ICCD	Late injection low temperature diesel combustion in a rapid compression machine (RCM) with a small bowl combustion chamber, high swirl and high EGR condition and small amount of premix hydrogen gas [151]
Natural luminosity imaging	Diethylene glycol diethyl ether (DGE);	ICCD camera [142]; High speed (1 MHz bandwidth) large area visible photoreceiver [143]	Influence of nitrogen dilution and charge-gas temperature on in-cylinder combustion processes and flame lift-off lengths [142,143]
Natural luminosity imaging	Diesel and bio-diesel	High speed digital video camera	Combustion processes of LTC employing different injection strategies in a diesel engine with a narrow angle injector [145]

pilot injection or increasing the total amount of fuel lead to larger amounts of OH radicals.

Musculus et al. [159] investigated the over-mixing and unburned hydrocarbon emissions in EGR-diluted LTC diesel engines by PLIF of a fuel tracer (toluene) The optical diagnostic images show that the transient ramp-down at the end of fuel injection produces a low-momentum, fuel–lean mixture in the upstream region of the jet, which persists late in the cycle. At LTC conditions with long ignition delay, this mixture likely becomes too lean to achieve complete combustion, thereby contributing to UHC emissions. Therefore, stagnant, fuel–lean regions that form during the ignition dwell after the end of injection are a likely, significant source of UHC emissions for EGR-diluted LTC diesel engines.

These researches show that the combustion process of generalized HCCI can be better understood by optical diagnostics; therefore, such technologies have significant meaning in optimizing the control strategies of generalized HCCI combustion. Finally, Table 5 showed the planar laser induced fluorescence diagnostics of the HCCI engines, which also includes the diagnosis methods, excited fuels, detection system, and the application focuses.

2.4.3. Optical diagnostics for emission formation

Due to the very low level of emissions of NO_x and PM over the HCCI operating range, research on these aspects is scant. However if fuel stratification is introduced using in-cylinder direct injection, the emissions of NO_x and PM will increase. Thus, study of NO_x and PM formation is necessary so that in the new combustion mode these emissions can be better controlled. In fact, optical research mainly focuses on the NO and soot emissions.

2.4.3.1. NO_x optical diagnostics. PLIF imaging of NO for engine applications has been developed and applied over the last decade to conventional CI or SI engines and GDI engines by many researchers [168–171], leading to the development of the theory of NO formation. For example, Dec and Canaan [168] investigated the NO-PLIF in a conventional diesel. The images show that NO is not produced by the initial premixed combustion (which is fuel-rich), but begins around the jet periphery just after the diffusion flame forms. NO formation then increases progressively throughout the remainder of the combustion event. The data also shows that NO formation continues in the hot post-combustion gases after the end of the apparent heat release, for the operating conditions studied. However, in HCCI combustion mode, NO emissions are very low

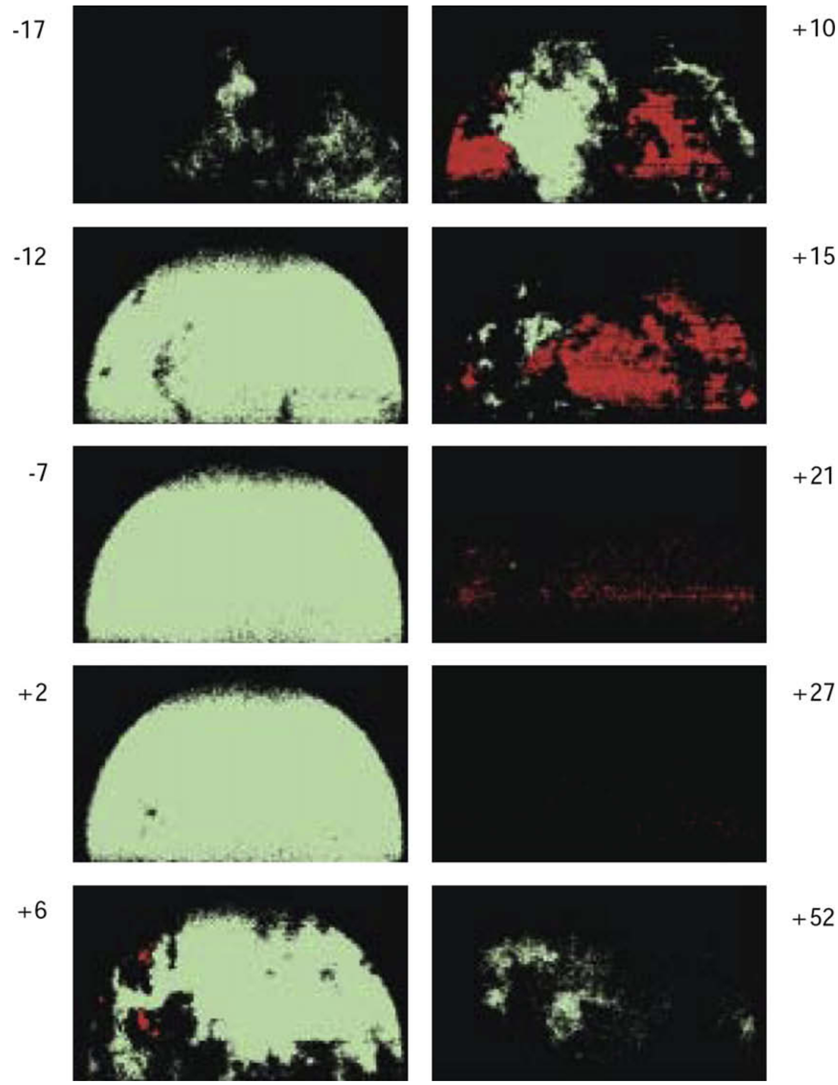

Fig. 11. Single-shot images from onset of LTR combustion until the end of the main combustion. Formaldehyde is shown in green (or white without color) and OH is shown in red (or gray without color) [153].

due to low combustion temperatures, which restrict NO formation. For this reason, there is little research on NO-PLIF of HCCI engine.

However, Zilwa and Steeper [172,173] have made emission predictions for HCCI engines using LIF fuel distribution measurements. The emissions include CO_2, CO, HC and NO_x. The method is based on the simplifying premise that each individual fuel–air packet burns as if in a homogeneous mixture at the same equivalence ratio. The relative success of the predictions indicates a strong correlation between in-cylinder charge distribution and engine-out emissions. In particular, it encourages the formulation of ideal fuel distributions to guide the development of advanced charge-preparation strategies.

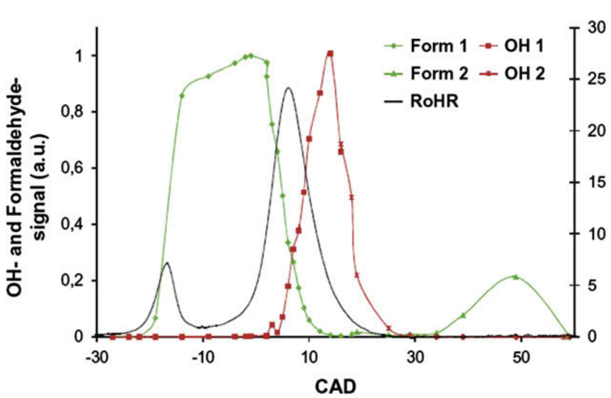

Fig. 12. Average rate of heat release and average OH- and HCHO-signals for the measurement series shown in Fig. 11 [153].

2.4.3.2. Soot optical diagnostics. Due to the premixed combustion, the soot emission in HCCI and conventional SI engine can be negligible. However, when charge stratification is introduced in HCCI through in-cylinder injection, early or late in the cycle, or in a GDI engine, soot emissions will not be negligible under some operating conditions. In this section, the focus will be on soot formation in the new combustion mode, especially for PCCI and LTC. More in-depth and detailed reviews about the soot formation in conventional diesel engine can be found in Tree and Svensson's work [117].

The study of soot generally involves the use of physical probing and optical techniques. In the last few years, rapid development and application of optical diagnostics in obtaining soot measurements have been seen in conventional diesel engines. Optical techniques for soot measurement in diesel engines include the two-colour

Table 5
Optical diagnostics for combustion process using planar laser induced fluorescence diagnostics.

Diagnosis methods	Excited fuels	Detection system spectral filters	Application and reference
OH-PLIF (Nd:YAG/532 nm; dye laser/ 289 nm); HCHO-PLIF (Nd:YAG/355 nm)	PRF50	Two ICCD cameras; OH: two 283 nm long-pass filters, one UG11 filter; HCHO: two GG395 Shott-filters	Simultaneous OH- and HCHO-LIF measurements in an HCCI engine [153]
Simultaneous HCHO-PLIF (Nd:YAG laser/355 nm) and the fuel tracer (Nd:YAG laser/ 266 nm)	Iso-octane and n-heptane, toluene as a tracer	Two ICCD cameras; a beam splitter (high reflection in the band 260-320 nm and high transmission above 400 nm); fuel tracer: a long-pass filter; HCHO: two GG395 and one UG5 Shott-filter	Simultaneous PLIF imaging of formaldehyde and a fuel tracer in a direct injection HCCI engine using early or late injection [154]
HCHO-PLIF (Nd:YAG laser/ 532 nm; tunable dye laser/ 355 nm)	n-Heptane; PRF30; PRF60	ICCD camera	Formaldehyde PLIF in a CAI engine using different PRF [155]
Simultaneous high speed fuel tracer PLIF (four Nd:YAG lasers/ 266 nm) and chemiluminescence imaging	Iso-octane 90% or ethanol 90% acetone 10%	A high speed framing cameras with eight independent ICCD; a long-pass filter (335 nm); a short-pass filter (470 nm)	Correlation between the consumption of fuel tracer and chemiluminescence intensity in HCCI combustion processes [160]; fuel distribution and combustion process in different combustion chambers of an HCCI engine [156]
High speed HCHO-PLIF (four Nd:YAG lasers/355 nm)	n-Heptane	A high speed framing cameras with eight independent ICCD; a long-pass filter, GG385; a short-pass filter with a cut-off wavelength at 500 nm	Formaldehyde visualization in HCCI combustion processes [161]
HCHO-PLIF (Nd:YAG laser/ 355 nm) OH-PLIF (Nd:YAG laser/532 nm; dye laser/ 283 nm)	n-Heptane; PRF75	ICCD cameras; HCHO: a GG385 long-pass filter and a short-pass filter at 450 nm; OH: a 305 nm long-pass filter (WG305) and a UG5	Formaldehyde and OH-PLIF in an HCCI engine using a simulated EGR rate of 10% and 50% [162]
HCHO-PLIF (Nd:YAG laser/ 355 nm) OH-PLIF (Nd:YAG laser/532 nm; dye laser/ 282.93 nm)	Dodecane fuel	ICCD cameras and a high speed image intensified camera OH: a band pass filter, (FWMH:16 nm) centred on 312 nm and a 358-nm low pass filter; HCHO: a 465 nm bandpass filter (FWMH: 70 nm) and a 435 nm high-pass filter	Effects of chambers shape, injection strategies, EGR ratio, and air–fuel ratio on the two-stage HCCI combustion processes using high pressure CR direct injection system [125,163]
Simultaneous HCHO-PLIF (Nd:YAG laser/355 nm) OH-PLIF (Nd:YAG laser/532 nm; dye laser/283 nm)	n-Heptane	Two ICCD cameras; a beam splitter, one with a filter for the OH signal and the other with a filter for the formaldehyde signal.	Simultaneous formaldehyde and OH-PLIF in an optical light duty diesel engine running with different injection strategies: UNIBUS, direct injection HCCI and port injection HCCI with 20% EGR [157]
HCHO-PLIF (Nd:YAG laser/ 355 nm) Fuel tracer (Nd:YAG laser/266 nm) OH-PLIF (Nd:YAG laser/532 nm; dye laser/283 nm)	PRF85 3-pentanone added as a fuel tracer;	ICCD cameras, a narrow band pass filter centred on 310 nm for OH	Distribution of fuel vapor, formaldehyde, OH in an HCCI engine use of NVO in combination with an SI stratified charge through fuel injected in pilot and main injections timing and spark-assistance [158,164]
OH-PLIF (Nd:YAG laser/ 532 nm; OPO/284 nm); fuel tracer PLIF	Diesel; toluene as a tracer	ICCD camera; OH: a 312 nm center wave length band pass (FWHM: 16 nm), a short wave pass filter with a near 358 nm, and a 2-mm thick WG305 long wave pass	Distribution of OH in the combustion processes of early injection LTC in a heavy-duty diesel engine [126]; Distribution of fuel of LTC [159]
Partially oxidized fuel–PLIF (Nd:YAG laser/355 nm)	PRF30, and a small amount of lubricity improver	ICCD camera two Schott GG385 and one GG395 long-pass filters	Evolution of bulk flow structures and their influence on the spatial distribution of partially oxidized fuel in a swirl-supported, direct injection diesel engine with a highly dilute (12% O_2) [165]
HCHO-PLIF (Nd:YAG laser/ 355 nm) Natural luminosity spectrum	n-Heptane	ICCD cameras, a beam splitter; HCHO: a long wave pass filter (GG375); Spectrum: a 410 nm band pass filter (FWHM: 33 nm)	In-cylinder unburned hydrocarbon visualization during low temperature CI engine combustion [166]
OH-PLIF (Nd:YAG laser/532 nm; OPO/284 nm); PLIF of HCHO and PAH (Nd:YAG laser/ 355 nm) Fuel tracer (Nd:YAG laser/266 nm) chemiluminescence spectrum	n-Heptane 71% iso-octane 29% toluene as a fuel-tracer	A 450 nm cut-off dichroic beam splitter, with three filters to isolate OH signal: a 312-nm band pass filter (FWHM:16 nm), a 358 nm short-pass filter, and a WG 305 long-pass filter; The 450 nm dichroic beam splitter transmitted visible light to a second broadband beam splitter, a CG385 long-pass filter and a 408-nm band pass filter (FWHM:40 nm) for HCHO/ PAH PLIF; An 4-nm resolution spectrograph with a CG375 long-pass filter	Low temperature combustion process from fuel–lean to fuel-rich reaction zones in late injection diesel engine using three different injector nozzles with different angles [167]

method, the light extinction method for soot concentration measurement, and laser induced incandescence (LII). A more detailed presentation on optical diagnostics for soot can be found in Zhao and Ladommatos' review [174], the book of ENGINE COMBUSTION INSTRUMENTATION AND DIAGNOSTICS by Zhao and Ladommatos [175], the Dec' review on advanced compression–ignition engines [176], the quantitative measurements of soot by Snelling et al. [177], and a review on the soot LII by Schulz et al. [178]. Indeed, such diagnostics are still used in generalized HCCI combustion to obtain soot measurements.

Singh et al. [179] and Huestis et al. [180] have investigated soot formation and destruction processes applying two-colour optical pyrometry in LTC. In particular, Singh et al. [179] presented a comparison of LTC with conventional diesel combustion conditions. The intake stream of LTC strategies was diluted with a metered flow of nitrogen gas to achieve 12.6% oxygen by volume. Soot thermometry and high speed soot luminosity imaging show that the low temperature operating conditions have lower in-cylinder soot volume and temperature than the high temperature conditions. Irrespective of conditions, the onset of in-cylinder soot occurs during the mixing-controlled combustion phase after the premixed burn. As the amount of soot decreases, the radiation heat loss also decreases drastically. For LTC conditions, there is enough time available for fuel to penetrate and mix with the air such that soot-producing combustion occurs primarily near the edge of the bowl, as compared with high temperature combustion conditions where soot is formed farther upstream in the fuel jet. Huestis et al. [180] investigated the in-cylinder and exhaust soot mass in an optical diesel engine under various intake oxygen concentrations from 8 vol% to 21 vol%. EGR in the optical engine was also simulated by dilution of intake gases with nitrogen. The results show that in-cylinder soot temperatures decreased with the reduction of in-cylinder oxygen concentration. With EGR, peak in-cylinder soot initially increased as intake oxygen concentrations were reduced from 21% to 12%. At lower intake oxygen concentrations, peak in-cylinder soot decreased. Trends in late cycle, in-cylinder soot mass measurements agreed well with exhaust soot mass measurements. At moderate EGR rates, lower in-cylinder temperatures reduced soot oxidation more than formation, leading to higher engine-out soot levels. At very high levels of EGR, in-cylinder soot formation rates became very low, such that engine-out soot was also low, even with very low oxidation rates.

Musculus [126] investigated the soot luminosity and soot laser induced incandescence measured development of in-cylinder soot in an early injection low temperature combustion. First, a single-channel PIN-10D large area (100 mm^2) silicon photodiode from united detector technologies (UDT) operating in photoconductive mode collected the soot luminosity transmitted through the large piston-crown window. This non-imaging diagnostic provided a measurement of the crank angle-resolved cylinder-averaged soot luminosity. The second diagnostic provided images of the soot luminosity for examining the spatial development of soot within the cylinder. The soot luminosity is normally many orders of magnitude brighter than chemiluminescence, requiring significant attenuation of the light delivered to the intensified camera. While combustion-heated in-cylinder soot is the strongest source of natural broadband luminosity (i.e., incandescence), soot heated by laser irradiation can be significantly hotter, and thus can emit even more strongly. Such LII for detection of soot is a fairly mature diagnostic. When the laser beam is formed into a thin sheet, the resulting soot incandescence originates from a thin sheet, which can be imaged using an intensified camera with appropriate spectral filters. The in-cylinder soot was heated by the 100-mJ frequency-doubled output of a 10-Hz pulsed Nd:YAG laser (532 nm), formed into a thin sheet. A short wave pass (SWP) filter with a cut-off near 450 nm helped to reject background natural soot luminosity, and a 532-nm notch filter rejected elastically scattered light. The results show that the soot formation is nearly always observed only in regions that are deficient in OH. Thus, soot and OH are not expected to persist in the same regions. Indeed, in previous OH-PLIF and soot LII studies of conventional diesel combustion in this facility, early in the combustion event, OH was typically confined to a thin sheet on the periphery of the soot cloud [181,182]. Even later in combustion, when the OH is more broadly distributed, soot and OH regions are distinct and do not overlap spatially [183]. That is, OH and soot generally did not persist within the same regions. Indeed, OH is one of the primary oxidizers in soot models, and such oxidation consumes both OH and soot at their interface. At this threshold-sooting condition, in both the soot luminosity and soot LII images, the soot is first observed far downstream near the cylinder liner, at the head of the jet. As the jet continues to penetrate and develop, the strongest luminosity and soot LII are typically found on either "side" of the jet, in a region within what is typically described the "head vortex" at the front of transient or "starting" jets, as shown in Fig. 13. In fact, this upstream soot luminosity is attributed to the head vortex soot of adjacent impinging jets, rather than to any upstream soot formation in the horizontal jet. This observation is in contrast with typical soot formation for conventional diesel conditions, for which soot is formed farther upstream, and throughout the jet cross section [184], as shown in Fig. 14. For conventional diesel combustion conditions, soot was still formed with the head vortex, but it was also formed within upstream regions of the jet cross section. Thus, for this threshold-sooting condition, upstream soot formation essentially has been eliminated, as the white dotted circle in Figs. 13 and 14, but soot formation within the head vortex is still observed. In other words, soot formed within the head vortex at the tip of the diesel jet is the most difficult to avoid by enhancing mixing or extending the pre-combustion mixing time.

Fang et al. [185] investigated the transient in-cylinder late cycle soot distribution in LTC conditions using the backward illumination light extinction (BILE) technique through side windows. With this technique, the soot evolution process can be obtained for late cycle crank angles for one continuous combustion cycle because it is not a destructive measurement like LII. A more detailed discussion of the principles and setup of light extinction can be found in Ref [129,185].

In inhomogeneous generalized HCCI combustion, soot is formed much farther downstream, only at the tip of the jet, in the head vortex or near the edge of the bowl. Injection parameters, such as injection pressure, injection timing and times, etc., and the EGR ratios, have a strong effect on the soot emissions. Therefore, the optimization of injection parameters and EGR ratio is needed in furthering LTC research. The most difficult aspect is how to avoid soot forming within the head vortex at the tip of the diesel jet by enhancing mixing or extending the pre-combustion mixing time.

3. Evolution in control strategies of diesel-fuelled HCCI engines

In diesel HCCI combustion, it is difficult to prepare homogeneous mixture because of the lower volatility, higher viscosity and lower resistance to auto-ignition of diesel fuel. First, elevated temperatures are required before significant vaporization occurs, making it easier to form a premixed homogeneous charge. Second, diesel fuel has significant cool-combustion chemistry, leading to rapid auto-ignition once compression temperatures

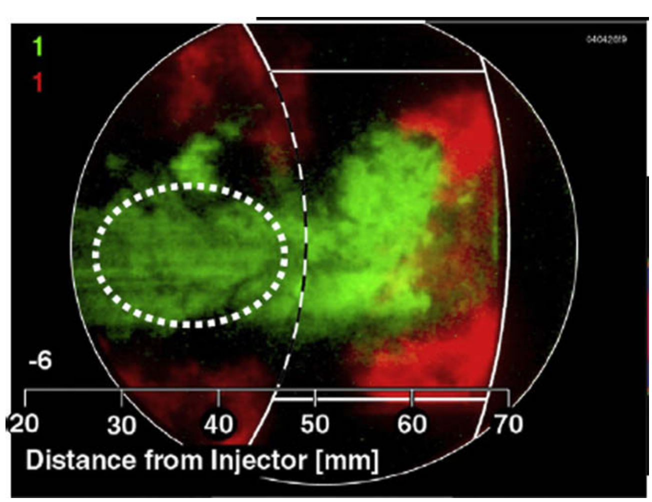

Fig. 13. OH (green, OH-PLIF, or white without color) throughout jet cross section, with soot (red, soot luminosity, or gray without color) only at head of jet in LTC mode [126].

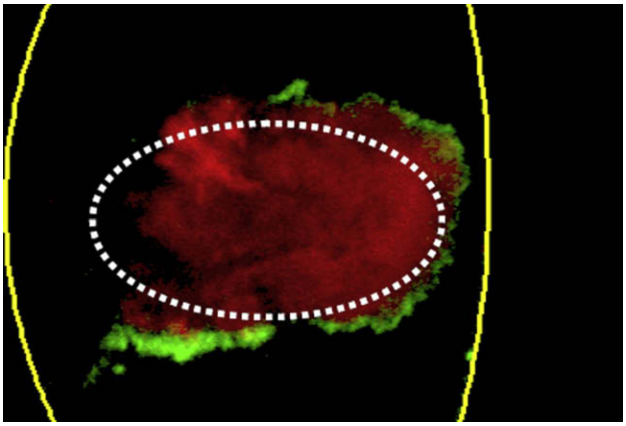

Fig. 14. OH (green, OH-PLIF, or white without color) in thin envelope surrounding soot (red, soot LII, or gray without color) throughout the jet cross section [182].

exceed about 800 K [186]. This can lead to overly advanced combustion phasing and high combustion rates. Accordingly, the essential factor needed to achieve diesel HCCI combustion is mixture control, including both charge components and temperature control in the whole combustion history and high pre-ignition mixing rates.

As mentioned in the introduction, port fuel injection has been used in early investigations of diesel-fuelled HCCI by Ryan et al. and Gray et al. [11,12]. Christensen et al. [79] investigated diesel-fuelled HCCI with port fuel injection as part of an investigation of variable compression ratios in controlling HCCI with various fuel types. The results are in general agreement with those of Ryan et al. and Gray et al. In addition, they found that smoke emissions were significant for some conditions, and although NO_x emissions were very low, they were not as low as those with gasoline. These trends were thought to be due to poor vaporization of diesel fuel creating an inhomogeneous mixture. Furthermore, in order to avoid knock, compression ratios need to be reduced in HCCI combustion with port fuel injection, which leads to the deterioration in engine economy. Therefore, at present, this approach is used in combination with other means. In this section, various kinds of typical generalized diesel-fuelled HCCI combustion concepts and combustion systems are introduced.

3.1. Early direct injection HCCI

Because of the disadvantage of port fuel injection, an early direct injection approach towards achieving diesel-fuelled HCCI has been investigated. Compared with premixing in the intake port, this approach offers three potential advantages [187]. 1) By injecting the fuel in the compression stroke, the higher in-cylinder temperatures and densities can help vaporize the diesel fuel and promote mixing. This allows cooler intake temperatures, reducing the propensity for early ignition. 2) With a carefully designed fuel injectors, the possibility exists to minimize fuel wall wetting that can cause combustion inefficiency and oil dilution. 3) In principle, only one fuelling system is required for both HCCI and conventional diesel operation. The main disadvantage of early DI for HCCI is that it is easy to produce wall wetting due to over-penetration of the fuel. Finally, it should be noted that controlling combustion phasing is still a critical issue for early-DI HCCI because injection timing does not provide an effective means of directly controlling combustion phasing as in conventional diesel combustion. Several different methods of early-DI HCCI have been investigated, including dual-injection techniques that combine early-DI HCCI with a conventional diesel injection. Many combustion systems of early direct injection HCCI have been developed such as PREDIC (premixed lean diesel combustion)/MULDIC (multiple stage diesel combustion), UNIBUS and MULINBUMP, which are reviewed next.

3.1.1. PREDIC

One of the more significant efforts in early-DI HCCI is the work conducted at the New ACE Institute in Japan. Initial reports of this work by Takeda et al. [188] and Nakagome et al. [189] discussed various fuel injection strategies and results show that it is possible to achieve very low NO_x and smoke emissions with this technique. However, similar to diesel-fuelled HCCI with port fuel injection, HC emissions were high and fuel consumption was significantly increased over conventional diesel combustion due to poor combustion efficiency and overly advanced combustion phasing. To minimize the over-penetration and wall wetting that can occur when fuel is injected well before TDC, when in-cylinder air densities and temperatures are low, three injectors were used in their study (one at the centre, and two on the sides) to control the injection timing and quantity of each injector independently. The double injectors mounted on either side of the cylinder, each with two sprays, were positioned so that the fuel sprays would collide in the middle of the chamber to limit wall wetting with very early injection. In addition, the nozzle hole diameter of the centre injector was reduced from 0.17 mm to 0.08 mm. The number of holes was also increased from 6 to 16 and some tests were carried out using a centre injector with 30 holes at various angles. Using this injection method, the engine could be operated with PREDIC. To overcome significant over-advanced combustion, special fuel blends with cetane numbers of 19 and 40 were developed and tested, along with changes in the intake temperature and compression ratio. A subsequent study [17] reported the development of a swirling-flow pintle-nozzle injector that produces a more uniform mixture and reduces wall wetting (as measured by reduced lube-oil dilution), thereby improving combustion efficiency.

In an attempt to achieve higher power densities, this early-DI work was expanded to include a second fuel injection near TDC [190]. This was accomplished by using the dual side-mounted, impinging-spray injectors to produce the lean premixed charge, whilst a centre-mounted injector was used to inject additional fuel near TDC. Approximately half of the fuel was injected in each stage, which was near the maximum amount of premixed fuel that could be used without causing knock. Various combinations of injection timings (for both stages) and fuel CN were used to optimize performance and emissions. Since the second stage burns similarly to conventional diesel combustion, NO_x and PM emissions were much higher with this approach than those obtained with pure HCCI. However, NO_x was reduced to about half that of conventional diesel combustion without a significant increase in fuel consumption. Smoke emissions also decreased, but HC emissions were higher. In order to solve the problem of high HC and CO emissions during HCCI operation, Akagawa et al. [191] investigated the effects of EGR rate and oxygenated fuels on fuel consumption and the effects of the top-land crevice volume on HC and CO emissions and on adhesion of fuel on the walls. They also investigated double injections, the first earlier injection to form a homogeneous and lean mixture, and the second later one for conventional combustion. Much higher loads were found possible whilst achieving low NO_x emissions but this was accompanied by an increase in fuel consumption. Multiple injections were also investigated by Morita et al. [192] and further developed by Nishijima et al. [193], investigating the effects of split injections and water injection. Split injections reduced HC and CO emissions but unfortunately increased NO_x emissions.

3.1.2. UNIBUS

Yanagihara et al. [194] have also applied HCCI combustion with diesel fuel (Toyota UNIBUS). As described by Yanagihara, the engine operates in UNIBUS mode up to about half load and half speed. As with other DI HCCI techniques, mixture preparation is a key issue. Yanagihara et al. used piezo-actuator injectors with pintle-type injector nozzles to reduce the spray penetration in an attempt to limit the over-leaning of the charge. The technique involves a combination of an early injection (around 50° BTDC) and a late injection (about 13° ATDC). The majority of data presented in the paper were taken with 12:1 compression ratio, presumably to avoid overly advanced combustion of the premixed fuel. The investigations examined the impact of the injection timing, level of fuelling, EGR rate and double injections on emissions, torque, instantaneous heat release and load. These results were compared with conventional diesel combustion with the same injector. Low levels of fuelling and low injection pressures were thought to be most appropriate to limit wall wetting and knocking with this particular injector nozzle. A luminous flame is not observed, due to the negligible PM available for oxidation, and confirmed by low PM emissions. The use of EGR is also discussed, and a 60% EGR level is shown to be effective for delaying the HCCI combustion by about 7° crank angle.

A dual-injection strategy has also been developed, with 50% of the fuel introduced in a second injection at 13° ATDC. The second injection significantly improves the combustion efficiency of the early-injected fuel, reducing HC emissions from 5000 ppm to 2000 ppm and CO emissions from about 8000 ppm to 2000 ppm. Therefore, fuel combustion from the second injection can be considered as a trigger for a portion of the heat release from the early-injected fuel. Although the second injection increases NO_x emissions above the near-zero levels for early injection alone, they are still very low for a diesel engine. Further studies [195] were then carried out looking at the possibility of using a second injection closer to TDC to trigger the combustion, which reduced the NO_x benefits but also the fuel consumption penalty.

3.1.3. MULINBUMP

Su et al. [196–198] proposed a compound diesel HCCI combustion system – MULINBUMP, combining premixed combustion with "lean diffusion combustion". The premixed combustion was achieved by the technology of multi-pulse fuel injection. The start of pulse injection, injection pulse number, injection period of each pulse and the dwell time between the injection pulses were controlled. The objective of controlling the pulse injection was to limit the spray penetration of the pulse injection so that the fuel will not impinge on the cylinder liner, and to enhance the mixing rate of each fuel parcel. The last or main injection pulse was set around TDC. A flash mixing technology was developed from the development of a so-called BUMP combustion chamber, which was designed with some special bump rings. The combustion of fuel injected in the main injection proceeds at much higher air/fuel mixing rate than in a conventional DI diesel engine under the effect of the BUMP combustion chamber, which leads to "lean diffusion combustion". The characteristics of auto-ignition and rate of heat release of the premixed fuel of multi-pulse injection were investigated, being one of the more important aspects that affect engine emissions. By advancing the start timing of multi-pulse injection, noisy auto-ignition can be avoided, but over-advancing of the start timing will lead to over-mixing, resulting in higher total hydrocarbon (THC) emissions. The results also show that in the compound combustion mode, the fuel proportion of multi-pulse injection should be high, under the condition of no noisy auto-ignition, which is helpful in the reduction of NO_x emissions. Small amounts of fuel in multi-pulse injection offer no benefits to NO_x emissions reduction, but worsen smoke emissions.

Compound combustion technology has potential in realizing HCCI combustion at rather wide operating conditions of a diesel engine. The basic idea of the MULINBUMP compound combustion system is: 1) at low loads of a diesel engine, there is control of the HCCI combustion process by the multi-pulse fuel injection strategy, which can achieve very low NO_x and PM emissions ($<$10 ppm); and 2) at medium and high loads, the strategy of combining premixed combustion with "lean diffusion combustion" is applied. The mixing rate of fuel and air is improved by combining a high mixing rate combustion chamber with a super high injection pressure, thus achieving rapid mixing. Over the full load range of a diesel engine, the best injection modes can be obtained at different conditions by controlling multi-pulse injection, thereby clean and high efficiency combustion of the diesel engine can be achieved over the full load range. To date, the IMEP of compound combustion has reached 0.93 MPa.

3.1.4. Narrow angle diesel injection

A narrowing of the spray cone angle is commonly used to reduce the fuel wetting on the piston head surface and cylinder wall. The use of a narrow fuel injector nozzle spray cone angle to avoid spray–wall interactions at the early injection timings shows promise for achieving HCCI combustion while maintaining a relatively high efficiency [199–202].

In the research of Kim and Lee [203], two injector nozzles with different spray cone angles (156° and 60°) were used to examine the potential of HCCI combustion. The compression ratio of the test engine was reduced from 17.8:1 to 15:1 to prevent the immoderately advanced ignition of the pre-mixture formed by early injection. The results showed that in the case of the conventional diesel engine, the IMEP was decreased rapidly as the injection timings were advanced beyond 20° BTDC and indicated specific fuel consumption (ISFC) were also deteriorated. In the case of advanced injection timing between 30° and 50° before the TDC, the IMEP was approximately half of that attained under the conventional diesel combustion with the injection timing near the TDC. Injecting the fuel at very early timing helps to create HCCI combustion. However, injecting too early leads to poor fuel evaporation and piston bowl spray targeting issues. At that condition, the fuel–air mixture is formed at the outside of the combustion chamber and the deteriorated combustion efficiency can be expected. Moreover, due to the shifting of combustion event to earlier side, this causes the increase of negative work during compression stroke. These trends are regarded as typical problems of early injection HCCI engine that lower the thermal efficiency and increase the incomplete combustion products such as the HC and CO emissions.

In contrast, the ISFC indicated a modest decrease in the IMEP although the injection timing was advanced to 50–60° BTDC in the case of a narrow spray angle configuration. This reveals that the narrow angle concept was effective in maintaining the high ISFC and IMEP when the fuel was injected at an early timing for HCCI combustion.

In addition, the IFP [199] has developed a combustion system able to reach near-zero particulate and NO_x emissions while maintaining performance standards of the DI Diesel engines, especially in terms of output power and torque. This dual mode engine application called NADI™ (narrow angle direct injection) applies homogeneous charge compression ignition at part load and switches to conventional Diesel combustion to reach high and full load requirements. Therefore, it can be concluded that the narrow spray cone angle injector can reduce the wall wetting problem and avoid an out of bowl injection when the fuel was injected at an early timing for HCCI combustion.

3.2. Late direct injection HCCI

One of the most successful late injection DI HCCI systems for achieving diesel-fuelled HCCI is the MK (modulated kinetics) combustion system developed by the Nissan Motor Company. The principles of this late injection HCCI combustion process are described by Kawashima et al. [204] and Kimura et al. [205]. In order to achieve the diluted homogeneous mixture required for HCCI, a long ignition delay and rapid mixing are required. The ignition delay is extended by retarding the injection timing from 7° BTDC to 3° ATDC and by using high levels of EGR, sufficient to reduce the oxygen concentration to 15–16%. Rapid mixing was achieved by combining high swirl with toroidal combustion-bowl geometry. The operating range for the first generation MK system was limited to about one-third of peak torque and half speed. In the MK mode, NO_x emissions were reduced substantially (to about 50 ppm) without an increase in PM. Combustion noise was also significantly reduced. In addition, it should be noted that with this late-DI HCCI technique, combustion phasing is controlled by injection timing, which is an advantage over port fuel injection and early-DI HCCI techniques.

In the development of a second-generation system, several modifications were made to expand the range of MK operation to higher loads and speeds [206]. Since more fuel must be injected at higher loads, a high pressure common-rail fuel system was used to provide high injection pressures at all speeds to reduce fuel injection duration. The ignition delay was increased by reducing the compression ratio to 16:1 and adding EGR cooling to reduce the intake temperature. To minimize the potential for liquid fuel impingement on the piston bowl wall, the piston bowl diameter was increased from 47 to 56 mm. This change significantly reduced HC emissions under cold-engine conditions. The results show that the second-generation MK system can be used over the entire range of everyday driving, indicating that the engine met the second-generation targets of about half load and three-quarters speed. NO_x emissions are stated to be reduced by 98% compared with conventional operation without EGR, whilst PM emissions are similar to the conventional engine.

More recent work by Kawamoto et al. [207] proposed the use of higher CN fuel, which went against other HCCI operating requirements, but was recommended to reduce HC emissions under cold start conditions. The major advantage of the MK combustion system is illustrated by its ease of implementation because it does not require additional or different hardware. Equally, its operation with conventional hardware did not affect negatively the engine's specific power output. However, the injection timing retard and lower compression ratio lead to a deterioration of cycle efficiency and higher unburned HC emissions.

3.3. Premixed/direct-injected HCCI combustion

In premixed/direct-injected HCCI combustion, port injection was chosen for the main fuel supply to create a homogeneous charge, and direct fuel injection into the cylinder was used to change the concentration and position of local fuel-rich regions with the purpose of controlling HCCI combustion. A homogeneous charge compression ignition diesel combustion (HCDC) system, which is the earliest premixed/direct-injected HCCI combustion, has been proposed by Odaka et al. [208–211]. In their system, most of the fuel was injected into the intake manifold to form a homogeneous pre-mixture in the combustion chamber and the pre-mixture was ignited with a small amount of fuel directly injected into the cylinder. This system can reduce both NO_x and smoke emissions better than ordinary diesel engines. Smoke is reduced near-uniformly as the premixed fuel ratio is increased.

Midlam-Mohler et al. [212–214] have also developed a premixed/direct-injected HCCI system. Port/manifold fuel injection is achieved using a low pressure atomizer system and direct injection is achieved using a high pressure injection system. The atomizer system delivers fuel as a premixed lean homogeneous mixture into the cylinder. At low loads, the main torque is from the homogeneous charge fuel and the direct injection fuel is mainly for ignition; at high loads, the maximum homogeneous charge fuel is used and direct injection fuelling is increased to full load. The basic injection scheme is shown in Fig. 15. Small droplet size (<1 μm mean diameter) allows rapid evaporation during the compression stroke, removing the need for intake air heating. The results indicated that IMEP can be up to 4.7 bar by varying intake conditions in the speed range from 1600 to 3200 rpm. Furthermore, the mixed-mode HCCI combustion can achieve very low NO_x and smoke emissions, less than 4 ppm and 0.02 FSN respectively. Foster et al. [215] presented results pertaining to expanding HCCI operations by charge stratification achieved by both port and direct fuel injection. The stratification of the charge was altered in two ways, (a) by altering the ratio of direct injected fuel to fuel supplied to the intake system and (b) by retarding the injection timing of the DI injector. Stratified charge shows potential as a viable enhancement for HCCI combustion at the lean limit. The combustion becomes more stable with more stratified charge. Berntsson and Denbratt [138] also investigated the control of HCCI combustion using premixed/directed-fuel injection. In their experiments, port injection was used for the main fuel supply to create a homogenous air–fuel mixture. Engine experiments in both optical and traditional single cylinder engines were carried out with PRF50 as fuel. The amount of stratification as well as injection timing of the stratified charge was varied. The maximum rate of heat release depends on the amount of stratification – a larger amount gives a lower rate of heat release but the main heat release is advanced. Varied injection timing results in different phasing of the main heat release. The use of charge stratification for HCCI combustion can lead to a wider operating range, due to its effect on combustion phasing and rate of heat release. Increasing the stratification amount or late injection timing of the stratified charge can lead to an advanced CA50 timing and higher NO_x emissions.

In the research mentioned above, the direct injected fuel is the same as the premixed fuel. In premixed/direct-injected HCCI combustion, these two fuels can also be different, and fuel characteristics can also be used to control combustion. In the system developed by Inagaki et al. [216], gasoline was supplied to the intake air port and diesel fuel was injected directly into the engine cylinder to act as an ignition trigger at timing before TDC. It was

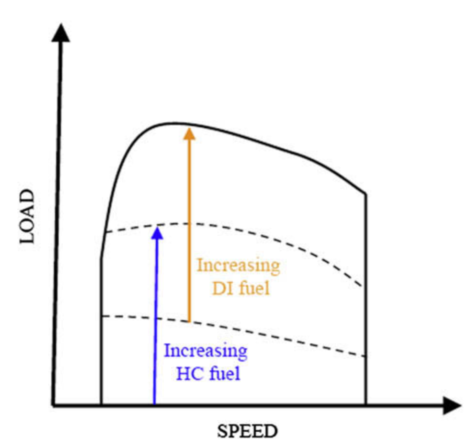

Fig. 15. Basic injection scheme in a premixed/direct-injected HCCI mode [212–214].

found that the ignition phasing of combustion can be controlled by changing the ratios of the two injected fuels, such that combustion proceeds very mildly. Spatial stratification of ignitability in the cylinder prevents the entire mixture from igniting instantaneously. The operable load range, where NO_x and smoke emissions were less than 10 ppm and 0.1 FSN respectively, was extended up to an IMEP of 12 bar using an intake air boosting system together with dual fuelling.

3.4. Low temperature combustion

The principle necessity and feasibility of LTC have already been discussed in section 2.3. In conventional diesel combustion, higher EGR reduces NO_x but increases soot. This is not the case in LTC regimes where increasing EGR reduces both NO_x and soot. In recent years, LTC has attracted more and more attention for its potential to simultaneously reduce NO_x and PM emissions, even at high loads. In consequence, the effects of many control parameters such as injection and intake parameters have been investigated by researchers in order to optimize LTC.

In the recent research of Natti [217], soot and PM characterizations were investigated on a single cylinder HSDI diesel engine under conventional and LTC regimes. Their research indicated that nucleation of soot requires higher temperature, whilst surface growth can proceed at lower temperature in the presence of suitable lower hydrocarbons. The increase of EGR slows down the reactions kinetically and gives the system more time to transfer heat to cooler regions. In conventional combustion, this reduces temperatures just enough to decrease oxidation but not accumulation. Whilst in the LTC regime, the addition reactions are also hindered and therefore the accumulation mode begins to slow down and this reduces soot with increase in EGR.

The effects of injection pressure and swirl on engine-out emissions under LTC were examined by Henein et al. [218] and Choi et al. [219]. Their experiments covered a wide range of injection pressures, swirl ratios and injection timings. In the research of Henein et al. [218], EGR rates were varied to cover engine operations from conventional to low temperature combustion regimes, up to the misfiring point. They found that combustion is very sensitive to small variations in EGR, which are a few percentages from the misfiring EGR limit in the LTC regime. The research of Choi et al. [219] was undertaken at O_2 concentrations of 15%. Increased injection pressure is found to enhance the early mixture formation process, resulting in increased peak of heat release, and generally decreased soot luminosity in all their investigations. Furthermore, there is an optimum swirl ratio beyond which any increase would result in a penalty in greater soot emissions. Trade-off between soot and NO_x at different injection pressures, swirl ratios and EGR rates is shown in Fig. 16.

Aoyagi et al. [220] investigated the effects of a high boost pressure and cooled EGR system on LTC in a single cylinder diesel engine. The authors studied engine performance and exhaust emissions under high air flows into the cylinder and EGR rates of up to 30–40% at full engine load using high boost and turbo intercooled technologies. It was demonstrated experimentally that both NO_x and PM emissions were reduced effectively, with little fuel consumption penalty, by this wide ranged, high boosted, cooled EGR system. Colban et al. [221] also investigated the effect of changes in intake pressure (boost) on engine performance and emissions in LTC regimes in a single cylinder, light duty diesel engine. In their research, moderate (8 bar IMEP) and low (3 bar IMEP) load conditions were tested at intake pressures of 1.0, 1.5, and 2.0 bar. The results indicated that increased intake pressure reduces emissions of UHC and CO, with corresponding improvements in combustion efficiency and ISFC, particularly at high load. Increased intake pressure also reduces peak soot emissions at high load and shifts the peak soot emissions towards lower oxygen concentrations. Soot emissions are reduced with increased intake pressure at high oxygen concentrations, but increased at low oxygen concentrations. Already low NO_x levels are reduced further at high intake pressures, though the influence of intake pressure is small compared with the influence of oxygen concentration.

Effects of piston bowl geometry on the in-cylinder mixing process under typical late injection LTC conditions in a heavy-duty diesel engine were investigated by Genzale et al. [222]. Three piston bowl diameters of 60%, 70% and 80% of the cylinder bore were considered. The data showed that piston bowl diameter influences in-cylinder mixing and pollutant formation processes by altering jet-jet and jet-wall interactions. When the fuel jets impinge on the bowl wall prior to ignition, adjacent jets merge, forming fuel-rich regions where soot formation occurs. By using a larger diameter bowl, wall impingement prior to ignition is reduced and delayed, and mixtures are leaner throughout the jet. However, a greater fraction of the jet becomes too lean for complete combustion. By using a smaller diameter bowl, a strong jet-wall interaction pushes the fuel-rich jet-jet interaction regions into the centre of the chamber, where mixtures are predominantly lean. This reduces net soot formation and displaces fuel-lean regions of otherwise incomplete combustion into the combusting regions near the bowl wall.

In addition, HC and CO emissions characteristics have also been investigated in the LTC regime. The effects of charge dilution on NO_x, soot and CO emissions of LTC were investigated by Kook et al. [150]. The fresh air was diluted with additional N_2 and CO_2, simulating 0–65% EGR in an engine. The results showed that NO_x emissions and soot luminosity increase with increasing flame temperature, whilst high flame temperatures reduce CO emissions due to rapid oxidation. In addition, the fuel conversion efficiency exhibited a maximum at moderate charge dilution levels and mixing remained important if high fuel efficiency and low CO emissions are to be achieved. Opat et al. [223] investigated the HC/CO emission mechanisms in LTC at part load using KIVA-3V Chemkin code. The effect of such variables as rail pressure, swirl number and inlet temperature was explored using statistical experimental designs, first to correlate with the kinetic behaviour of HC and CO, and second to help identify and understand the mechanisms of HC/CO formation and oxidation. This optimal split between fuel entering the squish region and fuel entering the piston bowl provided the most effective use of the limited oxygen

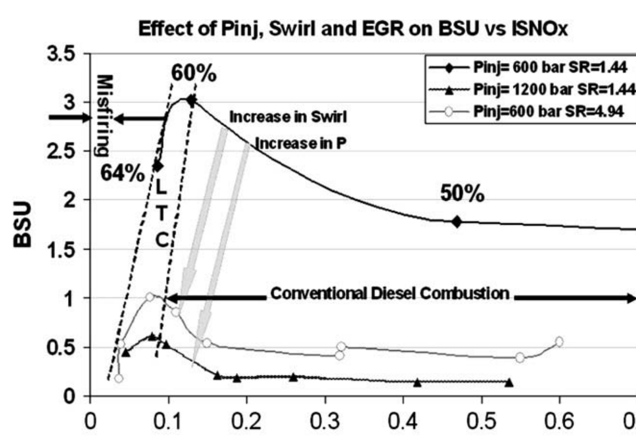

Fig. 16. Trade-off between soot and NO_x at different injection pressures, swirl ratios and EGR rates [219].

available by accessing oxygen from both the bowl and the squish region to burn out CO which would otherwise be unable to find sufficient oxygen. Increasing swirl inhibited mixing with available O_2 in the upper bowl region. Increased injection system rail pressures and intake temperatures reduced the magnitude of the CO. Kumar and Zheng [224] found that the combustion phasing dominated the maximum attainable fuel efficiency of the engine. The use of heavy EGR resulted in a significant drop in combustion efficiency. They investigated the pathways to improve the fuel efficiency of diesel LTC cycles in their recent research. The results indicated that the use of HCCI plus late main injection shows the possibility for better CO oxidation during the combustion process, thereby improved the burning efficiency.

From the above research, it is clear that the key factors that affect EGR-diluted LTC achieving clean and high efficiency combustion are also the factors that improve spray and air/fuel mixing such as fuel injection strategies, high injection pressure and proper swirl ratio. They are consistent with other diesel-fuelled HCCI combustion modes.

3.5. Summary of control strategies for combustion processes in diesel-fuelled HCCI combustion

From the above analysis, it can be concluded that mixture control (mixture preparation) is the key issue in achieving diesel HCCI combustion including charge components and temperature control in the whole combustion history and high pre-ignition mixing rate. There are two types of measures to improve mixture formation: 1) by improving the mixing rate of fuel and air; and 2) by extending ignition delay. Specific control strategies for these two measures are discussed in the following subsections.

3.5.1. Control strategies to improve the mixing rate of fuel and air

3.5.1.1. High pressure/ultra-high pressure injection and small nozzle holes. Increasing injection pressure can greatly increase the energy of fuel injection. Hence, atomization is improved, which leads to an improvement in the mixing rate of fuel and air, whilst reducing the size of nozzle holes increases the relative velocity of the fuel injected into the cylinder and the surrounding air. This can improve the break-up of droplets and accelerate the evaporation of droplets. The effects of high injection pressure and small hole size have been investigated by Dodge et al. [225]. Experiments on a high pressure, high temperature bomb were conducted to evaluate the effect on soot formation of very high injection pressures, to 300 MPa, and very small hole sizes, down to 0.086 mm. Fig. 17 shows the computed time to mix the centreline of the jet to an equivalence ratio of $\varphi = 1.0$ as a function of injector tip hole size, at two different injection pressures. It indicates that the time required in doing this decreases with decreasing nozzle hole diameter. The relation of computed Sauter mean diameters (SMD) versus hole tip diameters is shown in Fig. 18. The mixing parameter and SMD decrease with the reduction of hole size. Overall, high injection pressures and small holes can improve the mixing rates of fuel and air, and especially so for LTC as discussed in 3.4 [218,219].

3.5.1.2. High boost pressure. Enhancing boost pressure leads to an increase of in-cylinder density. And then, adequate atomization of the fuel injected into the cylinder improves the mixing process. Alfuso et al. [226] analysed high pressure diesel spray at high pressure and ambient temperature conditions. Fig. 19 shows the evaporating spray evolution, under 1.2, 3.0 and 5.0 MPa of ambient gas conditions, at 120 MPa of injection pressure. Comparison of the spray images shows that evaporation strongly reduces the liquid droplets dispersion. This effect is more evident at higher in-cylinder pressures (3.0 and 5.0 MPa), due to in-cylinder higher

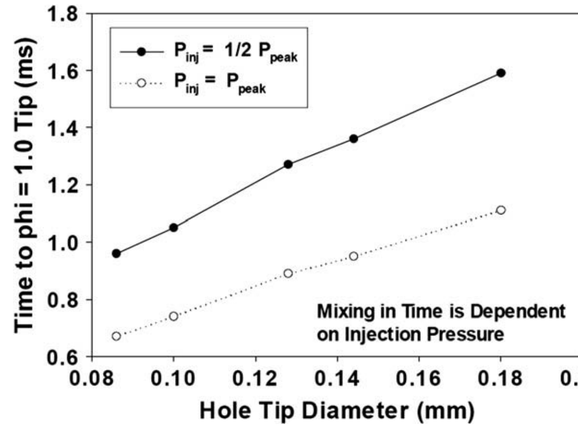

Fig. 17. The computed time required to mix the centerline of the jet to an equivalence ratio of $\varphi = 1.0$ as a function of injector tip hole size [225].

temperatures. Especially for LTC, low oxygen concentrations (high EGR rates) lead to a serious lack of oxygen in the combustion process, and therefore, high boost is necessary to ensure enough fresh air during combustion, as discussed in 3.4 [220,221].

3.5.1.3. Design of combustion chamber geometry and utilization of energy of spray wall impingement. Fig. 20 gives the combustion geometries used by advanced heavy-duty diesel engines in the world. The design of these combustion chambers is dominated by the concerns in matching of fuel, air and combustion chamber [227,228].

The BUMP combustion chamber, which is applied in MULIN-BUMP combustion system proposed by Su and Zhang [228], is a kind of high mixing rate combustion chamber. In order to understand the underlying mechanism of emission reduction, STAR-CD code [229] based on multi-dimensional combustion modelling study was carried out for a heavy-duty diesel engine with a BUMP combustion chamber and a conventional one without the bump ring [228]. Fig. 21 reveals that the flow patterns of the two combustion chambers are quite similar before fuel injection starts at 3° ATDC. At about 10° ATDC, the fuel injection induced air motion impinges with the chamber wall. The flow pattern in the BUMP combustion chamber then becomes more complicated. At 13° ATDC, the air motion in the chamber without the bump ring impinges on the chamber wall and then flows along the chamber wall with a large velocity of 123.5 m/s, but the rest of the chamber

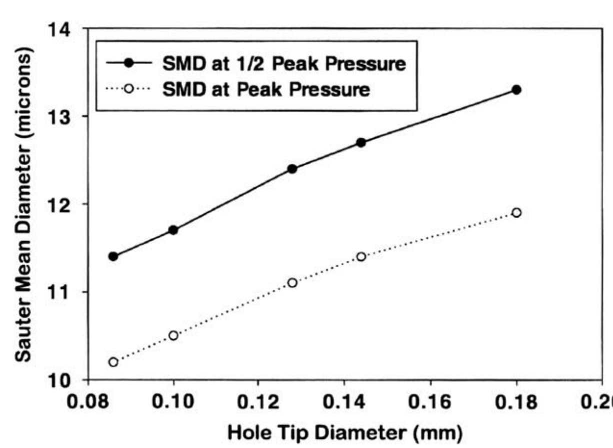

Fig. 18. Computed sauter mean diameters (roughly, average dropsizes) versus tip hole diameters for peak injection pressure and 50% of peak injection pressure [225].

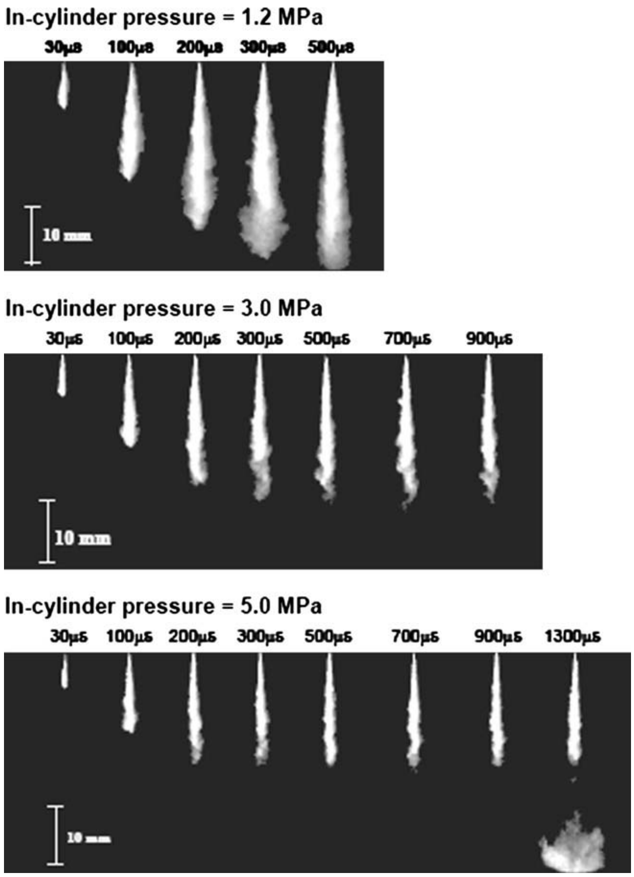

Fig. 19. Evaporating spray evolution, under different in-cylinder pressure conditions [226].

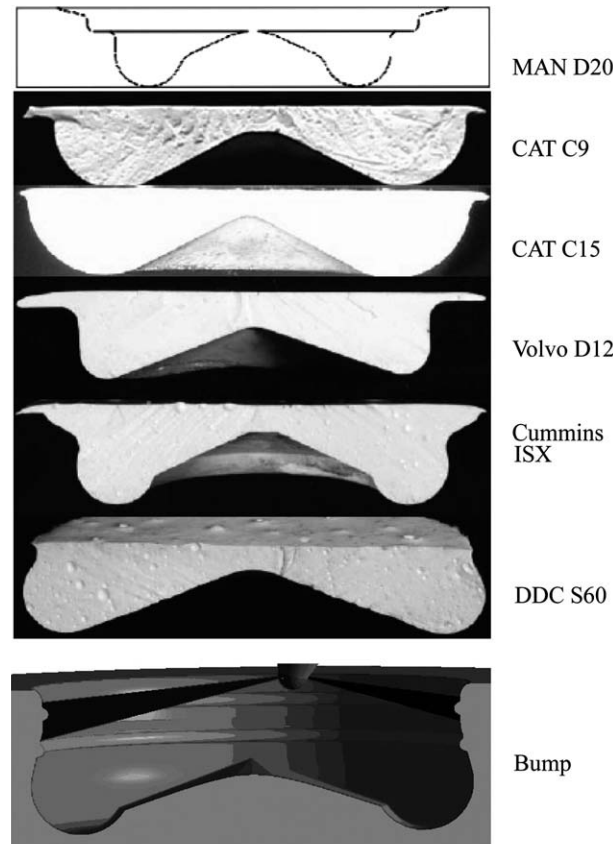

Fig. 20. Combustion geometries used by advanced heavy-duty diesel engine in the world [227].

is rather quiet. In contrast, in the chamber with the bump ring the air motion induced by fuel injection impinges on the wall and then is partially bent over, forming a medium scale vortex, under the disturbance of the bump ring. With the other air stream sweeping along the chamber wall, the two air flows stir the combustion chamber with different scales, increasing the turbulence kinetic energy and enhancing mixing rate. In addition, the results also indicate that the high kinetic energy zone in the combustion chamber with the bump ring occupies a larger space and possesses a higher level of turbulence energy than that of conventional combustion chambers, which implies that the injected fuel in the BUMP combustion chamber mixes with air in a wider space with stronger mixing energy.

Fig. 22 shows the calculated distributions of temperature and concentration in combustion chambers with and without the bump ring at instants just before auto-ignition. It indicates that the bowl with a bump has a more homogeneous concentration distribution. Considering the soot formation threshold ($\varphi > 1.67$), there are 38.2% and 28.9% of fuel fall in the range of $\varphi < 1.67$ for the chambers with and without the bump ring respectively. It shows the benefit in reduction of soot formation for the bumped bowl. Observing the optimal range of fuel/air equivalence ratio ($1.25 < \varphi < 1.67$), there are 16.8% and 10.7% of fuel in this range for the chambers with and without the bump ring respectively. It also shows the benefit in reduction of NO_x formation for the bumped bowl.

3.5.1.4. Multi-pulse fuel injection based on modulating injection mode. Early injection, well before top dead centre (TDC), has been perhaps the most commonly researched approach in obtaining HCCI combustion in a direct injection diesel engine. It relies on in-cylinder flow to promote mixing and high gas temperatures in the cylinder to evaporate diesel fuel. However, wall wetting due to over-penetration of the fuel spray can lead to unacceptable amounts of unburned fuel and the removal of lubrication oil. The MULINBUMP compound combustion system, which has been discussed in section 3.1.3, had been proposed by the Su et al. [196–198] in order to avoid wall wetting.

3.5.2. Control strategies of extending ignition delay

3.5.2.1. Technology of exhaust gas recirculation. That EGR mixes with fresh air as diluter can lead to the increase of specific heat capacity in the cylinder. Hence, compression temperatures before ignition rise more slowly and ignition delay becomes longer. In addition, flame temperatures after ignition decrease, which is beneficial in reducing NO_x emissions. The MK combustion system mentioned above is a successful example that employs this method. In addition, high level EGR is used in LTC. Therefore, EGR becomes one of the most important techniques used to control combustion.

3.5.2.2. Technology of variable compression ratio and variable valve actuation. Variable compression ratio technology changes in-cylinder pressure and density, which can produce effects on auto-ignition of fuel, and by which the in-cylinder temperature is controlled. Variable valve technology (timing or lift, VVT&L) can control mixing time by controlling the histories of in-cylinder temperature and pressure. It is an effective method combining reduced effective compression ratios with variable valve technology. Reitz et al. [230] explored the limitations and potential of an intake valve actuation system on a heavy-duty diesel engine using

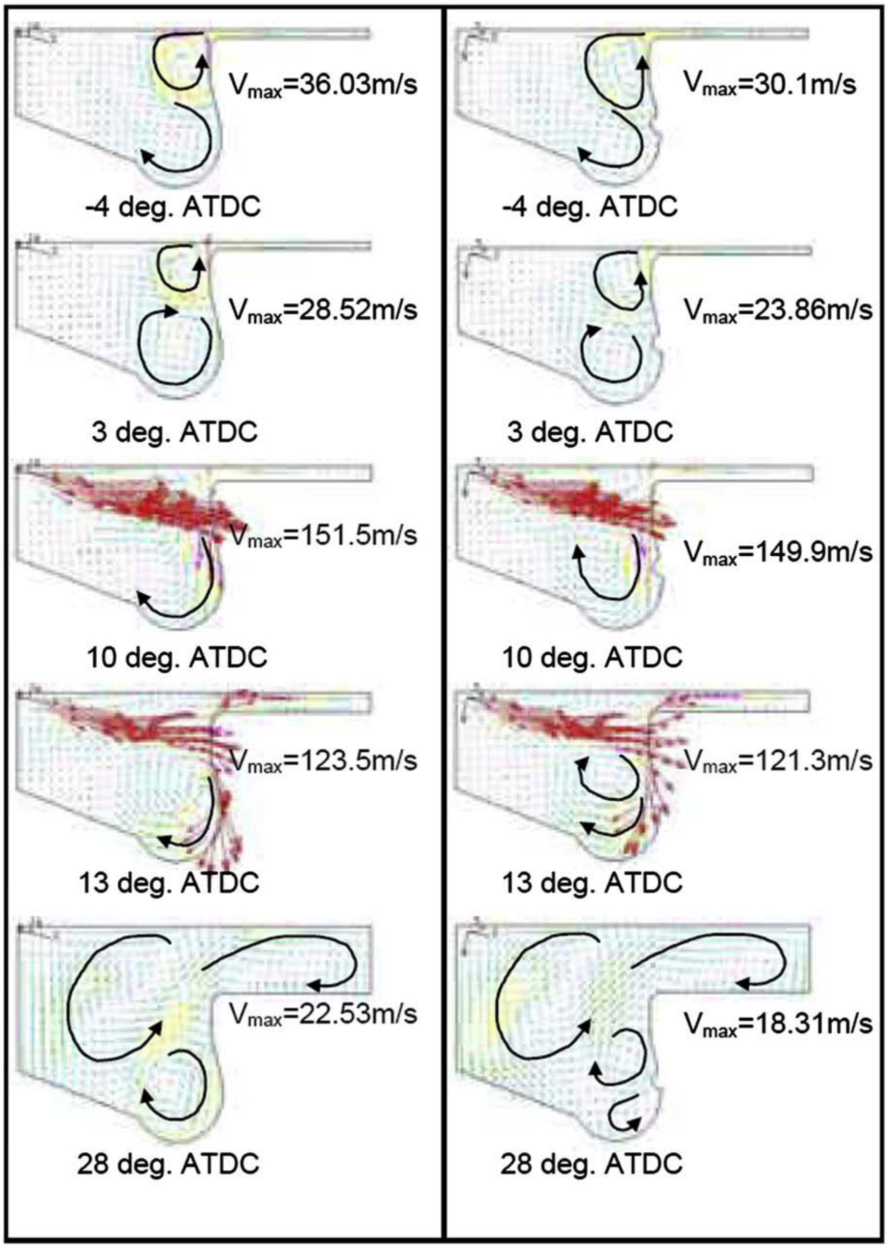

Fig. 21. Flow patterns in the combustion chambers with and without a bump ring [228].

this method. With the use of a variable intake valve-closing device, in-cylinder temperatures can be lowered by effectively reducing the compression ratio by closing the intake valve later in the compression stroke. Ideally, with lower in-cylinder temperatures, the injection can be started at a time that avoids wall wetting, and longer injection durations (lower fuelling rate) can be used without giving unacceptable in-cylinder pressure levels. Fig. 23 shows simulated pressures and temperatures at TDC for maximum valve lifts of 1, 3, 5, and 7 mm. It can be seen that with low lift, the peak pressure decrease is minimal with later IVC timing. Likewise, the temperature is also minimized with small valve lift. As shown in Fig. 24, retarding the IVC timing had a substantial effect on the overall level of NO_x emissions because the effective compression ratio is lowered which lowers compression temperatures and provides more time for homogeneous mixing and hence NO_x reduction. Furthermore because later IVC timing allows for in-cylinder gases to be further evacuated during the compression stroke, there is lower oxygen availability, which causes the air–fuel mixture to burn richer.

3.5.2.3. Fuel modification. By fuel blends and additives, the auto-ignition characteristic can be changed, and then the ignition delay can be extended. This technology has been discussed in section 2.2.

In the research of diesel-fuelled HCCI engines, the above combustion process control strategies are often not used alone, but combinations of them have achieved better effects. For example, according to the reports of Caterpillar Inc in 2004 [231], HCCI operation ranges in heavy-duty diesel engines have been extended to 80% loads of the original conventional diesel engine (about 1.6 MPa) by combining the techniques of high boost, improving the mixing rate of fuel and air, EGR and fuel modification. In 2006, Scania Inc [232] in Europe reported that meeting Europe 4 and 5

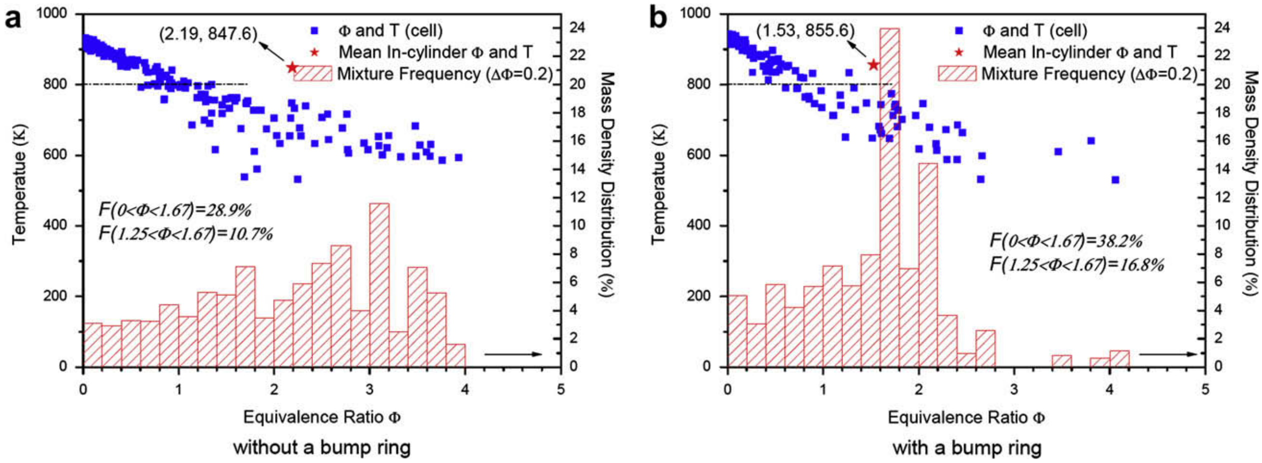

Fig. 22. Comparison of in-cylinder temperature and mass density distribution for the two bowls at ignition timing (7 deg. ATDC) [228].

emissions regulations does not need PM and NO_x after-treatments if adopting high pressure common-rail fuel injection systems and a new combustion mode.

4. Evolution in control strategies of gasoline-fuelled HCCI engines

4.1. Main path to achieve gasoline-fuelled HCCI combustion

Gasoline fuel, with the feature of high volatility, is easily vaporized which is beneficial in forming a homogeneous charge. Therefore, there is a great benefit to use gasoline as fuel (based on diesel fuel) in HCCI engines. However, the high octane rating of this fuel means that it needs high ignition temperature, which brings the difficulty of auto-ignition by compression. This is the fundamental difficulty in achieving gasoline-fuelled HCCI combustion. Since gasoline engines are mainly used for light duty vehicles, the importance of gasoline-fuelled HCCI combustion research is focused on energy conservation and reducing emissions at medium and low loads. In addition, the ‘‘CAI combustion’’ was used broader than ‘‘HCCI combustion’’ in the literatures. However, to keep the consistency of presentation, the ‘‘HCCI combustion’’ was also used in this section.

The main challenge for gasoline HCCI operation is obtaining sufficient thermal energy to trigger auto-ignition of the mixture late in the compression stroke. The most practical means to do this in a gasoline HCCI engine, where the compression ratio is limited, is through the use of large levels of residual exhaust gases. Ishibashi and Asai [233] used a variable exhaust port restriction to modulate residual exhaust in a single cylinder two-stroke engine, and in this way they were able to successfully control HCCI combustion phasing over one-third of the engine operating domain. However, they found that there was insufficient thermal energy to obtain repeatable HCCI combustion at near-idle loads, and there was too little charge dilution above medium loads to control the energy release process.

In four-stroke engines with flexible valve actuation, there are many strategies to obtain a high level of internal residuals. From an overall standpoint, all these valve strategies can be divided into two general modes. In the first mode, the exhaust gas re-enters the cylinder after leaving the engine through one of the ports. This is an exhaust re-breathing strategy. In the second mode, the exhaust valves are closed early in the exhaust stroke to trap the exhaust gas within the cylinder. This is an exhaust recompression strategy. There exists a wide range of valve timings that can be used with the re-breathing strategy. The intake valves can be opened for a short

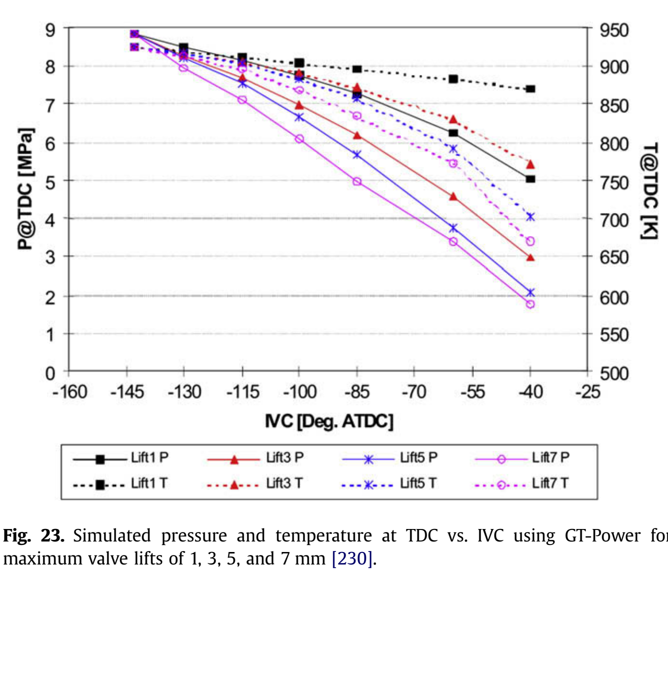

Fig. 23. Simulated pressure and temperature at TDC vs. IVC using GT-Power for maximum valve lifts of 1, 3, 5, and 7 mm [230].

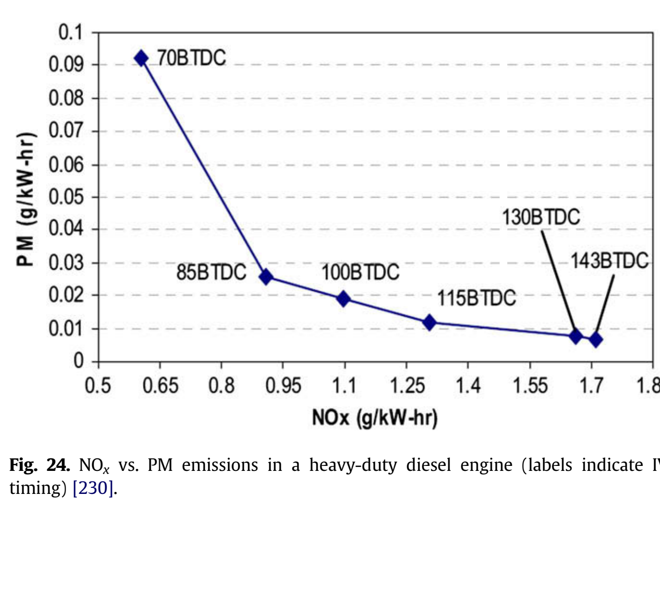

Fig. 24. NO_x vs. PM emissions in a heavy-duty diesel engine (labels indicate IVC timing) [230].

time during the exhaust stroke so that exhaust gas flows into the intake port, which then re-enters the cylinder with the remaining fresh charge during the intake stroke. Alternatively, residual gas dilution can be achieved by re-opening the exhaust valves during the intake stroke. Yet a third option is to hold the EVO through the exhaust stroke and well into the intake stroke (delay overall strategy). In comparison, the valve timings that can be used in the recompression strategy are simple. The exhaust valve is closed well before TDC of the exhaust stroke in order to trap the exhaust gas within the cylinder, and the intake valve opening location is symmetric with the exhaust valve-closing in order to minimize pumping losses during recompression.

All of these strategies have undergone experiments and been reported in the literature. For example, applying a VVA system, Kaahaaina et al. [234] achieved HCCI operation using the re-breathing strategy where the exhaust valve remained open throughout the intake stroke. Zhao et al. [235] used the exhaust recompression strategy in their HCCI experiments. Like the two-stroke experiences of Ishibashi et al., these researchers were able to successfully modulate the levels of residuals to control HCCI combustion over a range of speed and loads. However, these dilution strategies were not able to provide enough thermal energy for satisfactory HCCI combustion at near-idle loads, and thus HCCI operation at near-idle loads remains problematic. Law et al. [236] compared the two dilution strategies and found no significant differences in the cylinder pressure profile or combustion characteristics between them. From a practical perspective, however, the exhaust recompression strategy appears to be easier to implement and has become the strategy favoured in the literature.

4.2. Control strategy of gasoline-fuelled HCCI engines

Combustion control of gasoline-fuelled HCCI engines can be divided into two areas. One covers heat release control, which could be of great benefit to enlarge the operation range. The other is auto-ignition timing control. The key factor for the ignition control is the in-cylinder gas temperature. Some approaches to control gasoline-fuelled HCCI combustion, such as fuel injection strategies, charge boost, EGR, variable compression ratio, multi-mode and fuel modification are reviewed here.

4.2.1. Fuel injection strategies

To obtain the most homogeneous mixture, it is desirable to have a long mixing time between fresh air and fuel. Thus it seems that early injection using conventional port fuel injection would be more advantageous to obtain good homogeneous HCCI combustion. Successful HCCI operations have been achieved by many researchers using port fuel injection [236,237], but there are drawbacks to this operating mode. Port fuel injection offers no potential for additional combustion phasing control and limits the maximum usable compression ratio. A switch to direct injection offers the potential for increasing compression ratios and thus extension of the HCCI light load limits. Direct injection also offers the potential for combustion phasing control under conditions where the spark discharge is no longer effective.

Using direct injection, Marriott and Reitz [238] showed that combustion phasing could be controlled by changing injection timing. By altering the injection timing from early in the intake stroke to late in the compression stroke, they were able to obtain optimum combustion phasing over a range of intake air temperatures, engine loads, and speeds. CFD calculations indicated that these combustion phasing changes are linked to changes in charge stratification resulting from changes in injection timing and can have a significant effect on emissions. Indeed, Marriott et al. found that by varying injection timing and thus the level of mixture homogeneity, they could dramatically alter engine emissions. Early injection created very homogeneous mixture, which leads to achieving extremely low NO_x and PM emissions, but HC emissions were quite high. By shifting to late injection timing, which resulted in significant charge stratification, they were able to cut the HC emissions by a factor of three, but NO_x and particulate emissions became unacceptably high. With optimum timing, however, they were able to achieve low levels of all three emissions. Urata et al. [239] determined that the HCCI operational range could be expanded both to higher and lower load ranges through the adoption of an electromagnetic valve train and the use of direct fuel injection devices. The results indicated that direct injection of fuel into the combustion chamber during the compression stroke expanded operations up to about 650 kPa IMEP compared with that obtained through a premixed homogenous charge with port injection. However, the variability in combustion deteriorated due to cycle-by-cycle variation in the air–fuel mixture and internal EGR gas temperatures. Fuel injection during NVO expanded the operational range towards lower loads. A reaction involving residual unburned hydrocarbons may have facilitated compression ignition, judging from the luminescence observed during NVO without fuel injection.

More recently, Li et al. [240] investigated the effects of injection timing on mixture quality and gasoline-fuelled HCCI combustion. The results also indicate that injection timing has important effects on HCCI combustion and emissions via mixture quality as well as the pre-combustion reactions. It is possible for fuel injection timing to be used to control gasoline-fuelled HCCI combustion. Tian et al. [241] also investigated the control of gasoline-fuelled HCCI combustion through two-stage direct injection. They also conclude that the injection timing and ratio can control the HCCI combustion effectively. Partial fuel injection in NVO can realize fuel reformation, enhance ignition capability, and reduce the coefficient of variation (COV) of the combustion.

4.2.2. Charge boost

Yap et al. [242] applied inlet charge boosting to gasoline HCCI operation on an engine configuration that was typical for SI gasoline engines. In this study, residual gas trapping was applied in conjunction with forced induction to increase the upper load range. Fig. 25 shows the engine load achieved at various boost pressures and various exhaust valve timings without intake heating. As can be seen, a considerable increase in engine load can be achieved with increasing boost pressure. At a maximum boost pressure of 1.4 bars boost, the IMEP has reached 7.6 bar. This is approximately 75% of the total engine load possible in this engine configuration with SI combustion. Fig. 26 shows the same engine loads as Fig. 25, plotted against lambda (air excess ratio) and exhaust valve timing. Higher engine loads are possible with HCCI combustion at higher lambda (leaner air–fuel ratios). It can also be seen that as the exhaust valve timing is retarded, for a given lambda, the engine load achieved is higher. Boost pressures are an important approach to load control whilst lambda controls combustion phasing due to the presence of residual gases. There is a maximum amount of boost that can be applied without intake heating for any given amount of trapped residuals due to the limitations of their heating effect. Yao et al. [243] also found that the successful HCCI operating range is extended to the upper and lower load as the inlet pressure increased when running on the PRF and gasoline. The HC and NO_x emissions of the HCCI engine decrease when supercharging is employed, while CO emissions increase.

4.2.3. EGR

In the research of Cairns and Blaxill [244], a combination of internal and external EGR has been used to increase the attainable

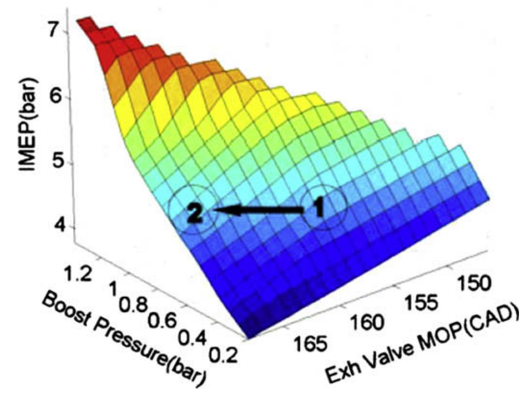

Fig. 25. Engine load for various boost pressures and exhaust valve timing at 1500 rpm. (Points "1" and "2" represent the same IMEP but very different NO_x emissions) [242].

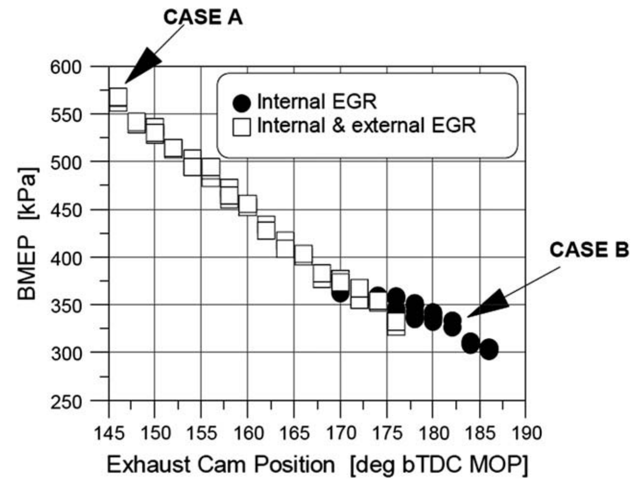

Fig. 27. BMEP versus exhaust cam position (1500 rpm, $\lambda = 1.0$) [244].

load in a multi-cylinder engine operated in gasoline controlled auto-ignition. The amount of residual gas trapped in the cylinder was adjusted via the NVO method (recompression). The flow of externally re-circulated exhaust gas was varied using a typical production level valve. Under stoichiometric fuelling conditions, the highest output achieved using internal exhaust gas was limited by excessive pressure rise and unacceptable levels of knock. Introducing additional external exhaust gas was found to retard ignition, reduce the rate of heat release and limit the peak knocking pressure. In Fig. 27, it can be seen that addition of external EGR enabled significant increase in peak engine output, rising from 350 kPa to 580 kPa (~65%). This supplementary cold diluting gas served to retard ignition, evident in the phasing of the 10% mass fraction burned (Fig. 28). In addition, under conditions of combined EGR, NO_x values further decreased at high loads. It should be noted that the flexible valve actuation can obtain a high level of internal residuals, which is the focus of gasoline-fuelled HCCI engines. All these researches have been review in section 4.1.

4.2.4. Variable compression ratio

CAI combustion can be achieved by increasing the compression ratio to the point where the required temperature and pressure for auto-ignition are achieved through compression alone. The combustion phasing can also be controlled by variable compression ratio. However, it is difficult to apply in practice due to the complex engine design.

Hyvonen et al. [245] investigated the operating range in terms of speed and load with naturally aspirated four-stroke multi-cylinder engine. HCCI combustion control is achieved by variable compression ratio and inlet air preheating with exhaust heat. Both reference fuels and gasoline fuel were tested. The range of variable compression ratio is between 9:1 and 21:1. Operating ranges in Figs. 29 and 30 show that lower compression ratios are used at intermediate loads as well as intermediate and high engine speeds to increase combustion efficiency, whilst compression ratios are close to maximum both at low load and high load. At low load and idle mode, maximum compression ratio and maximum available inlet air temperature are needed to initiate combustion and increase combustion efficiency. An operating range from 1000 to 5000 rpm and from 0 to 3.5 bars BMEP is achieved. In addition, the results also show that brake efficiency is increased by 35% at 3 bars BMEP and 57% at 1.5 bar BMEP compared with conventional SI combustion with the same engine using a 14:1 compression ratio. About more studies on the VCR can be found by Christensen et al. [79], Christensen and Johansson [246] and Hyvonen et al. [247].

4.2.5. Multi-mode

Multi-mode combustion is an idle combustion strategy to utilize HCCI for internal combustion engines. It combines HCCI combustion mode at low to medium loads with traditional SI mode at high speed and high loads.

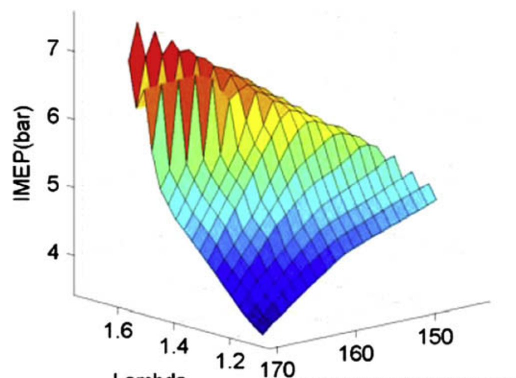

Fig. 26. Engine load for various air–fuel ratio and exhaust valve timing at 1500 rpm [242].

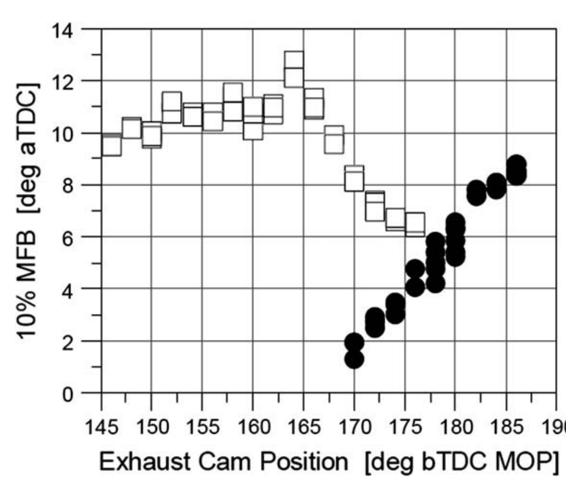

Fig. 28. 10% mass fraction burned versus exhaust cam position [244].

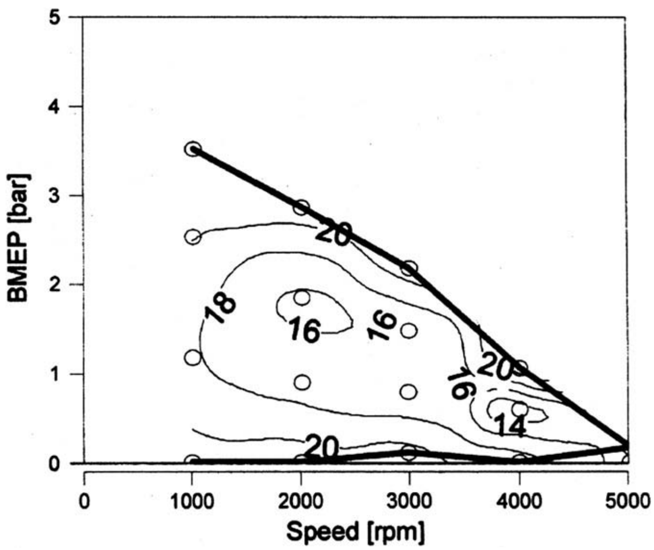

Fig. 29. Operating range and compression ratio iso-lines with gasoline fuel [245].

To fully utilize the potential of HCCI combustion for high fuel conversion efficiency and low NO_x emissions and to improve HCCI control for vehicle applications, a gasoline HCCI-SI dual mode engine system called the optimized kinetic process (OKP) was proposed by Yang et al. [248]. The OKP concept captures thermal energy from the hot residuals, coolant and exhaust gases, and uses compression heat with higher compression ratio (about 15:1) to promote HCCI auto-ignition. Hence, the OKP engine is able to operate over a broader range of operating conditions and to operate at lower ambient temperatures compared with the HCCI concept that uses hot residuals and conventional compression ratio (about 10–11:1) for auto-ignition [249]. Tests on a single cylinder OKP engine in HCCI mode have demonstrated a wide operating range and very low fuel consumption. To operate in SI mode at high loads or during cold start, a two-stage intake valve timing control device was needed to reduce the residuals and the effective compression ratio to operate in the Atkinson cycle. Intake pressure boosting was applied near full load to compensate for the loss in volumetric efficiency [248]. The AVL-CSI system, as proposed, developed and demonstrated by Fuerhapter et al. [250], was used in an OPEL 2.0L 4-cylinder engine. At HCCI mode, the intake camshaft was shifted to small valve lift for early intake closing and the amount of fresh air was adjusted via a cam phaser and by setting the intake manifold pressure with an electronic throttle. SI combustion with early intake closing (Miller cycle), offers remarkable fuel reduction potential and the intake camshaft was switched to big valve lift in order to provide sufficient full load performance. The results show that CSI 4-cylinder engine includes the demanded valve train functionality with moderate impact on the base engine structure and has an individual cylinder combustion control which enables the control of different combustion modes even under dynamic conditions.

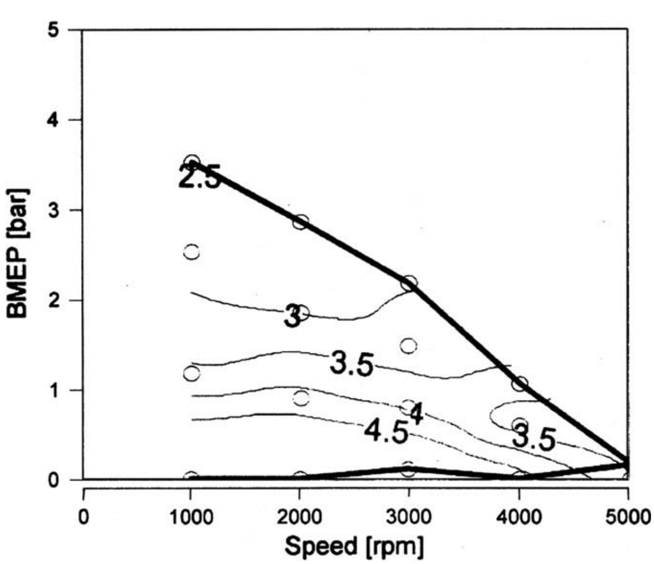

Fig. 30. Operating range and lambda iso-lines with gasoline fuel [245].

Xie et al. [251] described a flexible mechanical VVA system named 4-variable valve actuating system (4VVAS). The 4VVAS system is capable of independent control of intake valve lift and its timing, exhaust valve lift and its timing and it has been incorporated in a specially designed cylinder head for a single cylinder research engine. To investigate the potential of the 4VVAS system for HCCI engine operation and the switch between HCCI and SI operations on the same engine, an engine simulation program has been developed based on Matlab Simulink incorporating an engine block. Results show that the variable lift capability of the 4VVAS system enables the HCCI operation to be extended to higher load at high engine speeds. The investigation into the transient response has shown that a combination of synchronous change in the exhaust and intake valve lifts together with changing exhaust valve timing would offer the fastest control of the HCCI engine output. More recently, a 4VVAS-HCCI gasoline engine test bench was established by Zhang et al. [252]. The experimental research was carried out to study the dynamic control strategies for transitions between HCCI and SI modes on the HCCI operating boundaries. The results show that the engine can be operated in HCCI or SI mode. 4VVAS enables rapid and effective control over the in-cylinder residual gas, and therefore dynamic transitions between HCCI and SI can be stably achieved. Tian et al. [253] proposed HCCI/SI switch mode by changing the cam profiles from normal overlap for HCCI mode, as well as the adjustment of direct injection strategy. Their research indicates that the mode switch between SI and HCCI can be successfully realized with a two-stage cam system and direct injection. The switch process is smooth and reliable without abnormal combustion. The switch from HCCI to SI is easy, but there was fluctuation from SI to HCCI mode, which is optimized by a coupled BOOST and CHEMKIN model. More references on the multi-mode can be found in Hyvonen et al. [247], Koopmans et al. [254], Roelle et al. [255], and Santoso et al. [256], Shaver et al. [257], Zhao [258], and Wang [259].

The multi-mode is the developmental direction for the large scale production of gasoline-fuelled HCCI engines in the future. While the flexible valve actuation and injection strategies are the keystone to reach the combine HCCI combustion mode at low to medium loads with traditional SI mode at high speed and high loads.

4.2.6. Fuel modification

High CN fuels (e.g. diesel, DME and n-heptane) can be used to enhance HCCI combustion due to the lower auto-ignition temperature required. In applications to gasoline engines, those high cetane fuels would be mixed with high octane number fuel (gasoline, iso-octane, methanol, etc.). Thus dual fuel supply systems or additional additives are usually needed. This technology has been discussed in section 2.2.2. In addition, for the kinetics studies in fuel modification, a great deal of work is also necessary to understand the auto-ignition chemistry of practical fuels, alternative fuels, and different fuel modification at a fundamental level. A start could be made with simple fuels like toluene/n-heptane mixtures [260,261] which behave more like practical fuels than iso-octane/n-heptane mixtures. While the chemistry kinetics of fuel modification is also needed, such as the DME and methanol dual fuel HCCI engine simulation by Yao et al. [70].

4.2.7. Other control methods

In general, expensive pressure transducers are used in laboratories to determine combustion characteristics in the engine. Ion sensors are significantly less expensive than pressure transducers. For example, an existing spark plug can often be used as an ion sensor. An ion sensor signal has been used for combustion diagnostics [262,263], combustion stability control [264], peak pressure position control [265], predicting mass fraction burned [266] and the knock onset [267] in SI engines. The engines cited above operate at or near stoichiometric conditions where temperatures are high and the production of chemi-ions is significant.

As engines operate leaner, the lower temperatures dramatically reduce the production of chemi-ions. In the case of HCCI engines, where φ is seldom above 0.5, few researchers expected to find an ion signal. The paper written by Strandh et al. [268] has been the first to report an ion signal in a gasoline-fuelled HCCI engine.

Dibble et al. [269] investigated the ion current signal in homogenous charge compression ignition (HCCI) engines. This research is to show that a measurable ion current exists even in the very lean combustion ($\varphi = 0.35$) in an HCCI engine. Numerical models using detailed chemical kinetics for propane combustion, including kinetics for ion formation, support the experimental findings. The effects of the equivalence ratio, the intake mixture temperature, and the applied bias voltage on the ion signal are studied through a series of experiments. The findings are compared to the numerical model results. The research shows that an inexpensive ion sensor may replace the expensive pressure transducers currently used in HCCI engines. It is equally capable of capturing cycle-to-cycle variations in HCCI combustion. The ion signal becomes stronger with increases in equivalence ratio and intake temperature. In their further researches [270], experiments were conducted to measure ion signals produced from the combustion of gasoline, ethanol, and n-heptane in a 4-cylinder HCCI engine with different equivalence ratios and intake pressures. It was found that the ion signal is reduced under several situations: with an increase in intake pressure, reducing equivalence ratio (under lean conditions), and decreasing the bias voltage source. The gasoline and ethanol produces more ions than n-heptane during combustion under the same operating conditions.

Xie et al. [271] investigated the influence of spark ignition on CAI combustion based on internal EGR strategy in a single engine. The results show that spark ignition can play an important role in controlling CAI combustion ignition in low load boundary region. The low temperature chemical reaction process is shortened and the auto-ignition timing is advanced due to the spark discharge. Meantime, lower fuel consumption and cycle-to-cycle variations can be achieved. The spark discharge makes auto-ignition easier in low load boundary region, so that the CAI operation can be expanded to lower load through assisted spark ignition. Therefore, it has the potential of spark ignition for control of auto-ignition timing and control of CAI/SI mode switch. Hyvonen et al. [247] investigated the spark assisted HCCI combustion during combustion mode transfer to SI in a multi-cylinder VCR-HCCI engine. They found that spark assisted can be used for controlling the combustion phasing during a mode transfer between HCCI and SI combustion. However, the combustion fluctuations are large in the intermediate combustion region where some cycles have both spark ignited flame propagation and auto-ignition, but some cycles have only partially burnt flame propagation. More articles about spark assisted HCCI can be found by Persson et al. [137], Berntsson et al. [158], Bunting [272], and Felsch et al. [273].

5. Conclusions and future research directions

5.1. The conclusions of the fundamental theory

Four main results from the pioneering investigation of HCCI combustion have been established: first, HCCI combustion demonstrates a strong potential to improve the thermal efficiency of gasoline-fuelled engines and substantially reduce NO_x and soot emissions of diesel-fuelled engines; second, the chemical kinetics has a dominating role in HCCI combustion; third, difficulties associated with the successful operation of HCCI engines need to be overcome including: i) combustion phasing control, ii) high HC and CO emissions, iii) extending the operation range, iv) cold start problems and homogeneous mixture preparation; four, the HCCI concept could be expanded to all the new engine combustion modes achieving high efficiency and low emissions, such as PCCI, PREDIC, LTC, etc. The main characteristics of HCCI combustion include: premixed mixture, compression ignition and low temperature combustion. After a great deal of research, the evolution in the investigation of HCCI can mainly be described as follows.

Numerical simulation has become a powerful tool in realizing HCCI and seeking control strategies for HCCI, and has higher flexibility and lower cost compared with engine experiments. There are five categories of approaches applied to HCCI engine modeling: zero-dimensional single-zone models with detailed chemistry, quasi-dimensional multi-zone with detailed chemistry, one-dimensional engine cycle with detailed chemistry in-cylinder, multi-dimensional CFD with multi-zone detailed chemistry, and multi-dimensional CFD with detailed chemistry. In these models, multi-dimensional CFD models have the greatest potential in predicting realistic results when the geometry of the combustion chamber is resolved in full detail, in combination with a detailed chemistry approach to model combustion.

HCCI can be applied to a variety of fuel types and the choice of fuel will have a significant impact on both engine design and control strategies. Some chemical components have the ability to inhibit or promote the heat release process associated with auto-ignition. Therefore, HCCI auto-ignition can be controlled by modification of the fuel so that it is more chemically reactive or inhibitive by adding an ignition promoter or inhibitor.

Utilizing charge inhomogeneity is an important path to achieve clean and high efficiency combustion in engines. Advanced control strategies of fuel/air mixture are more important than simple "homogeneous charge" in the control of the HCCI combustion process. Stratification strategy has the potential to extend the HCCI operation range to higher loads. EGR-diluted LTC also has the potential to extend the operation range to high loads, and even full loads for diesel engines. The HCCI concept has been greatly enriched with deeper and deeper research, and enhancements are grouped under the heading of "generalized HCCI combustion"

The application of optical diagnostics has greatly accelerated developments in many aspects of HCCI research, including the measurements of in-cylinder mixture formation, combustion processes and emissions formation. By optical diagnostics, a multitude of effects of inhomogeneous charge on HCCI combustion process, key intermediate species and radicals (formaldehyde and OH) and NO_x and soot emissions can be observed. The research has proven to be making valuable contributions, providing foundations for a better understanding of HCCI combustion processes and for further development of simulation tools.

5.2. The conclusions in control strategies of HCCI engines

Typical generalized diesel-fuelled HCCI combustion modes include: early direct injection HCCI, late direct injection HCCI, premixed/direct-injected HCCI combustion and low temperature combustion. Mixture control (mixture preparation), including charge components and temperature control in the whole combustion history and high pre-ignition mixing rate, is the key issue to achieve diesel HCCI combustion. There are two measures to improve mixture formation: 1) by improving the mixing rate of fuel and air by such means as high pressure/ultra-high pressure fuel injection and small nozzle holes, high boost, design of combustion chamber geometry and utilization of energy of spray wall impingement and multi-pulse fuel injection based on modulating injection mode; and 2) by extending ignition delay by such means as EGR and variable compression ratio/valve actuation technology.

Since the diesel fuel has low volatility, the port fuel introduction is not a practical way without significant change of intake system. An early in-cylinder injection strategy, to some extent, can result in a quite homogeneous charge before ignition. Due to lower charge density, in-cylinder pressure, and temperature, the liquid fuel impingement on the liner wall or piston wall is unavoidable, which leads to high HC and CO emissions. Another issue for the early injection strategy is the ignition timing control. For early injection HCCI combustion, the ignition is purely controlled by the chemical kinetics. The ignition is often advanced due to early injection timing and other measures have to be taken to delay the ignition by using heavy EGR, variable compression ratio, changing fuel properties, etc. In practice, both the HCCI mode and conventional diesel combustion will have to be used to cover the complete engine operational range.

For the LTC, the short times between the fuel injection event and the start of combustion preclude thorough premixing, and significant regions exist where $\varphi > 1$ at the start of combustion. Even though there is a locally rich region in the mixture of this strategy, the soot formation can be suppressed. The main soot suppression mechanism is that the temperature is reduced by using large amounts of EGR, and this temperature reduction is sufficient to allow the combustion to avoid the soot formation region. This is the major reason why smokeless combustion can be accomplished with no adjustment required in the mixture formation by changing fuel spray system, combustion chamber geometry, etc., under rich operating condition.

Simultaneously, the NO_x emissions can also be avoided due to the high EGR rates and thus low combustion temperature. Furthermore, the EGR rate influences the path not only through changes in the flame temperature, but also in ignition delay and the amount of ambient fluid that must be mixed with the fuel to attain a given equivalence ratio. In addition, the injection strategies (including injection pressure, timing and multiple injections) influence the temperature (and density) during the ignition delay period, the peak flame temperature reached, and the premixing improvement. Finally, in order to keep the power density and the combustion efficiency of the engine at high EGR rates, high boost levels are required. Therefore, the control and optimize of EGR rate, injection strategies and high boost are the key issue to the LTC. The LTC has more benefits, such as high efficiency over broad load range, simple control of ignition timing, reduced pressure rise rates, high load capability. So, this strategy will be more promising in the future.

The high octane numbers of gasoline fuels means that such fuels need high ignition temperatures, which highlights the difficulty of auto-ignition. The main challenge for gasoline HCCI operation is focus on the obtaining sufficient thermal energy to trigger auto-ignition of mixtures late in the compression stroke, extending the operational range, and the transient control.

The most practical means to obtain sufficient thermal energy in a gasoline HCCI engine is through the use of large levels of recirculated exhaust gases. There are two EGR strategies with VVA: one is exhaust re-breathing and the other is exhaust recompression. These dilution strategies have no significant differences in the cylinder pressure profile or combustion characteristics. From a practical perspective, however, the exhaust recompression strategy appears to be easier to implement and has become the strategy favoured in the literature.

To reduce the fuel consumption and emissions over real-life drive cycles, the engine must operate in HCCI mode over the widest possible speed and load range. To extend gasoline-fuelled HCCI operation to high loads without transition to knock, some methods can be used, such as fuel modification, variable compression ratio, charge boost, the temperature or charge stratification, and the multiple fuel direction injections. To extend gasoline-fuelled HCCI operation to light loads, high in-cylinder temperatures are necessary to promote compression ignition. Meantime, the post-combustion temperatures need to be optimized between 1500 and 1800 K for low HC, CO and NO emissions. Approaches include variable valve strategies, variable injection timings, charge boost, and spark assisted ignition.

Active closed-loop real-time dynamic control is essential to maintain the desired ignition timing for any practical HCCI combustion system. Speed and load control within the HCCI mode and transitions between HCCI and SI modes have been demonstrated in single cylinder research engines. However, additional complications in multi-cylinder engines require individual cylinder control to ensure the same combustion phasing and reach HCCI/SI transition for all cylinders. A cycle-resolved, closed-loop control system with individual sensors and actuators for each cylinder allowed combustion phasing to be matched for all cylinders, but any changes in the combustion phasing in one cylinder resulted in changes in another cylinder due to exhaust-manifold coupling.

Therefore, it can be concluded that the SI/HCCI dual mode is the developmental direction for the large scale production of gasoline-fuelled HCCI engines in the future. While the flexible valve actuation and direct multiple injection strategies are the keystone to reach the combine HCCI combustion mode at low to medium loads with traditional SI mode at high speed and high loads. However, to realize the practical HCCI combustion system, active closed-loop real-time dynamic control is necessary for the gasoline-fuelled HCCI engines.

5.3. Future research directions

Over the past decade, substantial progress has been made in understanding the in-cylinder processes in HCCI and its extension, such as PCCI, SCCI, LTC, etc., which has greatly expanded our ability to operate in these combustion modes. Challenges remain, however, and research is needed in several areas.

For the fundamental theory of HCCI combustion, some of the most important areas are: first, the effects of turbulence on fuel/air/ EGR mixing and combustion processes are needed to be studied further; second, a great deal of work is also necessary to understand the auto-ignition chemistry of practical fuels at a fundamental level., which will better than the kinetic mode of iso-octane, n-heptane, or their mixtures; third, the formation and oxidation kinetic mechanism of exhaust emissions, such as partially oxidized fuel and soot, should be developed in low temperature combustion conditions, which is different from the conventional CI or SI combustion mode; last but not the least, optical diagnostics will provide foundations for the in-depth physical understanding and reliable predictive submodels for HCCI in-cylinder processes.

For the practical HCCI combustion system, some technologies should be developed further. The flexible injection strategies, EGR control and the closed-loop feedback control, will be the key issue for both the diesel-fuelled and gasoline-fuelled HCCI engines. For the diesel-fuelled HCCI engines, more attention should be paid to extend the LTC operation to higher loads, even the full load, while maintaining good fuel economy and low emissions (especially for the HC and CO emissions). For the gasoline-fuelled HCCI engines, more attention should be paid to individual cylinder control to ensure the same combustion phasing and reach HCCI/SI transition for multi-cylinders engines.

Finally, through the review of these HCCI researches, the authors think that the combustion mode design will be become one of the most interesting topics in the future. To realize the high efficiency and clean combustion in the complete engine operational range, the flexible multi-mode combustion processes must be necessary to various operating conditions and fuel properties by controlling the Φ-T history. First of all, to the various operating conditions and fuel properties, the fuel/air/EGR mixing processes are different. Next, the combustion mode will be changed according to the different fuel/air/EGR mixing processes. And then, the combustion process should not be a single mode as the conventional CI or SI engines, but a multi-mode combustion process to the different fuel/air/EGR mixing processes. The fuel/air/EGR mixing processes can be controlled by some methods, such as multi-injection strategies, in-cylinder turbulence control, boosting, internal and external EGR rates, and flexible valve actuation. Therefore, the multi-mode combustion processes can be organized and optimized by the control of fuel/air/EGR mixing processes. Meantime, a closed-loop control system will construct a bridge between combustion mode and mixing processes. In the end, the optimization of fuel economy and emissions can be reached to the complete engine operational range through an optimum combustion mode design.

Acknowledgements

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