Chapter 4

Homogenous Charge Compression Ignition (HCCI) Engines

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Additional information is available at the end of the chapter

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1. Introduction

With stricter regulations imposed by the European Union and various governments, it is not surprising that the automotive industry is continuously looking for alternatives to Spark Ignition (SI) and Compression Ignition (CI) Internal Combustion (IC) engines. A promising alternative is Homogeneous Charge Compression Ignition (HCCI) engines that benefit from low emissions of Nitrogen Oxides (NO_x) and soot and high volumetric efficiency. In an IC engine, HCCI combustion can the achieved by premixing the air-fuel mixture (either in the manifold or by early Direct Injection (DI) – like in a SI engine) and compressing it until the temperature is high enough for autoignition to occur (like in a CI engine). However, HCCI enignes have a limited operating range, where, at high loads and speeds, the rates of heat release and pressure rise increase leading to knocking and at low loads, misfire may occur. Thus, a global investigation is being undertaken to examine the various parameters that effect HCCI combustion.

HCCI – also referred to as Controlled AutoIgnition (CAI), Active Thermo-Atmosphere Combustion (ATAC), Premixed Charge Compression Ignition (PCCI), Homogeneous Charge Diesel Combustion (HCDC), PREmixed lean DIesel Combustion (PREDIC) and Compression-Ignited Homogeneous Charge (CIHC) – is the most commonly used name for the autoignition of various fuels and is a process still under investigation. Autoignition combustion can be described by the oxidation of the fuel driven solely by chemical reactions governed by chain-branching mechanisms [1],[2]. According to various researchers [3]-[6], the autoignition process in an HCCI engine is a random multiple-autoignition phenomenon that starts throughout the combustion chamber possibly at the locations of maximum interaction between the hot exhaust gases and the fresh fuel/air mixture [7], while others [8] argue that it is a more uniform process. Thus, further understanding of this autoignition process is required in order to control HCCI combustion.

This book chapter consists of six sections including this introduction. In Section 2, the oxidation mechanism behind autoignition combustion and HCCI is analysed, while in the third section, a historical review on the early research on autoignition is presented. In section 4, HCCI combustion is presented in more detail, including aspects such as the effect of fuels, and fuel additives, engine design, etc, as well as the HCCI engines in production. In Section 5, a theory on controlling HCCI is presented, with emphasis on fuel injection strategies, Exhaust Gas Recirculation (EGR) and temperature inhomogeneities. In the final Section, the conclusions of the chapter are presented.

2. What are autoignition combustion and HCCI?

The phenomenon of autoignition combustion is still under investigation, even though HCCI combustion has been applied in a two-stroke engine in a commercial motorcycle [9]. Does HCCI combustion and “hot spots” in the burned area in SI engines propagate in the same way? Is there a flame front propagation present in an HCCI engine? How does autoignition combustion propagate in an HCCI engine? Does turbulent mixing affect HCCI combustion? What fuel properties drive cool flame combustion and what the main combustion? What engine parameters affect HCCI combustion? And most importantly of all, how can HCCI combustion be controlled in the most effective way? This section presents an overview of the nature of the autoignition combustion and what is believed to define HCCI combustion, regardless of the fuel used or the engine parameters.

Autoignition combustion can be described by the oxidation of the fuel driven solely by chemical reactions governed by chain-branching mechanisms [1],[2]. Furthermore, two temperature regimes exists – one below 850K (low temperature oxidation or cool flame combustion) and one around 1050K (high temperature oxidation or main combustion) – that can define the autoignition process [10]-[13]. At high temperatures, the chain branching reactions primarily responsible for the autoignition, are given by

$$HO_2\bullet + RH \rightarrow H_2O_2 + R\bullet$$
$$H_2O_2 + M \rightarrow OH\bullet + OH\bullet + M$$

where R• is any hydrocarbon radical and M are other molecules in the system. At low temperatures, the decomposition of H_2O_2 is quite slow and the reaction mechanisms responsible for the low-temperature combustion are:

$$HO_2R'\bullet + O_2 \leftrightarrow HO_2R'O_2\bullet$$
$$HO_2R'O_2\bullet + RH \rightarrow HO_2R'O_2H + R\bullet$$
$$HO_2R'O_2H \rightarrow HO_2R'O\bullet + HO\bullet$$
$$HO_2R'O\bullet \rightarrow OR'O + HO\bullet$$
$$HO_2R'O_2\bullet \rightarrow HO_2\dot{R}''O_2H$$
$$HO_2\dot{R}''O_2H \rightarrow HO_2R''O + HO\bullet$$
$$HO_2R''O \rightarrow OR'O\bullet + HO\bullet$$

Depending on the structure of the fuel, under engine conditions some fuels would exhibit cool flame combustion and some others will not. In general, long straight chain alkanes, normal paraffins, and low Research Octane Number (RON) fuels would exhibit cool flame combustion while branched-chain alkanes, aromatics and high RON fuels would not [14], [15]. However it was also shown [16] that *iso*-octane may also exhibit cool flame combustion under certain conditions. Furthermore, Kalghatgi [17],[18] has also shown that the temperature is not the only parameter that affects the aforementioned mechanisms and that depending on the fuel composition and the engine conditions, the autoignition process varies significantly. It was therefore suggested that other parameters affect the autoignition process and that a better understanding on the fuel property is needed. Neither the Motor Octane Number (MON) nor RON of different fuels alone can be used to describe the fuel characteristics and it was proposed that the Octane Index (OI) of a fuel should be used as defined by:

$$OI = RON - KS \tag{1}$$

where S=RON-MON and K is a variable that is determined by the engine parameters and operating conditions.

Regardless of the chemical reactions associated with autoignition, the spatial initiation and the development or “propagation” of the autoignition sites is another point of interest. Chemiluminescence and Planar Laser Induced Fluorescence (PLIF) imaging of the autoignition phenomenon has shown that autoignition would start at various locations throughout the combustion chamber [3],[4],[6] probably due to local inhomogeneities. Due to the heat released from the burn regions, the temperature and pressure in the cylinder increase and therefore more autoignition sites appear, until the whole fuel-air mixture is ignited. It was also shown [19] using both chemiluminescence and formaldehyde PLIF imaging in a highly stratified engine (hot EGR gases and cold fresh fuel/air mixture) that these autoignition sites initiated at neither the location of maximum temperature nor the location of maximum fuel concentration, but at the boundary of these two regions. Once the first autoignition sites appeared, double-exposure PLIF or chemiluminescence imaging showed that these sites grow in size at different speeds – more or less they can appear to be “flame fronts” in the absence of any other information (i.e. A/F ratio, in-cylinder temperature, “flame front” speed, double-exposure timings).

This autoignition phenomenon has been applied in IC engines as an alternative to SI and CI engines, and is generally referred to as HCCI combustion. Since under HCCI combustion the fuel/air mixture does not rely on the use of a spark plug or direct injection near Top Dead Centre (TDC) to be ignited, overall lean mixtures can be used resulting to high fuel economy. Thus, the combustion temperature remains low and therefore NO_x emissions decrease significantly [20],[21] compared to SI and CI operation. An illustration of the combustion differences between the three modes of IC operation is shown in Figure 1.

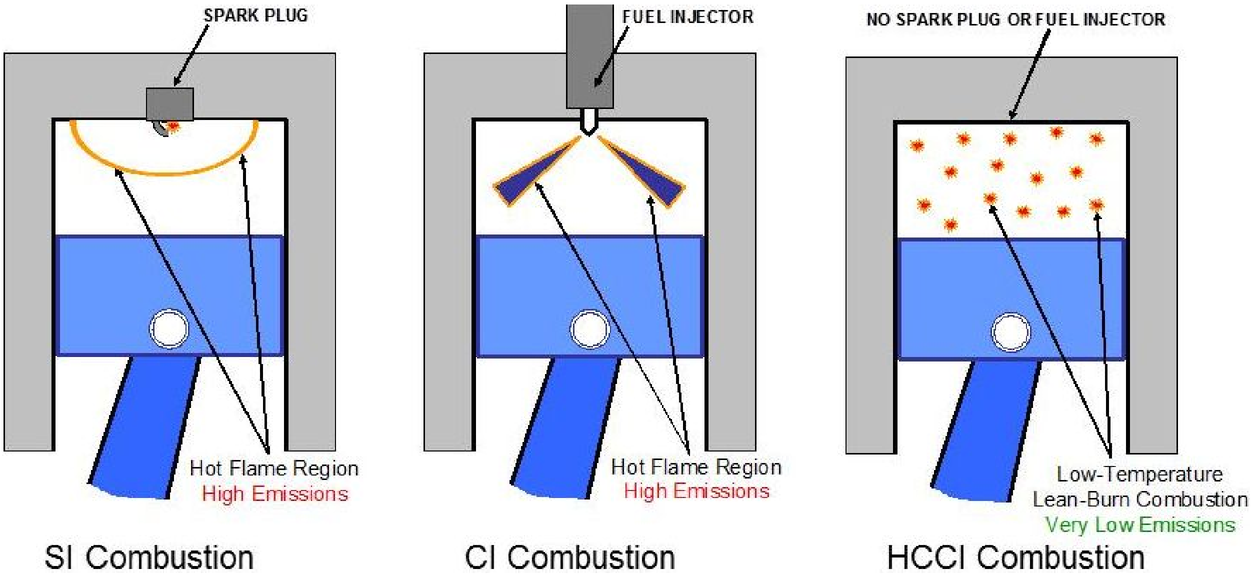

Figure 1. Combustion differences between the three modes of IC operation.

Under optimum operating conditions, HCCI combustion can offer comparable Carbon Monoxide, CO, and HydroCarbon, HC, emissions with SI and CI combustion, but under very lean conditions – and thus low combustion temperatures (approximately below 1500K) – incomplete combustion can occur in the bulk regions leading to partial oxidation of the fuel, decrease in combustion efficiency and increase in CO and HC emissions [12],[22]-[24]. Furthermore, since a homogeneous fuel/air mixture can be prepared in the manifold, low soot can be achieved [20]. However, when HCCI combustion is operated at richer fuel/air mixtures, knocking can occur. In conclusion, HCCI combustion in a production engine is therefore limited by two main regimes [25],[26]:

1. Lean Air to Fuel (A/F) ratio limit – Leading to incomplete combustion, which results to low power and high HC and CO emissions.
2. Rich A/F ratios limit – Leading to knocking if the rate of pressure rise is too high causing damage to the engine or high NO_x emissions due to high combustion temperatures.

3. Early work on autoignition combustion

The autoignition combustion process has been studied and analysed since the beginning of the 20th century. However, it has been studied in an attempt to understand fuel properties and how easily fuels can autoignite and not as the process itself. Only more recently [27], the autoignition combustion has been used to produce positive work in an engine.

As early as 1922 [28], experiments were conducted on the autoignition of *n*-heptane, ether and carbon bisulphide by sudden compression. An apparatus designed by Messrs. Ricardo & Co. that would allow researchers to simulate the conditions obtained in an engine cylinder was used. A heavy flywheel was kept spinning by an electric motor at about 360 Revolutions Per Minute

(RPM) and the Compression Ratio (CR) was varied by altering the cylinder position. The two-stage combustion of *n*-heptane was observed by recording the pressure traces. It was also observed that the ignition temperature (above which an explosion took place), depended both on the properties of the combustible substances (i.e. octane number), on the conditions of the experiments (i.e. CR, initial temperature and pressure) and on the rate of heat losses from the gas. Furthermore, an equation was derived for the time for complete combustion of the explosive mixtures of gases when suddenly compressed to a temperature above its ignition temperature.

A rapid-compression machine capable of producing CRs up to 15:1 was used in the 1950s [3], [29],[30] to investigate the effect of fuel composition, compression ratio and fuel-air ratio on the autoignition characteristics and especially the ignition delay (i.e. the time from when the mixture was suddenly compressed until autoignition) of several fuels that included heptane, *iso*-octane, benzene, butane and triptane. An air-fuel mixing tank was used to ensure the correct ratio, the pressure records were taken with a catenary-type strain-gage indicator and a Fastax camera (operated at a rate of 10,000 frames per second) was used in taking flame and Schlieren photographs. It was concluded that all fuels had a minimum value of ignition delay at their chemically correct air-fuel ratio that increased with decreasing compression ratio. Furthermore, the detonating or knocking properties of the fuels depended both on the ignition delay and on the rate of combustion after autoignition. The flame photographs recorded [3] revealed that autoignition in the rapid-compression machine fell in three loose classifications that is also evident in modern IC engines:

- Uniform combustion throughout the combustion chamber.
- Isolated points of autoignition that developed sporadically in all parts of the combustion chamber.
- The inflammation began in small regions and proceeded across the chamber in the form of a “flame front”.

The possibility of fuel droplets, non-homogeneity in the air-fuel mixture, dust particles, piston contact with cylinder head and temperature gradients causing the non-uniformity in the autoignition process was also investigate by using flame and Schlieren images [30]. They have concluded – in the absence of any data to provide a different reason – that temperature gradients are the primary reason for the inhomogeneities in the autoignition process. It was observed that before the main ignition of the mixture, a first-stage smaller-scale reaction, called “cool flame” was also present for some hydrocarbons. It was found that the pressure required to initiate the first-stage reaction was a linear function of the compression pressure at TDC, while depending on the fuel, the required compression pressure to initiate autoignition decreased with increasing fuel concentration. However, no analysis of the result was presented.

Onishi *et al*. in 1979 [27] were amongst the first researchers to investigate the possibility of using autoignition combustion as a combustion mode in an engine. They have applied autoignition combustion using gasoline in a two-stroke gasoline engine and named this process ATAC. They showed that there was very small Cycle-By-Cycle Variations (CBCV) in the peak combustion pressure, the reaction occurred spontaneously at many points and combustion proceeded slowly. They investigated the significance of the hydroxyl, OH, hydrated carbon and diatomic carbon radicals and showed that their concentration was significantly higher and that the radicals had a longer life than in a SI engine (40° life compared to 25°). They suggested that to attain ATAC, the quantity of the mixture and the A/F ratio must be uniform from cycle to cycle, the temperature of the mixture must be suitable and the cyclic variability of the scavenging process must be kept to a minimum to ensure the correct conditions of the residual gases remaining in the combustion chamber. They obtained satisfactory combustion over a wide range of A/F from 11 to 22 and they concluded that ATAC reduces both fuel consumption and exhaust emissions over the whole of that range.

Around the same time, the autoignition and energy release processes of CIHC combustion and what parameters affect them were investigated using a single-cylinder four-stroke cycle Waukesha Cooperative Fuel Research (CFR) engine with a pancake combustion chamber and a shrouded intake valve [10]. It was deduced that this controlled autoignition/ combustion mode was not associated with knocking but a smooth energy release that could be controlled by proper use of temperature and species concentrations. In their experiments they controlled independently the intake charge temperature (600-810K) and the recirculated exhaust products (35-55% EGR), which were evaluated using carbon dioxide measurements. They used three different fuel; (a) 70% *iso*-octane and 30% *n*-heptane, (b) 60% *iso*-octane and 40% *n*-heptane and (c) 60% *iso*-propylbenzene and 40% *n*-heptane), and it was concluded that:

- Chemical species in the EGR gases had no effect on the rate of energy release and therefore EGR was primarily used to control combustion by means of regulating the initial gas temperature.
- Delivery ratio affected the combustion process through changes in the concentrations of fuel and oxygen in the reacting mixture. Therefore, at high delivery ratios the energy release became violent and for a CR of 7.5:1, it was found that a delivery ratio of 45% was the maximum.
- Fuels with lower octane numbers were ignited more easily.

In 1989, Thring [31] investigated the possibility of autoignition combustion in a single-cylinder, four-stroke internal combustion engine by Labeco CLR and was the first to suggest using SI operation at high loads and HCCI at part load. Even though the term ATAC [27] and CIHC [10] were previously used to describe this autoigntion/combustion process, Thring used the term HCCI. Intake temperatures (up to 425°C), equivalence ratios (0.33-1.30), EGR rates (up to 33%) and both gasoline and diesel were used to explore the satisfactory operation regions of the engine. There were three regions of unsatisfactory operation labelled "misfire region", "power-limited region" and "knock region." In the misfire and knock region the mixture was too rich while in the power-limited region the mixture was too lean. It was concluded that, under favourable conditions, HCCI combustion exhibited low cyclic variability and produced fuel economy results comparable with a diesel engine. However, high EGR rates (in the range of 30%) and high intake temperatures (greater than 370°C) were required.

HCCI combustion was later on also tested in a production engine [32] by using a 1.6 litre VW engine which was converted to HCCI operation with preheated intake air. By using λ=2.27, a CR of 18.7:1 and preheating the intake air up to 180°C, an increase in the part load efficiency from 14 to 34% was achieved. A NiCE-10 two-stroke SI engine with a CR of 6.0:1 was also used [33] to investigate this autoignition phenomenon by measuring the radical luminescence in the combustion chamber using methanol and gasoline as fuels. Luminescence images were acquired using an image intensifier coupled with a Charge-Coupled Device (CCD) camera and the luminescence spectra of the radicals OH, CH and C_2 were acquired by using a band-pass filter in front of the Ultra Violet (UV) lens. With conventional SI combustion, radical luminescence indicated a flame propagating from the centre of the spark plug towards the cylinder walls, while with ATAC combustion, radical luminescence appeared throughout the combustion chamber. The total luminescence intensity exhibited with ATAC combustion was less compared to SI combustion. Furthermore, with SI combustion OH radical species were formed 30° Crank Angle (CA) Before Top Dead Centre (BTDC) and assumed that it occurred at the same timing as the main combustion process, while in the case of ATAC combustion, OH radical species increased before the main combustion process as indicated by the rate of heat release.

This combustion phenomenon of premixed lean mixtures due to multi-point autoignition has also been given the name PCCI combustion [34]. A port-injected single-cylinder with a CR of 17.4:1 was operated at 1000RPM, with an initial mixture temperature of 29°C, an A/F ratio of 40 and gasoline as fuel and the multi-point autoignition combustion has been recorded by direct-imaging. The operation of PCCI combustion however was also limited by misfire in the lean range and knocking in the rich range.

Following the work of these early researches, a drive towards investigating further this autoignition phenomenon was initiated. In the following section, the fundamental parameters that affect HCCI combustion in IC engines are presented, and the term "HCCI" is used throughout, regardless of the terminology given by the individual researchers.

4. HCCI combustion fundamentals

In the last decades, extensive testing had been conducted on HCCI combustion in a race to develop a user-attractive HCCI engine-driven passenger car. Various ways have been employed ranging from trying different fuel combinations to supercharging the engine. An overview of the experimental work on HCCI combustion carried out is presented in this section. This section concludes with an overview of the operation maps produced by various research institutes to describe the effect of load and speed, amongst others, on engine performance and emissions under HCCI combustion mode and a presentation of a commercial HCCI engine.

4.1. Fuel and fuel additives

The difference between alcohols and hydrocarbons on HCCI combustion was studied [35] using 3 blends of unleaded gasoline, a Primary Reference Fuel (PRF) blend of 95% *iso*-octane and 5% *n*-heptane (95RON), methanol and ethanol. It was found that all three blends of gasoline behave in a very similar way even though their RONs were very different. Furthermore, the variations in paraffinic or aromatic content of the blend, did not affect HCCI combustion parameters. Finally, they concluded that:

- Hydrocarbons fuels showed a much lower tolerance to air and EGR dilution than alcohols.
- Higher thermal efficiencies were achieved with alcohols.
- IMEP covariance was smaller for alcohols for the same region of operation.
- NO_x emissions were minimal, with methanol exhibiting the lowest emissions.

The results obtained showed clearly that there was a difference between the various fuel types, due to differences in their oxidation kinetics. The effect of octane number of the fuels on HCCI was also investigated [36] with *iso*-octane, ethanol and natural gas (with octane numbers of 100, 106 and 120 respectively) as fuels. It was concluded with high octane number, high inlet air temperature and rich mixtures are required for autoignition. Furthermore, the levels of NO_x were found to decrease by at least 100 times and the levels of CO and HC emissions increased by factors of 2 and 20 respectively, compared to SI combustion. The effect of RONs and MONs on HCCI combustion has also been investigated with a variety of fuels, such as n-butane, PRF 91.8, PRF 70, indolene [37]. It was found that even though some fuels had identical RONs and similar MONs, they exhibited very different combustion characteristics with engine speed and inlet temperature and that the need to find a fuel property that will correlate with the ignition timing of the fuel under HCCI conditions was therefore apparent.

The effect of various additives was also the focus of various researchers. Water injection [44], [45], resulted in lower initial gas temperature and it was concluded that water injection can control the ignition timing and combustion duration. Water injection decreased the cylinder pressure, increased the combustion duration and retarded the ignition timing. However, the combustion efficiency decreased resulting in higher emissions of CO and HC, while the NO_x emissions decreased. Others [46] have studied HCCI combustion using hydrogen-enriched natural gas mixture. It was found that hydrogen affects the ignition timing, but large (>50%) H_2 mass fractions were needed at low inlet temperatures and pressures to achieve high loads; this is not feasible in a production engine. It was concluded that the natural gas/hydrogen mixture did not control the ignition timing as well as the heptane/*iso*-octane mixture [47], but at lower temperatures the efficiency was not sacrificed as much as with the heptane/*iso*-octane mixture. Furthermore, formaldehyde-doped lean butane/air mixtures [48] exhibited ignition timing retardation and a decrease in combustion efficiency – indicated by higher levels of CO concentration and lower levels of CO_2 concentration.

The use of reaction suppressors, namely methanol, ethanol and 1-propanol as additives was also investigated [45]. The suppression exhibited was due to their chemical effect on radical reduction during cool flame combustion. This was deduced by the fact that with increasing amount of alcohol injection, the magnitude of the cool flame combustion was significantly reduced and the cool flame timing retarded. For all suppressors under investigation, it was deduced that they had no effect on the reaction process with injection timings after the appearance of cool flame combustion, while when injected too early, they behaved more like a fuel (instead of a suppressor). Finally, the idea of switching from SI to HCCI combustion with only the addition of a secondary fuel to the main fuel supply was also investigated [49]. A natural gas-fuelled engine, with a fisher-tropsch naphtha fuel as the secondary fuel, was used. However, they have identified problems in the practicality of using two different fuels in a production engine and on the commercial availability of the FT naphtha fuel.

4.2. HCCI engine design

4.2.1. Variable Compression Ratio (VCR)

The effect of CRs ranging from 10:1 to 28:1 on various fuels was extensively studied [50],[51]. VCR can be achieved using a modified cylinder head that its position can be altered during operation using a hydraulic system. NO_x and smoke emissions were not affected by CR and were generally very low. However, an increased CR resulted in higher HC emissions and a decrease in combustion efficiency [50]. Others [52] reported that decreasing inlet temperatures and lambdas, higher CRs were need to maintain correct maximum brake torque and concluded that variable CR can be used instead of inlet heating to achieve HCCI combustion. Furthermore, the effect of CR on HCCI combustion in a direct-injection diesel engine was also investigated [53]. The CR could be varied from 7:5:1 to 17:1 by moving the head and cylinder liner assembly relative to the centreline of the crankshaft. Acceptable HCCI combustion was achieved with ignition timing occurring before TDC – with misfire being exhibited if ignition timing was further delayed – with CRs from 8:1 to14:1. However, with a knocking intensity of 4 (where audible knock occurs at 5 on a scale from zero to ten), the acceptable HCCI operation was limited at CRs from 8:1 to 11:1.

4.2.2. Supercharging and turbocharging

Supercharging (2bar boost pressure) was shown to increase the Indicated Mean Effective Pressure (IMEP) of an engine under HCCI combustion to 14bar [54]. Supercharging was used because of its capability to deliver increased density and pressure at all engine speeds while turbocharging depends on the speed of the engine. However, this resulted in lower efficiency due to the power used for supercharging. Supercharging resulted in greater emissions of CO and HC, greater cylinder pressure, longer combustion duration and lower NO_x emissions. There were no combustion related problems in operating HCCI with supercharging and the maximum net indicated efficiency achieved was 59%. On the contrary, others [55] investigated the effect of turbo charging on HCCI combustion. A Brake Mean Effective Pressure (BMEP) of 16bar (compared to 6bar without turbo charging and 21bar with the unmodified diesel engine) and an efficiency of 41.2% (compared to 45.3% with the unmodified diesel engine) were achieved. Furthermore, CO and HC emissions decreased with increasing load, but NO_x emissions increased. However, at higher loads, as the rate of pressure increased and the peak pressure approached their set limit (i.e. peak pressure greater than 200bar), ignition timing was retarded at the expense of combustion efficiency. Thus, in order to improve the combustion efficiency at high boost levels, cooled EGR rates was introduced [56], and it was shown that under those conditions, the combustion efficiency increased only slightly.

4.2.3. Exhaust Gas Recirculation (EGR)

Even though EGR has been employed by various researchers, the results are not always consistent within the research community. Depending on the method of EGR used (trapped exhaust gases due to valve timing, or exhaust gases re-introduced in the manifold), the results can vary, since both the temperature and chemical species present might not be the same in all cases.

Both aforementioned methods were employed [57],[58] where the first method relied on trapping a set quantity of exhaust gas by closing the exhaust valves relatively early, while in the second method, all the exhaust gases were expelled during the exhaust stroke, but during the intake stroke, both the inlet and exhaust valves opened simultaneously, to draw in the engine cylinder both fresh charge and exhaust gas. It was shown that HCCI combustion is possible with EGR and without preheating the inlet air and that increasing the quantity of exhaust gases advances the ignition timing. Furthermore it was concluded that HCCI can become reproducible and consistent by controlling the ignition timing by altering the EGR rate. Others achieved EGR [59],[60] by throttling the exhaust manifold, which increased the pumping work and reduced the overall efficiency. They concluded that:

- With increasing EGR, and thus decreasing A/F ratio and slower chemical reactions, the inlet gas temperature must also be increased
- With increasing amounts of EGR, the combustion process becomes slower, resulting in lower peak pressure and lower rate of heat release and therefore longer combustion rates.
- Both the combustion and gross indicated efficiencies increased with increasing EGR.

Based on further work [61], it was concluded that EGR had both thermal and chemical effects on HCCI combustion and that active species in the exhaust gases promoted HCCI. Others [62] however, reported contradicting results, where varying the EGR had little effect on combustion timing, on gross IMEP, combustion efficiency and net indicated efficiency. However, in those cases, the EGR was taken from the exhaust pipe and through a secondary pipe re-introduced in the inlet pipe where it was mixed with the fresh air mixture. There was no indication of pipe insulation or of the temperature of the EGR gases. Therefore, if the temperature of the gases was lower or of the same order as the intake gas temperature, then the effect of the EGR might have been reduced to only dilution effects.

Others on the other hand [63], investigated the importance of EGR stratification on HCCI combustion. It was found that HCCI combustion started near the centre of the combustion chamber at the boundary between the hot exhaust gases, situated at the centre due to poor scavenging characteristics of the valves, and the fresh intake charge. The importance of the mixing of the EGR and the fresh-air mixture was identified, since by controlling the EGR stratification, the combustion timing might also be controlled. The effect of homogeneous and inhomogeneous cooled EGR on HCCI combustion has also been investigated [64]. For the homogeneous case, the fresh air and EGR gases were mixed upstream of the intake port and thus well-mixed before the fuel injector. For the inhomogeneous case, EGR gases were introduced downstream the fuel injector and therefore there was no time for proper mixing.

With inhomogeneous EGR supply, autoignition timing was advanced (due to local hot spots of fresh air-fuel mixture) but the overall combustion was slower (due to local cold spots of exhaust gas-fuel mixture), than with homogeneous EGR supply.

4.3. Fuel injection strategies

Fuel injection strategies is one of the most important topics under research for HCCI combustion, as it can be easily controlled, compared to VCR, multiple fuel injection, etc, to alter HCCI combustion, by varying the injection timing and duration, and the injector location and type. It was shown that even injector nozzle optimizations can be employed to alter the fuel spray and affect HCCI combustion [65]. Injector location was also investigated [66] by using both port injection – to create a premixed fuel-air mixture – and direct injection – to control the timing of autoignition. Others [67], focused on different fuel injection strategies; injecting the fuel in a 20 litre mixing tank before the engine intake port and injecting the fuel just outside the engine intake port. The first treatment resulted in a homogeneous mixture, while the second treatment resulted in a mixture with fluctuations of the order of 4 to 6mm. Regardless of the preparation method however, combustion was inhomogeneous with very large spatial fluctuations. Furthermore, the local combustion kernels did not have a tendency to be more frequent in the central part of the combustion chamber, where the temperature was assumed to be higher than in the vicinity of the walls. They were unable though to identify the process that caused the very inhomogeneous combustion initiation.

Others also investigated the effect of various injection patterns and their combination on HCCI combustion. In particular [68], the following three fuel injection patterns were investigated: (i) Injection during the negative valve overlap interval to cause fuel reformation, (ii) injection during the intake stroke to form a homogeneous mixture and (iii) injection during the compression stroke to form a stratified mixture. It was found that with fuel reformation, the operating range of HCCI combustion was extended without an increase in the NO_x emissions. Furthermore, limited operation was observed with late injection timing that also led to high NO_x emissions. Two other injection systems were also employed [69]: (i) a premixed injection injector in the intake manifold to create a homogeneous charge and (ii) a DI injector to create a stratified charge. Thus by varying the amount of fuel injected through the DI injector (from 0 to 100%) and varying the injection timing of the DI injector (from 300 to 30°CA BTDC) different stratification levels were achieved. It was found that HCCI combustion was improved at the lean limit with charge stratification, while CO and HC emissions decreased. On the contrary, at the richer limit, a decrease in combustion efficiency was evident at certain conditions. It was concluded that charge stratification causes locally richer regions that, in the lean limit, improved combustion efficiency by raising the in-cylinder temperature during the early stages of the autoignition process, while at the rich limit, the change in the in-cylinder temperature does not affect the combustion efficiency to such an extent.

The possibility of using a Gasoline Direct Injection (GDI) injector and varying the injection timing to control HCCI combustion has also been investigated [70]. It was concluded that the most homogeneous mixture was formed with early injection timings, while fuel inhomogeneities (and thus regions with richer fuel concentration) were present with retarded injection timings. With retarded injection timing and thus increased fuel inhomogeneity, combustion of locally richer mixtures caused an increase in the combustion temperature that as a result, caused a higher combustion efficiency, an increase in NO_x emissions but a decrease in CO and HC emissions. Furthermore, with late retarded injection timings, a decrease in the combustion efficiency (and increase in the CO and HC emissions) was observed due to fuel impingement on the piston surface. It was concluded that fuel stratification can be used to improve HCCI combustion under very lean conditions but that great care is needed to avoid the formation of NOx due to locally near-stoichiometric fuel concentrations.

4.4. Operational limits and emissions

With stable HCCI combustion over a range of CRs, fuels, inlet temperatures and EGR rates, operation maps of HCCI operation have been produced by various researchers for a wide number of production engines. The effect of these parameters on BMEP, IMEP, combustion efficiency, fuel economy and NO_x, HC and CO emissions has been analysed in detail. There is a vast, and some time contradicting, background literature especially on emissions and in the present subsection, no assumptions have been made on the author's behalf; the data is presented in this subsection as analysed by the various researchers. This subsection is not aimed to act a complete review on all the experiments conducted on all engines, but to present to the reader the complexity in analysing HCCI engine operation.

The modified Scania DSC12 engine was used [47] to run a multi cylinder engine in HCCI mode and to provide quantitative figures of BMEP, emissions and cylinder-to-cylinder variations. The engine was run at 1000, 1500 and 2000RPM and various mixtures of *n*-heptane and *iso*-octane were used to phase the combustion close to maximum BMEP. A BMEP of up to 5bar was achieved by supplying all cylinders with the same fuel, but for higher loads, the fuel injected in each cylinder had to be individually adjusted as small variations led to knocking in individual cylinders. Even though a wide load range (1.5 to 6.15bar) was achieved with no preheated air, preheating at low loads improved the CO and HC emissions. It was concluded that HCCI was feasible in a multi cylinder engine and that the small temperature and lambda cylinder-to-cylinder variations were acceptable. However, it would be impractical to alter the fuel mixture in a commercial engine in order to vary the octane number, as was done in the experiments.

A naturally-aspired Volkswagen TDI engine with propane as fuel, was used [71] to investigated the effect of different fuel flow rates and intake gas temperature on BMEP, IMEP, efficiency and CO, HC and NO_x emissions. It was concluded that:

- Combustion efficiency increased with increasing fuel flow rate or increasing intake gas temperature.
- NO_x emissions increased with increasing fuel flow rate and increasing intake gas temperature.
- CO and HC emissions decreased with increasing fuel flow rate and increasing intake gas temperature.

Furthermore, at the lowest intake gas temperature operating point, the combustion process varied considerably from cylinder to cylinder, but became more consistent with time as the engine temperature increased.

Allen and Law [72] produced operation maps of the modified Lotus engine under HCCI combustion when running at stoichiometric A/F ratio. The operational speed range was found to lie between 1000-4000RPM with loads of 0.5bar BMEP at higher speeds and 4.5bar at lower speeds. The limitation at high speeds was due to knocking while at low speeds it was though to be due to insufficient thermal levels in the cylinder due to the very small amount of fuel being burned. It was concluded that compared with SI combustion:

- Fuel consumption was reduced by up to 32%.
- NO_x emissions were reduced by up to 97%.
- HC emissions were reduced by up to 45%.
- CO emissions were reduced by up to 52%.

The HCCI operating range with regards to A/F ratio and EGR and their effect on knock limit, engine load, ignition timing, combustion rate and variability, Indicated Specific Fuel Consumption (ISFC) and emissions for the Ricardo E6 engine were also produced [7],[25]. Comprehensive operating maps for all conditions were produced and the results were compared with those obtained during normal spark-ignition operation. From their experiments they were able to conclude the following:

- A/F ratios in excess of 80:1 and EGR rates as high as 60% were achieved.
- ISFC decreased with increasing load.
- IMEP increased with decreasing lambda.
- NO_x emissions were extremely low under all conditions.
- HC emissions increased near the misfire region at high EGR rates.
- CO emissions increased with increasing lambda and EGR rate.
- ISFC increased with increasing lambda due to oxygen dilution and decreasing combustion temperatures.

A 4-stroke multi-cylinder gasoline engine based on a Ford 1.7L Zetec-SE 16V engine was used to achieve HCCI combustion [73],[74]. The engine was equipped with variable cam timing on both intake and exhaust valves, and it was found that internal EGR alone was sufficient to induce HCCI combustion over a wide range of loads and speeds (0.5 – 4BMEP and 1000 – 3500RPM). All the tests were conducted using unleaded gasoline. It was concluded that:

- BMEP decreased slightly with increasing lambda.
- Brake Specific Fuel Consumption (BSFC) decreased as lambda changed from rich to stoichiometric but increased as the mixtures becomes leaner.
- CO emissions decreased while HC emissions increased with increasing lambda.

An operational maps for HCCI combustion at λ=1 for various loads and speeds was also constructed. The upper load limit was limited by the restrictions of gas exchange due to the operation of the special cam timings and not due to knocking that did not occur at the upper limit. The lower load limit was limited by misfire due to too much residual gases and to very low temperatures. The BSFC did not change with speed but decreased with increasing load. NO_x and CO emissions did not vary with speed, while HC emissions decreased with increasing speed. The results obtained with HCCI combustion were compared with SI results and they concluded that:

- Both HCCI and SI exhaust temperatures increased with increasing load and speed.
- HCCI combustion showed a maximum of 30% reduction in BSFC.
- There was a reduction of 90-99% in NO_x and 10-40% in CO but an increase of 50-160% in HC emissions with HCCI combustion.

A Caterpillar 3401 single cylinder engine running on gasoline was used to study the effect of fuel stratification on NO_x, HC, CO and smoke emissions [75],[76]. With retarded Start Of Injection (SOI) and therefore increased fuel stratification, HC emissions decreased (compared to early SOI) indicating improvement in combustion efficiency, NO_x emissions increased at late SOI indicating increased local combustion temperatures, soot increased due to fuel impingement, indicated by carbon deposit on the piston surface while CO and indicated efficiency remained constant. Furthermore HC emissions decreased while NO_x emissions increased with higher load and later SOI and CO emissions decreased with higher load and earlier SOI. Further results showed that combustion efficiency increased with increasing load and fuel stratification. At low loads and decreased fuel stratification, efficiency fell to as low as 91%. HC emissions decreased with increasing load and fuel stratification, while CO emissions decreased with increasing load, indicating more complete oxidation of the fuel due to the higher temperatures. NO_x emissions were low and did not affect the combustion efficiency.

The effect of very lean HCCI combustion (φ=0.04 to 0.28) on CO and HC emissions and combustion efficiency has been investigated in a Cummins B-series diesel engine with *iso*-octane as fuel over a range of intake temperatures, engine speeds, injection timings and intake pressures [77]. It was found that CO emissions start to increase dramatically for φ<0.2 while CO_2 emissions decrease and the combustion efficiency decreases from 95% down to 30%. HC emissions also start to increase for φ<0.14. This result indicated that for very lean combustion, CO and HC emissions are not only formed in the crevices and in the boundary layers but are also produced due to incomplete combustion in the bulk region of the combustion chamber. It was also found that engine speed and intake pressure have almost no effect on CO and HC emissions. Finally, higher equivalence ratios were needed for complete combustion with decreasing intake temperature because more combustion heat release was needed to reach the same combustion temperature as with higher intake temperatures and due to retarded combustion timing resulting to less time for complete reaction before expansion.

A diesel engine with a CR of 16.5:1 was also modified to operate with gasoline in both SI and HCCI combustion modes in order to investigate the possibility of a hybrid SI/HCCI engine

[72]. Specifically, the effect of HCCI combustion on thermal efficiency, IMEP, COV of IMEP and CO, HC and NO_x emissions for a wide range of BMEP and engine speeds was studied. It was found that under all operating conditions the COV of IMEP was very low (less than 2.5%), NO_x emissions were very low while CO and HC emissions were rather high. In the operating window of BMEP 6-8bar – where the hybrid engine would operate under HCCI combustion mode – the highest brake thermal efficiency was achieved with the minimum emissions. However, with decreasing load and especially near idle conditions, the brake thermal efficiency was very low and CO and HC emissions increased even further.

4.5. HCCI engines in production

A motorcycle with a two-stroke engine that operates in a hybrid SI/HCCI mode has been developed by Honda R&D CO., Ltd [9],[73]-[78]. However, in two-stroke engines, the term Active Radical Combustion (ARC) is used instead of HCCI to describe the phenomenon of autoignition. Two-strokes engines over perform four-stroke engines in weight, compactness and higher specific power output, but under perform in fuel economy and high HC emissions. These shortcomings are due to the fresh fuel-air mixture which short-circuits the cylinder directly to the exhaust system during the scavenging process and incomplete combustion at low load operation. ARC was achieved by taking advantage of the exhaust gases trapped in the combustion cylinder. The original two-stroke engine was modified to include a throttle in the exhaust, and by varying the throttle position, ARC was achieved at lows loads and its timing controlled. The ARC two-stroke motorcycle with a displacement of 403cm^3 was tested and used in the Grenada-Dakar Rally 95 and it was shown to have better fuel economy, HC emissions and durability than the 780cm^3 four stroke racer (which held a series of championships) under the given conditions. Furthermore, the two-stroke engine would operate in the ARC mode for up to 75% of the time for low loads (0-35% of throttle opening) and a wide range of speeds (2000-5000RPM). With the intention of commercialisation of the AR engine, ARC was tested in a 250cm^3 motorcycle, and it was shown to reduce HC emissions by 60% and under 50kh/h cruise conditions and A/F=15, fuel economy was improved by 57%. Fuel efficiency was further improved in the ARC engine by the introduction of a low pressure Pneumatic Direct Injection (PDI) injector that reduced the amount of the fuel short-circuiting the cylinder directly to the exhaust system. The final 250cm^3 hybrid ARC/SI two-stroke motorcycle with the PDI injector exhibits 23% fuel economy compared to the four-stroke engine with the same displacement without a large increase in manufacturing cost.

5. Theory on controlling HCCI combustion

According to previous research [3]-[6], the autoignition process was a random multiple-autoignition phenomenon that started throughout the combustion chamber, possibly at locations of maximum interaction between the hot exhaust gases and the fresh fuel–air mixture [7]. In other cases, however, a uniform autoignition front was observed [8]. Thus, a lot of research has focused on investigating the propagation speed and spatial development of the autoignition process, and how these parameters can be altered to control HCCI combustion.

Using a high CR engine and PLIF [79], the autoignition front propagation was investigated experimentally. It was found that with HCCI combustion there were no sharp edges in the intensity histogram of the PLIF images indicating that the transition from fuel to products was a gradual process. Furthermore the global propagation speed was found to be 82m/s while the growth of small autoignition sites showed that the local propagation speed was of the order of 15m/s. Similar speeds have been measured in the development of self-ignited centers in the unburned end-gas ahead of a flame front in a SI engine [80]. It was shown that the propagation speed of these self-ignited centers was in the range of 16-25m/s, and thus they have concluded that, under their engine conditions, the self-ignition was not driven by a shock-wave (i.e. no knocking was observed). Similar propagation speeds has also been shown in HCCI engines by others as well, both computationally [81] and experimentally with fast camera imaging [82].

Various techniques and computational models have also been used to investigate the parameters that affect the spatial development of autoignition. PLIF was used [67] to obtain imaging of fuel and hydroxyl radicals in order to investigate the extent to which charge homogeneity affected the combustion process in an HCCI engine. Regardless of the preparation method, LIF of both OH and fuel showed that combustion was inhomogeneous with large spatial and temporal variations. Both direct imaging and PLIF [83] were used to investigate the effect of the stratification of burned gases on spatial development of autoignition. It was found that combustion started near the centre of the combustion chamber at the boundary between the hot exhaust gases, situated at the centre owing to poor scavenging characteristics of the valves, and the fresh intake charge. Charge inhomogeneity was also investigated using chemiluminescence measurements [84]. In the homogeneous case, luminescence was observed for a short duration over a large spatial area of the combustion chamber while luminescence appeared locally over a wider time period in the inhomogeneous case. They reported that varying the charge inhomogeneity could be used as a method for controlling the combustion duration in HCCI engines. Similar results were acquired by others, where the autoignition process was spatial uniform, and this uniformly decreased with increasing the inhomogeneity in the charge [85].

Computationally, mathematical analysis has been performed [86],[87] to investigate the spreading of “hot spots” (autoignition regions of high temperature, which may have been caused by a chemical reaction) to the surrounding colder gases. Depending on the temperature gradient across the “hot spots”, they have shown that the autoignition front moves into the unburned mixture at either approximately the acoustic speed, leading to a developing detonation, or at a lower speed (higher than flame propagation), leading to either autoignitive deflagration or thermal explosion where autoignition is driven by the ignition delay and not by molecular transport processes. It was shown that a thermal explosion occurred at very low temperature gradients, a developing detonation occurred at a specific medium temperature gradient, and a deflagration occurred at high temperature gradients. The effect of inhomogeneities of EGR on the spatial autoignition process has also been investigated computationally [88]. A temperature profile was created by distributing the EGR gases at different locations within the engine cylinder. When EGR gases were distributed near the wall of the cylinder (lower temperature zone) (and thus the fuel mixture was concentrated near the centre of the cylinder) HCCI combustion was improved in comparison with the homogeneous EGR distribution case. When gases from EGR were concentrated near the centre of the cylinder (higher-temperature zone) (and thus the fuel mixture was distributed near the wall of the cylinder) HCCI combustion became slower in comparison to the homogeneous EGR distribution case.

Based on the above research, a theory is being proposed and analyzed in the present section on a possible mechanism of controlling HCCI combustion in a production engine. A possible explanation of the aforementioned discrepancies on the uniformity and propagation of HCCI combustion might be accounted to differences in the CR of the engine, the inlet conditions, and the mixing of "hot" gases and the injected "fuel". Let us first consider an engine with a high CR and with low temperature gradients, where where the possible increasing temperature distributions through an arbitrary line in the combustion chamber are shown in Figure 2. The temperatures shown are not based on experimental data or calculations and are used for illustration purposes. Figure 2 shows multiple spatial autoignition sites at the locations of maximum temperature that rapidly consume the whole mixture in an apparent absence of an autoignition front. The combustion process is therefore primarily driven by the increasing temperature and pressure due to the CR.

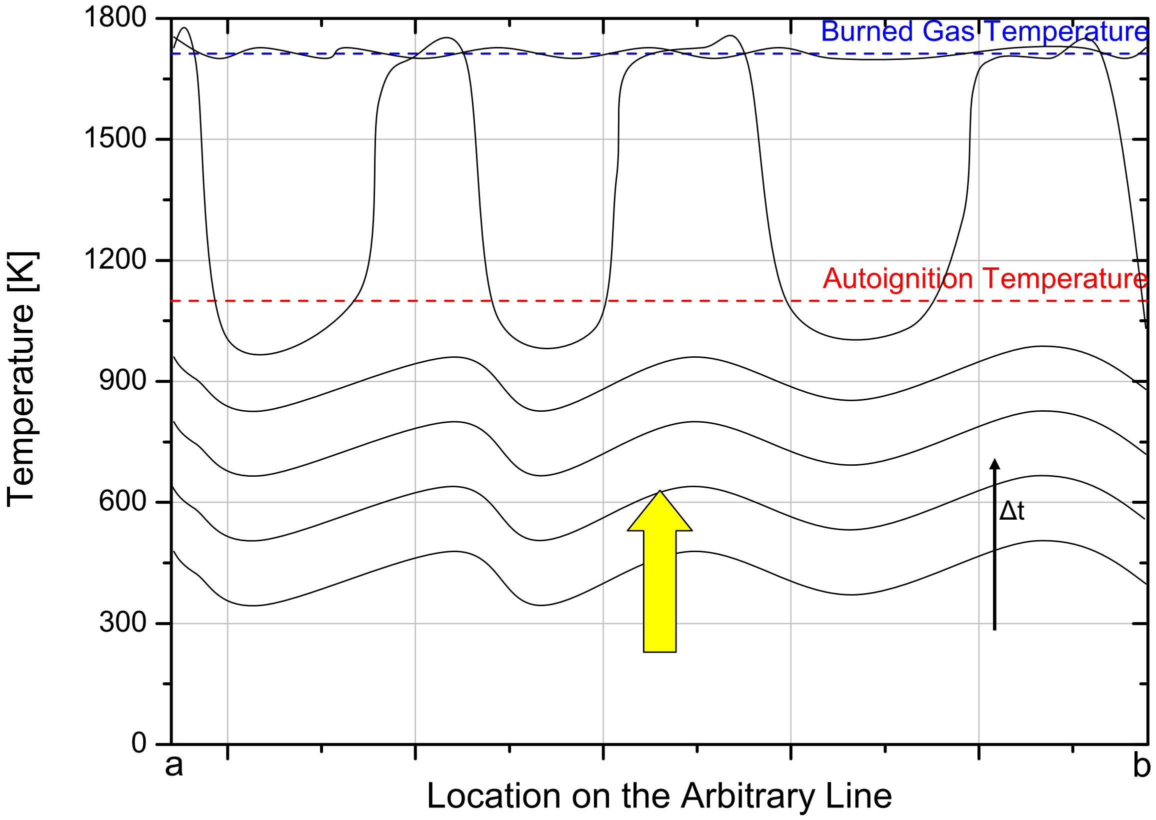

Figure 2. Possible Temperature Distribution in a High CR Engine and with Low Temperature Gradients through an Arbitrary Line in the Combustion Chamber: Black Lines indicate the Increase in Temperature per Δt; Yellow Arrow indicates the Magnitude of Temperature Increase due to Compression.

Les us now consider an engine with the same CR but with higher temperature gradients (due to either EGR or inlet heating), where the possible increasing temperature distributions through an arbitrary line in the combustion chamber are shown in Figure 3.

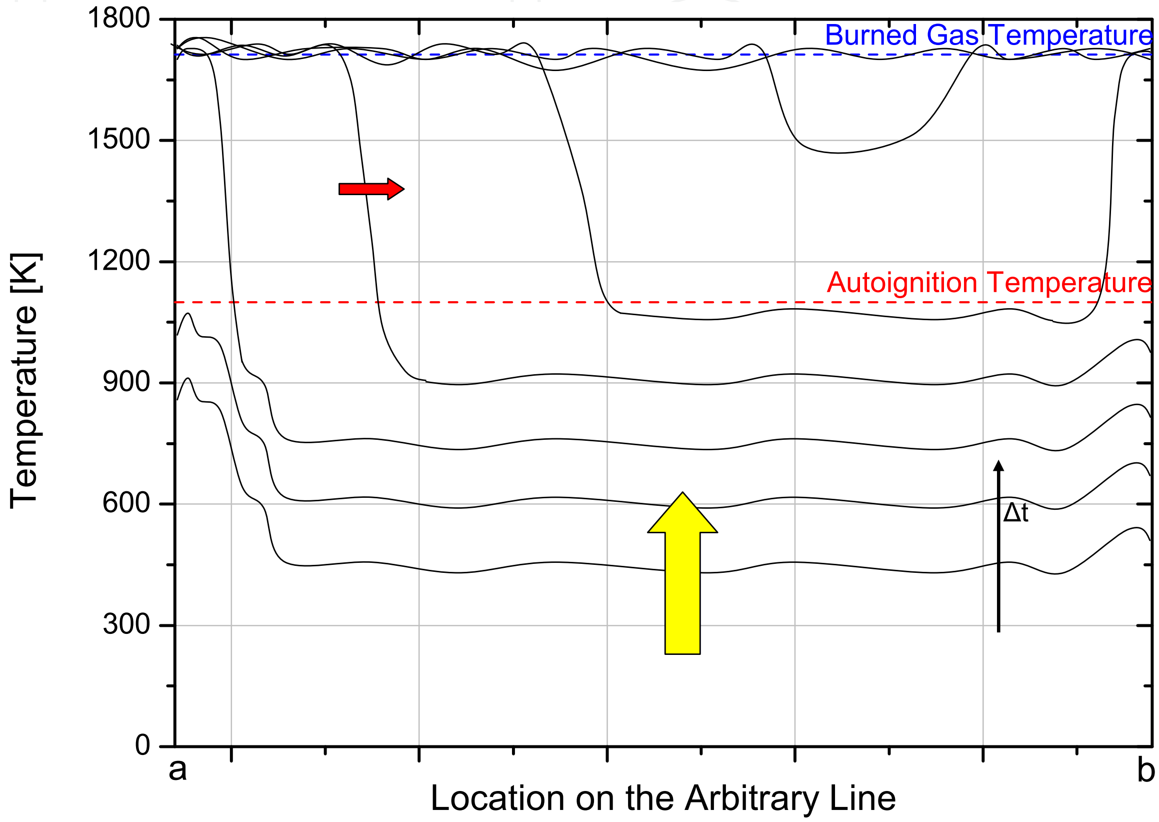

Figure 3. Possible Temperature Distribution in a High CR Engine and with High Temperature Gradients through an Arbitrary Line in the Combustion Chamber: Black Lines indicate the Increase in Temperature per Δt; Yellow Arrow indicates the Magnitude of Temperature Increase due to Compression; Red Arrow indicates the Magnitude of Temperature Increase due to Diffusion from the Burned Gases.

As can be seen from Figure 3, autoignition occurs earlier due to the higher temperatures present in the combustion chamber. However, there is a difference in the way the autoignition process develops. Since some gases combust earlier than the adjacent gases, the possibility of some heat transfer occurring from the high-temperature burned gases to the low-temperature unburned-gases is possible. However, with high CRs, the diffusion rate is very small and is usually neglected from calculations. Again, the multi-point nature of autoignition nature of HCCI combustion can be observed. However, with decreasing CRs, a balance between the diffusion rate and the increase in temperature and pressure due to compression might be possible, and we can now consider an engine with a relatively low CR with high temperature gradients, where the possible increasing temperature distributions through an arbitrary line in the combustion chamber are shown in Figure 4.

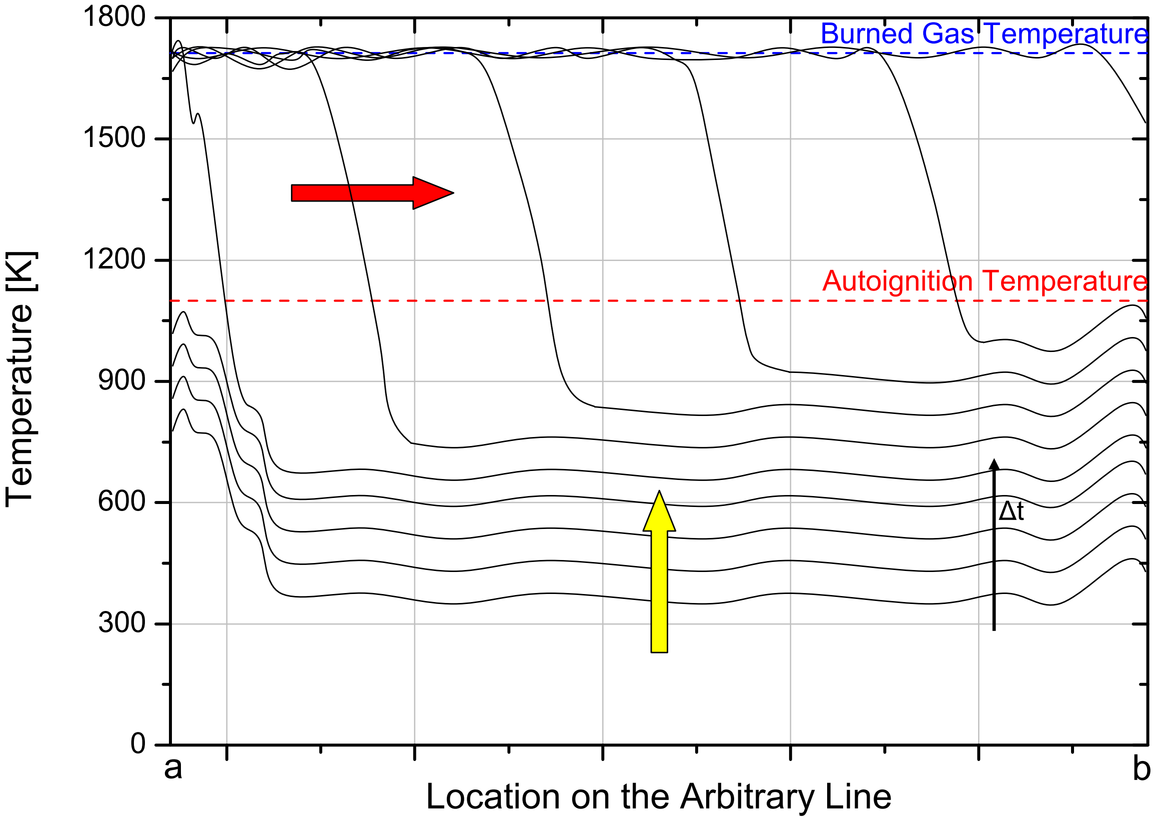

Figure 4. Possible Temperature Distribution in a Low CR Engine and with High Temperature Gradients through an Arbitrary Line in the Combustion Chamber: Black Lines indicate the Increase in Temperature per Δt; Yellow Arrow indicates the Magnitude of Temperature Increase due to Compression; Red Arrow indicates the Magnitude of Temperature Increase due to Diffusion from the Burned Gases.

Figure 4 shows the same temperature distribution as in Figure 3, but in an engine with a lower CR. This results to a retarded autoignition and longer combustion duration, since more time is needed for the whole fuel/air mixture to reach its autoignition temperature. Diffusion now plays a more dominant role in increasing the temperature of the unburned mixture compared to the cases of higher CR. Therefore, in the cases where only one spatial location of high temperature is present in the combustion chamber, and the temperature of the rest of the mixture is low enough as to not autoignite due to compression before being "reached" by the autoignition front, then a uniform autoignition front is possible.

Thus, altering the temperature distribution in a combustion chamber can therefore offer the possibility of controlling HCCI combustion. At low loads, HCCI combustion is limited by misfire and incomplete combustion and at high loads, by knocking or high NO_x. By creating an "extreme" temperature distribution in the combustion chamber, as shown in Figure 5, HCCI combustion timing and duration can be controlled.

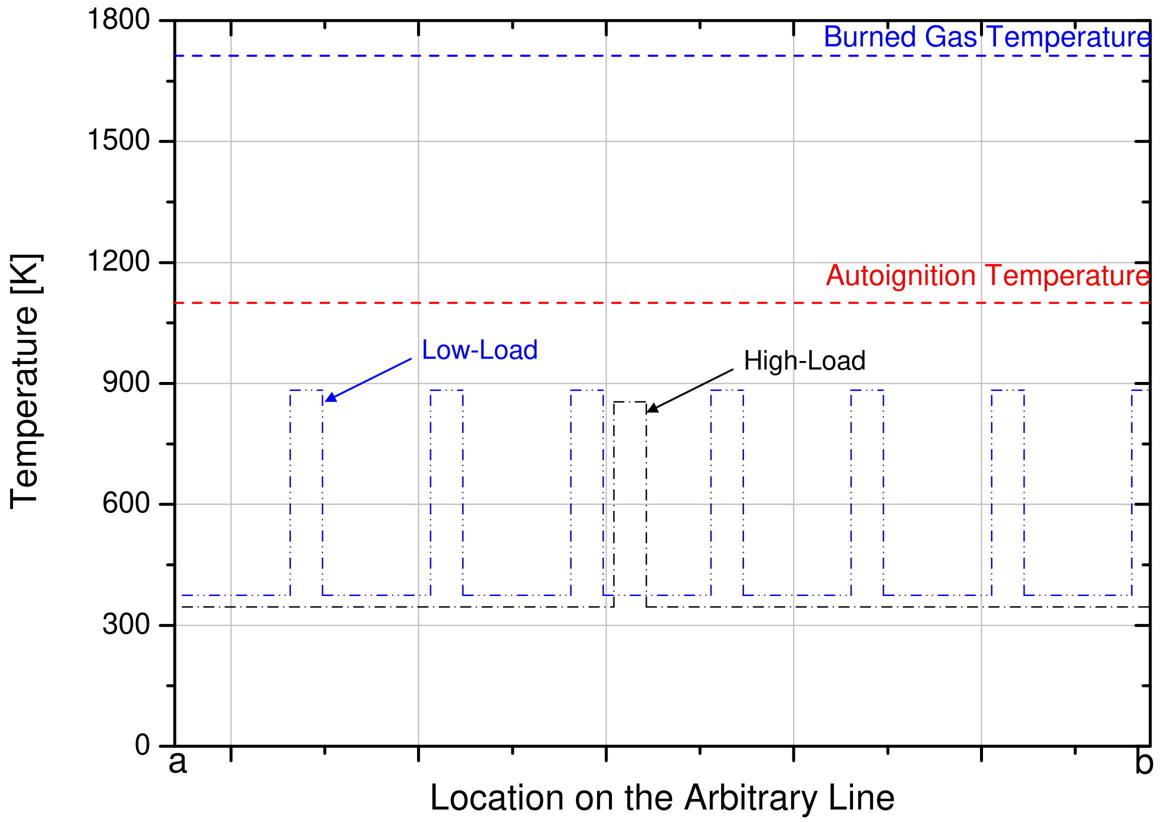

Figure 5. Extreme Temperature Distribution to control HCCI Combustion at Low and High Loads through an Arbitrary Line in the Combustion Chamber: Black Dotted Line indicates Temperature Distribution at High Loads; Blue Dotted Line indicates Temperature Distribution at Low Loads

At high loads, where richer mixtures are needed, a single high-temperature region might be needed to drive the autoignition process, while the rest of the combustion chamber can be kept at a low enough temperature as to not combust before being "reached" by the autoignition front. Therefore, the combustion process can be slowed down and it might be possible to be as slow as SI combustion (i.e. the high-temperature region acting as a spark). It would be further advantageous if at the location of maximum temperature the A/F ratio is as lean as possible but still provide enough energy to drive the combustion process. On the contrary, at low loads, where leaner mixtures are used, multiple high-temperature regions would be needed to control the timing and duration of the autoignition process. The simultaneous ignition of multiple points would compress the remaining gases and further increase their temperature and the in-cylinder pressure that would lead to a more stable combustion process.

6. Conclusions

HCCI is the most commonly used name for the autoignition of various fuels and is one of the most promising alternatives to SI combustion and CI combustion. In an IC engine, HCCI combustion can the achieved by premixing the air-fuel mixture and compressing it until the temperature is high enough for autoignition to occur. HCCI combustion can be described by the oxidation of the fuel driven solely by chemical reactions governed by chain-branching mechanisms and two temperature regimes exist for these reactions – one below 850K (low temperature oxidation or cool flame combustion) and one around 1050K (high temperature oxidation or main combustion). This autoignition phenomenon has been the focus of various researchers since the early 20th century.

Since under HCCI combustion the fuel/air mixture does not rely on the use of a spark plug or direct injection near TDC to be ignited, overall lean mixtures can be used resulting to high fuel economy. Thus, the combustion temperature remains low and therefore NO_x emissions decrease significantly compared to SI and CI operation. Under optimum operating conditions, HCCI combustion can offer comparable carbon CO and HC emissions with SI and CI combustion, but under very lean conditions – and thus low combustion temperatures (approximately below 1500K) – incomplete combustion can occur in the bulk regions leading to partial oxidation of the fuel, decrease in combustion efficiency and increase in CO and HC emissions. Furthermore, since a homogeneous fuel/air mixture can be prepared in the manifold, low soot can be achieved. However, when HCCI combustion is operated at richer fuel/air mixtures, knocking can occur. HCCI combustion is therefore limited by these two main regimes: (a) Lean A/F ratio limit – Leading to incomplete combustion, which results to low power and high HC and CO emissions and (b) Rich A/F ratios limit – Leading to knocking if the rate of pressure rise is too high causing damage to the engine or high NO_x emissions due to high combustion temperatures. Various parameters, namely VCR, EGR ratio and composition, fuel additives, inlet temperature and fuel stratification and their effect on the magnitude, timing and emissions associated with HCCI combustion have been the focus of various research institutes. A VCR engine has been introduced but it has not yet been shown to effectively control HCCI at the limits of misfire or knocking. EGR gases can be used to alter the timing of autoignition due to their temperature and the duration of autoignition due to dilution effects. Fuel additives work effective at either suppressing knock, or enhancing the ignitability of various fuels, but more work is still needed to find the appropriate fuels to expand the operation region of HCCI engines. Varying the inlet temperature with the use of inlet heaters can alter the combustion timing, but have a generally low response and can not be used in transient operations. Furthermore, it has been shown that by varying the injection timing and/or by varying the opening of the inlet and exhaust valves, HCCI combustion can be controlled on a cycle-by-cycle basis in a production engine.

Finally, in the present chapter, a theory was also proposed on a possible way of controlling HCCI through temperature stratification, where, at high loads, a local high-temperature inhomogeneity (i.e. like spark discharge) would be the driver of a uniform, slow-propagating HCCI combustion. On the other hand, at low loads, multiple temperature inhomogeneities can be introduced in the combustion cylinder to simultaneously ignite the fuel mixture at multiple locations, thus improving the stability of HCCI combustion.

It is the author's belief that the future of HCCI engines looks promising in two different paths. On one hand, dual engine operation might be able to be achieved (SI/CI and HCCI) with electronic control of the valve timing and of the injection strategy (timing and duration of the injection), but other methods (VCR, dual-fuel, etc) might not be as applicable in a production line engine. On the other hand, the possibility of using HCCI engine in combined heat and power engines for home use, where the operating conditions are less variable and no incomplete combustion and/or knocking problems will be encountered, should be evaluated

Abbreviations

A/F	Air to Fuel ratio
ARC	Active Radical Combustion
ATAC	Active Thermo-Atmosphere Combustion
BMEP	Brake Mean Effective Pressure
BSFC	Brake Specific Fuel Consumption
BTDC	Before Top Dead Centre
CAI	Controlled Auto Ignition
CBCV	Cycle-By-Cycle Variations
CCD	Charge-Coupled Device
CFR	Cooperative Fuel Research
CI	Compression Ignition
CIHC	Compression-Ignited Homogeneous Charge
CO	Carbon MonOxide
CR	Compression Ratio
DI	Direct Injection
EGR	Exhaust Gas Recirculation
GDI	Gasoline Direct Injection
HC	Hydro Carbon
HCCI	Homogeneous Charge Compression Ignition
HCDC	Homogeneous Charge Diesel Combustion
IC	Internal Combustion
IMEP	Indicated Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
MON	Motor Octane Number
NO_x	Nitrogen Oxides
OI	Octane Index
PCCI	Premixed Charge Compression Ignition

PLIF	Planar Laser-Induced Fluorescence
PPM	Parts Per Million
PREDIC	PREmixed lean DIesel Combustion
PRF	Primary Reference Fuel
RON	Research Octane Number
RPM	Revolutions Per Minute
SI	Spark Ignition
SOI	Start Of Injection
TDC	Top Dead Centre
UV	Ultra Violet
VCR	Variable Compression Ratio

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