Review Article

Homogeneous Charge Compression Ignition Combustion: Challenges and Proposed Solutions

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Engine and car manufacturers are experiencing the demand concerning fuel efficiency and low emissions from both consumers and governments. Homogeneous charge compression ignition (HCCI) is an alternative combustion technology that is cleaner and more efficient than the other types of combustion. Although the thermal efficiency and NO_x emission of HCCI engine are greater in comparison with traditional engines, HCCI combustion has several main difficulties such as controlling of ignition timing, limited power output, and weak cold-start capability. In this study a literature review on HCCI engine has been performed and HCCI challenges and proposed solutions have been investigated from the point view of *Ignition Timing* that is the main problem of this engine. HCCI challenges are investigated by many IC engine researchers during the last decade, but practical solutions have not been presented for a fully HCCI engine. Some of the solutions are slow response time and some of them are technically difficult to implement. So it seems that fully HCCI engine needs more investigation to meet its mass-production and the future research and application should be considered as part of an effort to achieve low-temperature combustion in a wide range of operating conditions in an IC engine.

1. Introduction

Although electric and hybrid vehicles (EVs and PHEVs) have emerged on the market, still the internal combustion engines are the most popular automotive power plant. However, in recent decades, serious concerns have piled up considering the environmental impact of the gaseous and particulate emissions arising from operation of these engines. As a result, ever tightening legislation, that restricts the levels of pollutants that may be emitted from vehicles, has been introduced by governments around the world. In addition, concerns about the world's finite oil reserves and CO_2 emissions have led to heavy taxation of road transport, mainly via on duty on fuel. These factors have led to massive pressure on vehicle manufacturers to research, develop, and produce ever cleaner and more fuel-efficient vehicles [1].

Over the last decade, an alternative combustion technology, commonly known as homogeneous charge compression ignition (HCCI), has emerged and it has the potential to

decrease emissions and fuel consumption in transportation [2, 3]. HCCI is a clean and high efficiency technology for combustion engines that can be scaled to any size-class of transportation engines as well as used for stationary applications [4]. These benefits of HCCI (especially relative to spark ignition engines) are acquired by virtue of lean/dilute operation.

The two dominating engine concepts commonly used today are the diesel and SI engines. A comparison between the two engines shows that the SI engine equipped with a catalytic converter provides low emissions but lacks in efficiency. The diesel engine on the other hand provides high efficiency but also produces high emissions of NO_x and particles. An engine concept capable of combining the efficiency of a diesel engine with the tailpipe emissions level of an SI engine is the homogeneous charge compression ignition (HCCI) engine [5]. In other words, HCCI is the autoignition of a homogeneous mixture by compression.

The following literature review has focused on HCCI challenges and proposed solutions from the point view of

Ignition Timing as the most critical problem of HCCI engine. This point of view has been tried to be discussed through the paper as its particular characteristic. At first, previous studies in the field of HCCI engine including two-stroke and four-stroke HCCI engines are discussed. Next, HCCI challenges and proposed solutions are reviewed. Finally, HCCI ignition timing as the most important problem of HCCI is considered and the main controlling methods such as mixture dilution, changing fuel properties, fast thermal management, and direct injection are presented.

2. HCCI/CAI Engine

The homogeneous charge compression ignition (HCCI) or controlled autoignition (CAI) combustion has often been considered a new combustion process amongst the numerous research papers published over the last decade. However, it has been around perhaps as long as the spark ignition (SI) combustion in gasoline engine and compression ignition (CI) combustion in diesel engines [1].

In the case of gasoline engines, the HCCI combustion had been observed and was found responsible for the “after-run”/“run-on” phenomenon that many drivers had experienced with their carbureted gasoline engines in the sixties and seventies, when a spark ignition engine continued to run after the ignition was turned off [1].

In the case of diesel engines, the hot-bulb oil engines were invented and developed over 100 years ago. In these engines, the raw oil was injected onto the surface of a heated chamber called hot-bulb. This early injection gives the fuel lots of time to vaporize and mix with air. The hot-bulb had to be heated on the outside for the start-up and once the engine had started, the hot-bulb was kept hot by using the burned gases. Later design placed injection through the connecting passage between the hot-bulb and the main chamber so that a more homogeneous mixture could be formed, resulting in auto-ignited homogeneous charge combustion [6].

2.1. Two-Stroke HCCI Engine. For solving one of the main problems of the two-stroke engine which was the unstable, irregular, and incomplete part load combustion responsible for excessive emissions of unburned hydrocarbons, a significant research work was performed from the end of the 1960s to the end of the 1970s [1]. Lots of studies were performed during this period by Jo et al. to investigate the part load lean two-stroke combustion [7]. He found that the irregularities of the combustion and the autoignition which were considered as the weak points of the two-stroke engine could be effectively controlled. This period was successfully concluded by the innovative work he published with his colleague, Onishi et al. who managed to get a part load stable two-stroke combustion process for lean mixtures in which ignition occurs without spark assistance [8]. Remarkable improvements in stability, fuel efficiency, exhaust emissions, noise, and vibration were reported. Onishi and his colleagues called this new combustion process “ATAC” (Active Thermo-Atmosphere

FIGURE 1: History of production and most advanced prototype CAI two-stroke engines [1].

Combustion). The first electric generator using an ATAC two-stroke engine was then commercialized in Japan from this period during a few years as shown in Figure 1.

Another paper concerning two-stroke autoignition was published in 1979 [9]. Noguchi and his colleagues named this autoignition combustion the TS (Toyota-Soken) combustion process. They also concluded that TS combustion occurred similarly without flame front while showing great efficiency and low emissions. They were one of the first to suggest that active radicals in residual gases could play an important role in the autoignition process.

In the late 1980s, Duret tried to apply Onishi’s pioneering work to DI two-stroke engines for improvement of part load emissions. For this purpose, he investigated the idea of using a butterfly exhaust throttling valve as previously shown by Tsuchiya et al. in a carburetted engine [10]. The first application of ATAC autoignition with direct fuel injection engine was then described in 1990 [11]. CFD calculations showed that mixing between the residual gas and fresh intake air may be reduced by precisely regulating the introduction of the intake flow through the use of an exhaust control valve [1].

This research work was further developed until the mid-1990s and the interest of using transfer port throttling (the

transfer duct in a two-stroke engine is the duct in which the fresh charge is transferred from the pump crankcase to the combustion chamber through a port on the wall of the cylinder) to even better control the degree of mixing between the fresh charge and the hot and reactive residual gas was demonstrated [1].

As shown in Figure 1, the first automotive two-stroke direct injection engine prototype using the transfer port throttling technique (a transfer duct for better controlling the degree of mixing between the fresh charge and the hot and reactive residual gas) for running in controlled autoignition (CAI) was presented by Duret and Venturi in 1996 [12]. Considering the benefits of combining direct injection with CAI, this engine was easily able to meet the European emissions standards valid up to the year 2000 without NO_x after treatment and with more than 20% fuel economy improvement compared to its four-stroke counterpart of equivalent power output [1].

In this period the possibility of using the autoignition in two-stroke motorcycle engines was investigated by Ishibashi. He showed that by using a charge control exhaust valve it was possible to control the amount of active residual gases in the combustion chamber as well as in cylinder pressure before compression [13]. He called this combustion process “Activated Radicals combustion (AR combustion).” Honda EXP-2 400 cc AR prototype was prepared for the 1995 Grenada-Dakar rally and performed very well compared to the four-stroke motorcycles, thanks in particular to their high fuel economy. This work was further developed [14, 15] up to the first industrial application of AR combustion in production in a Japanese motorcycle model in 1996 and in a European scooter model in 1998 (Figure 1) [1].

Recently, in 2008, Ricardo has developed a new prototype engine called 2/4 SIGHT which uses HCCI concept. This gasoline engine concept uses novel combustion, boosting, control, and valve actuation technologies to enable automatic and seamless switching between two- and four-stroke operations, with the aim of delivering significant performance and fuel economy improvements through aggressive downsizing. An engine equipped with this new system is capable of running on either the 2- or 4-stroke engine cycle, allowing their V6 test-bed to be downsized from 3.5 liters to 2.0 liters while making the same power output. This downsizing leads to a 27% reduction in fuel consumption and correspondingly lowered emissions. This engine is shown in Figure 2.

A further recent HCCI engine was reported by Lotus in 2008 [16]. As shown in Figure 3, a single-cylinder research engine called OMNIVORE has been built, employing loop scavenging and direct injection with the ability to vary geometrically the compression ratio from 8:1 to 40:1 or from 6.4:1 to 24.4:1 on a trapped basis (after exhaust port closure).

Blundell et al. and Turner et al. have published this engine data showing very low NO_x emission levels and a minimum part-load indicated a specific fuel consumption of 218 g/kW h using gasoline and 217 g/kW h using E85 [17, 18]. The engine was designed to be able to operate in HCCI modes and is intended to explore CO_2 reduction and the ability to operate on alternative alcohol-based fuels and gasoline, allowing flexible fuel vehicle operation.

FIGURE 2: Ricardo 2/4 sight engine [16].

FIGURE 3: Lotus OMNIVORE two-stroke engine [16].

2.2. Four-Stroke HCCI Engine. Based on the previous work in two-stroke engines [8], in 1983 Najt and Foster extended the work to four-stroke engines and attempted to gain additional understanding of the underlying physics of HCCI combustion [19]. They are the first to apply HCCI combustion concept in a four-stroke gasoline engine. In this work they considered that HCCI is controlled by chemical kinetics, with negligible influence of turbulence and mixing. They conducted experiments using PRF fuels and intake preheating. By means of heat release analysis and cycle simulation, they pointed out that HCCI combustion process was governed by low temperature (smaller than 950°K) hydrocarbon oxidation kinetics. Also they concluded that HCCI combustion is a chemical kinetic combustion process controlled by the temperature, pressure, and composition of the in-cylinder charge.

In 1989, Thring further extended the work of Najt and Foster in four-stroke engines by examining the performance of an HCCI engine operated with a full-blended gasoline [20]. The operating regime of a single-cylinder engine was mapped out as a function of air fuel equivalence ratio, EGR rate, and compression ratio.

Studies on four-stroke engines have shown that it is possible to achieve high efficiencies and low NO_x emissions

by using a high compression ratio and lean mixtures [21]. In the four-stroke case, a number of experiments have been performed where the HCCI combustion in itself is studied. This has mostly been done with single cylinder engines, which normally do not provide brake values. However, Stockinger demonstrated brake efficiency of 35% on a 4-cylinder 1.6 liter engine at 5 bar Brake Mean Effective Pressure (BMEP) [22]. Later studies have shown brake thermal efficiencies above 40% at 6 bar BMEP [23].

3. HCCI/CAI Challenges and Proposed Solutions

Although advantageous over traditional engines in thermal efficiency and NO_x emission, HCCI combustion has several main difficulties. These difficulties include “control of combustion timing,” “limited power output,” “homogenous mixture preparation,” “high unburned Hydrocarbon (HC) and carbon monoxide (CO) emissions,” and “weak cold-start capability” [4].

HC and CO emissions of HCCI engine are relatively higher in comparison with those of diesel engines [24]. Some potential exists to mitigate these emissions at high load by using direct in-cylinder fuel injection to achieve appropriate partial-charge stratification. However, in most cases, controlling HC and CO emissions from HCCI engines will require exhaust emission control devices where fuel optimization was not used. Catalyst technology for HC and CO removal is well understood and has been standard equipment on automobiles for many years. However, the cooler exhaust temperatures of HCCI engines may increase catalyst light-off time and decrease average effectiveness. As a result, meeting future emission standards for HC and CO will likely require further development of oxidation catalysts for low-temperature exhaust streams. However, HC and CO emission control devices are simpler, more durable, and less dependent on scarce, expensive precious metals than are NO_x and PM emission control devices [25]. Thus, simultaneous chemical oxidation of HC and CO in an HCCI engine is much easier than simultaneous chemical reduction of NO_x and oxidation of PM in a Compression-Ignition Direct-Injection (CID) engine.

At cold start, the compressed-gas temperature in an HCCI engine will be reduced because the charge receives no preheating from intake manifold and the compressed charge is rapidly cooled by heat transferred to the cold combustion chamber walls. Without some compensating mechanism, the low compressed-charge temperatures could prevent an HCCI engine from firing. Various mechanisms for cold-starting in HCCI mode have been proposed, such as using glow plugs, using a different fuel or fuel additive, and increasing the compression ratio using variable compression ratio (VCR) or variable valve timing (VVT). Perhaps the practical approach would be to use Spark Assisted Compression Ignition (SACI) approach as a bridge to the gap between HCCI and SI engines [26]. For engines equipped with VVT, it may be possible to make this warm-up period as short as a few fired cycles, since high levels of hot residual gases could be retained from

previous spark ignited cycles to induce HCCI combustion. Although solutions appear feasible, significant research and developing will be required to advance these concepts and prepare them for production engines [27].

Table 1 lists three major HCCI challenges and solutions proposed to address specific problems. The problem of high HC and CO emissions in HCCI is also linked to control of combustion timing since HC and CO emissions highly depend on the location of ignition timing. Despite the plurality of different proposed solutions, each of the proposed solutions has its own drawbacks. Variable intake temperature, variable intake pressure, and variable coolant temperature have slow response time, while VCR and VVT are technically difficult to implement. Practicality and cost effectiveness are the main concerns with most of the proposed options such as water injection and modulating two or more fuels [4].

As mentioned (Table 1), the main problem of HCCI is control of HCCI combustion timing. To have more discussion, this problem and its proposed solutions are the subject of the next part of this study.

4. Control of HCCI Ignition Timing

Several strategies have been investigated, with various levels of success, for controlling HCCI combustion timing and extending the load range. Most of these strategies can be divided into the broad categories of mixture dilution, modifying fuel properties, fast thermal management, and in-cylinder direct fuel injection. Many studies investigating HCCI control employ more than one method due to the complicated and highly coupled nature of the HCCI combustion problem [71].

4.1. Mixture Dilution for HCCI Control. In order to achieve CAI/HCCI combustion, high intake charge temperatures and a significant amount of charge dilution must be present. In-cylinder gas temperature must be sufficiently high to initiate and sustain the chemical reactions leading to autoignition processes. Substantial charge dilution is necessary to control runaway rates of the heat releasing reactions. Both of these requirements can be realized by recycling the burnt gases within the cylinder.

One approach to HCCI combustion phasing control is to advance or retard combustion timing by diluting the cylinder mixture. Najt and Foster showed that HCCI combustion in a four-stroke engine could be controlled by introducing recirculated exhaust gas into the cylinder intake mixture [19]. Christensen and Johansson showed combustion timing to be slower with higher amounts of EGR [72].

The presence of the recycled gases has a number of effects on the CAI combustion and emission processes within the cylinder. Firstly, if hot burnt gases are mixed with cooler inlet mixture of fuel and air, the temperature of the intake charge increases owing to the heating effect of the hot burnt gases. This is often the case for CAI combustion with high octane fuels, such as gasoline and alcohols. Secondly, the introduction or retention of burnt gases in the cylinder replaces some of the inlet air and hence reduces the oxygen

TABLE 1: Main HCCI challenges and proposed solutions [4].

HCCI challenges	Proposed solutions
Control of combustion timing	(i) Changing temperature history of mixture:
	(a) VVT and residual/exhaust gas trapping
	(1) Exhaust gas trapping [28, 29]
	(2) Modulating intake and exhaust flows [30, 31]
	(3) Combination of both [32]
	(b) Variable compression ratio (VCR) [33–36]
	(c) Variable EGR [31, 37, 38]
	(d) In-cylinder injection timing [39–41]
	(e) Modulating intake temperature [42–44]
	(f) Water injection [45]
Limited power output	(g) Variable coolant temperature [46]
	(ii) Changing mixture reactivity:
	(a) modulating two or more fuels [21, 47–49]
	(b) fuel stratification [50–54]
	(c) fuel additives and reforming [55–57]
	(d) variable EGR [37, 38, 58]
	(i) Boosting intake air flow:
	(a) supercharging [35, 59–61]
	(b) turbocharging [61–63]
	(ii) Dual-mode engines (HCCI at low load):
Homogenous mixture preparation	(a) SI-HCCI [58, 64, 65]
	(b) diesel-HCCI [66, 67]
	(i) Fuel injection in a highly turbulent port flow for gaseous and highly volatile fuels [68, 69]
	(ii) Early in-cylinder injection with sophisticated fuel injectors for diesel fuels [60, 70]

concentration (specially with a large amount of EGR). The reduction of air/oxygen due to the presence of burnt gases is called the dilution effect. Thirdly, the total heat capacity of the in-cylinder charge will be higher with burnt gases, mainly owing to the higher specific heat capacity values of carbon dioxide (CO_2) and water vapor (H_2O). This rise in the heat capacity of the cylinder charge is responsible for the heat capacity effect of the burnt gases. Finally, combustion products present in the burnt gases can participate in the chemical reactions leading to autoignition and subsequent combustion. This potential effect is classified as the chemical effect [1].

EGR or recycling of burned gases is the most effective way to moderate the pressure rise rate and expand the HCCI operation to higher load regions. The studies done related to EGR include both external EGR and internal EGR (residual combustion products) to achieve proper combustion phasing. External EGR is the more commonly utilized method for recycling exhaust gases. However, external EGR control has issues, such as, slow response time and difficulties in handling transient operating conditions [73]. A second way of reintroducing exhaust gases is through internal exhaust gas recirculation where the amount of exhaust gas residual in the cylinder is varied by changing the timing of the intake and exhaust valve's opening and closing events.

4.1.1. External Exhaust Gas Recirculation. External exhaust gas recirculation has been investigated by many researchers in the last decades. The study done by Thring investigated the effects of EGR rate (between 13 and 33%) on the achievable HCCI operating range and engine-out emissions [20]. Their

study found out that the maximum load of HCCI operating range for a four-stroke engine was less than that of a two-stroke engine under the selected conditions.

Christensen and Johansson observed that the upper load limit of a supercharged HCCI engine could be increased to an IMEP of 16 bars through the addition of approximately 50% EGR to the intake mixture, which retarded combustion and avoided knock [74]. In this study high EGR rates were used in order to reduce the combustion rate. While external EGR is promising for load range and combustion phasing improvement, some drawbacks still exist. For recirculation of the exhaust gas into the intake mixture, the exhaust manifold pressure has to be increased to a level over that of the intake manifold pressure. This pressure increase is often achieved by throttling the exhaust manifold, which can result in higher pumping losses and thus an overall lower net efficiency of the engine. Efficiency losses are also seen as a result of cooling the exhaust gases before reinduction to prevent early autoignition [74].

In 2001, Morimoto et al. found similar results using a Natural Gas fueled engine [75]. In this study external cooled EGR was used to control combustion phasing and extend the load range of an HCCI engine. He also concluded that the total hydrocarbon emissions were reduced at higher loads with the introduction of EGR.

Numerical studies conducted by Narayanaswamy and Rutland, using a multizone model coupled with GT-Power, confirmed that the effects of EGR (external) on diesel HCCI operation vary with different levels of EGR [76]. Interestingly they pointed out that ignition was advanced initially for low EGR cases and then began to retard with increase in EGR

percentage. The effect of cold EGR on the start of combustion was explained by competing effects, with the increase of the equivalence ratio advancing the ignition timing and the diluting effects retarding the combustion. As the EGR increases, the advancing effect prevails at first, and then evidently the retarding effect becomes dominant for further increase in EGR.

Atkins and Koch also observed that diluting the intake mixture using EGR is effective in retarding SOC timing. Similarly the introduction of EGR (around 62%) resulted in increasing maximum gross efficiency to 51%, much higher than that which could be achieved in an SI engine [77].

In 2011, Fathi et al. investigated the influence of external EGR on combustion and emissions of HCCI engine [38]. In his study, a Waukesha Cooperative Fuel Research (CFR) single cylinder research engine was used to be operated in HCCI combustion mode fueled by natural gas and n-Heptane. The main goal of the experiments was to investigate the possibility of controlling combustion phasing and combustion duration using various Exhaust Gas Recirculation (EGR) fractions. The influence of EGR on emissions was discussed. Results indicated that applying EGR reduces mean charge temperature and has profound effect on combustion phasing, leading to a retarded Start of Combustion (SOC) and prolonged burn duration. Heat transfer rate decreases with EGR addition. Under examined condition EGR addition improved fuel economy, reduced NO_x emissions, and increase HC and CO emissions.

4.1.2. Internal Exhaust Gas Recirculation. Internal exhaust gas recirculation is another promising method for achieving stable HCCI combustion. By changing the valve timing of the engine the amount of trapped residual gases (TRG) in the cylinder can be changed, thereby changing the temperature, pressure, and composition of the cylinder mixture at IVC. In 2001, Law et al. found that it was possible to change the amount of internal EGR by varying valve timing, which in turn allows for control of combustion phasing of HCCI combustion [28].

A systematic study on the effects of internal EGR was carried by Zhao et al. in a four-stroke gasoline HCCI engine via analytical and experimental approaches [78]. He revealed that the charge heating effect of the hot recycled gases was mainly responsible for advancing the autoignition timing and reducing the combustion duration. The dilution effect extended the combustion duration but had no effect on the ignition timing. The total heat capacity of the in-cylinder charge with EGR (internal) was found to rise due to the presence of species with higher specific heat capacity, such as CO_2 and H_2O . This effect reduced the heat release rate, thereby increasing the combustion duration. Furthermore, the EGR chemical effect was shown to have no influence on the autoignition timing and heat release rate but slightly reduced the combustion duration at high concentration of burned gases.

Milovanovic et al. studied the influence of a fully variable valve timing (VVT) strategy on the control of a gasoline HCCI engine and found that EVC and IVO timing have the

greatest impact on the ability to control HCCI combustion timing [79]. EVO and IVC timing were found to have little effect on HCCI combustion phasing control. A different research on fully VVT control of HCCI combustion was seen in the research of Urata et al. where a combination of direct injection, fully VVT with an electromagnetic valve train, and intake boost was used to control HCCI [80]. He hypothesized that injecting a small amount of fuel during negative valve overlap would allow unburned hydrocarbons in the internal residual to react, which could facilitate compression ignition during the following cycle.

In 2004, Yap et al. showed that while using internal EGR is promising for extending the load range and achieving the benefits of low NO_x operation in gasoline engines, the same cannot be said for natural gas (NG) HCCI engines [81]. It was found that due to the energy requirements for NG autoignition, intake heating and high compression ratios are required to achieve autoignition in the NG HCCI engine. Internal EGR has the potential to reduce the intake heating requirement for NG combustion, but because of the high compression ratios necessary to achieve autoignition the amount of internal EGR available for mixture dilution was significantly reduced. In addition high combustion temperatures from NG HCCI combustion can lead to significantly higher NO_x emissions when compared to a gasoline HCCI engine [81].

Cairns and Blaxill combined the concepts of internal and external EGR to extend the load range of a multicylinder gasoline HCCI engine while avoiding knock [82]. It was also found that this combined EGR scheme could be used to facilitate a smooth transition between controlled autoignition (or HCCI) and SI modes, utilizing a hybrid combustion technique expanding the engine's operating range. Kawasaki et al. addressed some of these problems by experimenting with the opening of the intake valve (a small amount during the exhaust stroke). This "pilot opening" allows for exhaust gases to be pulled into the intake manifold, thus heating the intake mixture and increasing the total amount of internal EGR [83].

4.2. Changing Fuel Properties for HCCI Control. Changing fuel properties of the cylinder mixture is a method that can be used for HCCI control. The required time and conditions needed for autoignition vary between fuels, so combustion timing can be controlled and the operating range can be expanded by varying the fuel properties in an HCCI engine [71].

4.2.1. Modulating Two or More Fuels. Dual fuel usage is a method that can be used to actively vary the fuel octane number by mixing a fuel with a high octane number and a fuel with a low octane number to create a fuel mixture with an intermediate octane number. Furutani et al. were one of the first that combined two different fuels to control the autoignition timing [47]. A low octane fuel (n-heptane) was injected into a high octane homogeneous air/fuel mixture (propane or hydrogen) just before the intake valve. They found that more torque can be obtained by using fuels with more octane number differences. However, some amount of

high-octane fuel does not participate in oxidation reactions because of its poor self-ignition tendency, so hydrocarbon emissions increase.

Stanglmaier et al. found that HCCI combustion timing could be controlled by mixing Fischer Tropsch (FT) Naptha with NG in an NG HCCI engine, allowing for optimization of efficiency and NO_x emissions at part loads [84]. Shibata et al. conducted a study on the effects of fuel properties on HCCI engine performance [85]. In this study fuels with different octane numbers were used in a four-cylinder engine. The resulting values of low temperature heat release (LTHR) and high temperature heat release (HTHR) varied with fuel composition. The low temperature chemical kinetics during LTHR as well as the negative temperature coefficient regime between LTHR and HTHR has been observed to have a large impact on HCCI combustion [84].

In an expanded study of HCCI control in 2007, Wilhelmsen et al. used dual fuels, NG and n-heptane, and a variable geometry turbocharger to develop an operational scheme in a NG engine by adding the lowest possible boost pressure to reduce pumping losses and minimize NO_x emissions [86].

The effects of different primary reference fuel blends on HCCI operating range, start of combustion, burn duration, IMEP, indicated specific emissions, and indicated specific fuel consumption were investigated by Atkins and Koch who found that by changing the fuel octane number the HCCI operating range could be expanded [77]. Recently an experimental and numerical study has been performed by Dumitrescu et al. to determine the influence of isooctane addition on the combustion and emission characteristics of a HCCI engine fueled with n-heptane [49]. Results show that for the operating conditions studied (CR from 10 to 16, engine speed of 900 rpm, AFR 50, 30°C intake temperature, and no EGR), isooctane addition retarded the combustion phasing and reduced combustion efficiency. As shown in Figure 4, when compression ratio increased from 10 to 15, CA50 advanced 14 deg CA for PRF0, while CA50 advanced 17 deg CA for PRF50 when CR increased from 11.5 to 16. This suggests that a blend with more isooctane is more sensitive to compression ratio. Also the operating compression ratio range narrowed with increasing isooctane fraction in the fuel. The NO_x emissions at advanced CA50 increased with increasing isooctane fraction, but the difference became negligible once CA50 approached TDC and beyond.

In 2004, Strandh et al. designed a PID controller and a model based linear quadratic Gaussian controller to establish cycle-by-cycle ignition timing control of an engine using blends of ethanol and n-heptane [87]. Dec and Berntsson separately found that a large amount of fuel stratification can lead to retarded ignition timing, which provides an additional actuator for control; however, too much stratification can ultimately lead to unstable combustion [88, 89].

4.2.2. Fuel Additives and Reforming. A potential technique for controlling the combustion timing of an HCCI engine is to change the fuel chemistry using two or more fuels with different autoignition attributes. Although a dual-fuel

FIGURE 4: Combustion phasing (CA50) versus compression ratio for four PRF blends [49].

engine concept is technically achievable with current engine technologies, this is not ordinarily seen as a practical solution due to the indispensability of supplying and storing two fuels. Reformer gas (RG) is a combination of light gases dominated by hydrogen and carbon monoxide that can be produced from any hydrocarbon fuel using an onboard fuel processor. Reformer gas has the wide flammability limits and high resistance to autoignition [57].

Significant research exists on the addition of reformer gas to fuels of various compositions to control HCCI combustion, which is interesting because of the ability to produce reformer gas from other fuels, effectively eliminating the need for two separate fuel sources. As shown in Figure 5, the experimental study of Hosseini and Checkel demonstrates that increasing the reformer gas fraction retards the combustion timing to

a more optimized value causing indicated power and fuel conversion efficiency to increase. Reformer gas reduces the first stage of heat release, extends the negative temperature coefficient delay period, and retards the main stage of combustion. In their study, two extreme cases of RG composition with H_2/CO ratios of 3/1 and 1/1 were investigated. The results demonstrate that both RG compositions retard the combustion phasing, but that the higher hydrogen fraction RG is more effective. Experimental work in this area has been completed by Hosseini and Checkel [90–93] and numerical works by Kongsereparp and Checkel [94, 95].

4.3. Fast Thermal Management for HCCI Control. Fast Thermal Management (FTM) is a controlling technique that involves rapidly changing the temperature of intake charge to control the combustion phasing. Many studies have indicated that HCCI combustion timing is sensitive to intake air temperature [19, 42, 44, 96, 97]. Haraldsson et al. and Yang et al. suggested the use of two air streams and regaining heat from exhaust gases to heat one of the air streams [43, 98]. By mixing two air streams, one direct from atmosphere and the other heated by exhaust gases, it is possible to control the temperature of the final intake air stream (each stream with independent throttles for mixing). Both studies observed the ability of the FTM system to control the combustion phasing of HCCI combustion. The study by Yang indicates that while FTM is effective to control combustion phasing in HCCI engines, the “thermal inertia” of the system makes cycle by cycle temperature adjustment difficult, which in turn complicates the control of HCCI combustion during transients [98]. This lag in achieving the desired HCCI combustion phasing was also observed by Haraldsson research, although in that study FTM was presented as an acceptable alternative to use variable compression ratio in closed loop control of HCCI combustion [43].

4.3.1. Intake Temperature. The effects of intake charge temperature on HCCI combustion on-set have been widely reported by many researchers. In 1983, Najt and Foster showed that HCCI of lean mixtures could be achieved in a SI engine that has a low compression ratio with elevated intake charge temperatures (300–500°C) [19]. In general, the intake charge temperature has a strong influence on the HCCI combustion timing. Figure 6 demonstrates the combustion chamber pressure versus the crank angle for a 2-stroke engine at the speed of 6000 rpm [99]. As shown in this figure, increasing the overall gas temperature significantly advances the HCCI combustion timing. In temperature of 575 [°K], the ignition is so advanced and the combustion is not so efficient but by decreasing the temperature the ignition would be retarded. Also by decreasing the intake temperature the maximum pressure of cylinder decreases but at the intake temperature of 525 [°K] the ignition timing would be so retarded that causes some misfiring.

Figure 7 shows the NO_x , CO, and HC emissions for various intake temperatures in the same engine. By decreasing the temperature and retarding the ignition timing, the NO_x emission has decreased, but CO and HC emissions have

increased. These adverse trends of CO and NO_x emissions are one of the main difficulties for controlling the emissions since by reducing one of them, another one increases. Also as demonstrated in this figure, the trend of emissions at intake temperature of 525 [°K] has changed and NO_x emission has suddenly increased because of some misfiring occurring in this point that was mentioned before.

The study performed by Iida and Igarashi also indicated that an increase in intake charge temperature (from 297°K to 355°K) increased the peak temperature after compression and advanced the HCCI combustion on-set [96]. Furthermore, the authors found that the effect of intake charge temperature on combustion on-set was greater for higher engine speed (1200 RPM) compared to the lower engine speed (600 RPM). Aceves and his coworkers carried out some investigations including analysis as well as experimental work [42]. On analysis, they developed two powerful tools: a single zone model and a multizone model. On experimental work, they did a thorough evaluation of operating conditions in a 4-cylinder Volkswagen TDI engine. The engine had been operated over a wide range of conditions by adjusting the intake temperature and the fuel flow rate. They found out that it may be possible to improve combustion efficiency by going to a lower fuel flow rate and a higher intake temperature. For the high load operating points, the trend was that lower intake temperature results in higher BMEP.

The effect of intake temperature on HCCI operation using negative valve overlap was investigated by Persson et al. [97]. They tested several points in the range between 15°C and 50°C to investigate the effects of intake charge temperature on spark assisted and unassisted HCCI combustion stabilities (COV_{IMEP} and $COV_{p_{max}}$) for a particular load and negative valve overlap condition. The study indicated that either increase in the residuals or intake charge temperature resulted in low coefficient of variation (COV) and stabilized the combustion. Recently, Maurya and Agarwal experimentally investigated the effect of intake air temperature on cycle-to-cycle variations of HCCI combustion and performance parameters [44]. The cycle-to-cycle variations in combustion and performance parameters of HCCI combustion were investigated on a modified two cylinder direct injection diesel engine. The inlet air was supplied at 120, 140, and 160°C temperature. It was found that at lower intake air temperature it is possible to ignite the richer mixture (up to $\lambda = 2$) in HCCI combustion mode. As intake air temperature increases, engine running on richer mixture tends to knock with very high rate of pressure rise. But at higher intake air temperature it is possible to ignite the leaner mixture (up to $\lambda = 5.5$) in HCCI combustion mode.

4.3.2. Compression Ratio. Compression ratio as an effective means to achieve HCCI combustion control has been carefully investigated by Christensen et al. for several years [33, 69]. His studies demonstrated that regardless of fuel type used increasing the compression ratio (9.6:1–22.5:1) had a strong influence on ignition timing and assists in decreasing the necessary intake charge temperature. Hiraya et al. also reported the effect of compression ratio (12:1–18.6:1) on

FIGURE 5: Effect of reformer gas on (a) net rate of heat release and (b) gross cumulative heat release [57].

FIGURE 6: Cylinder pressure versus crank angle for various intake temperatures [99].

FIGURE 7: CO, NO_x, and HC emissions for various intake temperatures [99].

combustion on-set [100]. Their study on a gasoline HCCI engine showed that higher compression ratios allowed for lower intake charge temperature and higher intake density for higher output. Furthermore, higher compression ratio contributed to higher thermal efficiency. The study done by Iida also has confirmed that change in compression ratio has a strong influence on HCCI combustion on-set [96]. Their results also showed that compression ratio has a greater effect on HCCI combustion on-set compared to changes in either intake charge temperature or coolant temperature.

The study done by Olsson et al. investigated the influence of compression ratio on a natural gas fuelled HCCI engine [34]. The experimental engine had a secondary piston that

was installed in the cylinder head whose position can be varied to attain variable compression ratio (VCR). In their tests, the compression ratio was modified (21:1, 20:1, 17:1, and 15:1) according to the operating condition to attain autoignition of the charge close to TDC. This VCR engine showed the potential to achieve satisfactory operation in HCCI mode over a wide range of operating conditions by using the optimal compression ratio for a particular operating condition. The study also showed that the maximum pressure rise rate increased with higher compression ratio for early combustion timing and a reverse effect was seen with delayed combustion on-set.

Haraldsson et al. investigated HCCI combustion phasing with closed-loop combustion control using variable compression ratio in a multicylinder engine [36]. In his study, closed-loop combustion control using accurate and fast variable compression ratio was run with acceptable performance. Time constant of three engine cycles was achieved for the compression ratio control. The closed-loop combustion control system of cascade coupled compression ratio and CA50 controllers had a time constant of 14 engine cycles or 0.84 s at 2000 rpm with a $dCA50/dt$ of 6.0 CAD/s.

4.4. Direct Injection for HCCI Control. Fuel injection into the cylinder at different stages of the engine cycle allows HCCI combustion timing to be advanced by improving mixture ignitability or retarded by increasing fuel stratification, creating the possibility of expanding the low and high load operating limits. Direct injection can be a good way to control HCCI combustion, but it depends heavily on the type of fuel and the timing of the direct injection [71].

A numerical study by Gong et al. showed that power density of an HCCI engine could be improved by the injection of a small amount of diesel fuel during the compression stroke of the engine. This pilot fuel injection also decreased the sensitivity of the HCCI combustion to intake conditions [101].

In 2003, Wagner et al. demonstrated that it would not be possible to use n-heptane as a port injection fuel for HCCI and instead a carefully timed n-heptane direct cylinder injection is used to avoid wall impingement and utilize the benefits of HCCI combustion [102]. In that year, Urushihara et al. found that a small injection of fuel during the NVO interval and a second injection during the intake stroke result in internal fuel reformation, which improves the ignitability of the cylinder mixture [103].

Dec and Sjöberg found that direct injection of fuel early in the intake stroke produced near identical results to a premixed charge. However, injection close to TDC improved the combustion efficiency of very low fuel load mixtures [104]. Numerical models by Strålin et al. showed that fuel stratification caused by injection of fuel around TDC results in pockets of rich fuel and air mixture, which promotes ignitability. Overall fuel stratification extended the combustion duration helping to avoid knock, thus extending the operating range of the engine [105]. Helmantel and Denbratt used multiple injection scheme of n-heptane to allow for sufficient mixing to operate a conventional diesel common passenger rail car engine with HCCI combustion [106].

In agreement with the recent study of Lu et al., for stratified charge compression ignition (SCCI) combustion with Port Fuel Injection of the two-stage reaction fuel combined with in-cylinder direct injection, the heat release rate demonstrates a three-stage heat release, as shown in Figure 8 [53]. The combustion phasing and the peak value of first-stage combustion play a vital role in the ignition timing and the peak point of the second-stage combustion, while the crucial factors of the first-stage reaction are the chemical properties of the premixed fuel. The second-stage ignition timing and

FIGURE 8: Three-stage heat release and its influential factors of SCCI combustion [53].

peak point have an important influence on the combustion phasing of the third-stage combustion, the thermal efficiency, the maximum gas temperature, and the knock intensity or the pressure rise rate. The dominant factors of the second-stage reaction are the premixed ratio and the physical properties of the premixed fuel. The third-stage combustion controls the engine thermal efficiency, the overall combustion efficiency, and NO_x and other emissions. Its decisive factor is the in-cylinder injection timing. If the ignition timing and peak value of each stage reaction can be flexibly dominated using mixture concentration stratification, composition stratification, and temperature stratification, then the expanded engine load, optimized thermal efficiency, and lowest NO_x emissions may be achieved [53].

Recently Yang et al. did an experimental study of fuel stratification for HCCI high load extension [51]. The investigation was performed in a single-cylinder four-stroke engine equipped with a dual fuel injection system, a port injector for preparing a homogeneous charge with gasoline and a direct in-cylinder injector for creating the desired fuel stratification with gasoline or methanol. Both the effect of gasoline fuel stratification and gasoline/methanol stratification were parametrically investigated. Test results indicated that weak gasoline stratification leads to an advanced combustion phase and an increase in NO_x emission, while increasing the stratification with a higher quantity of gasoline direct injection results in a significant deterioration in both the combustion efficiency and the CO emission. Engine tests using methanol for the stratification retarded the ignition timing and prolonged the combustion duration, resulting in a substantial reduction in the maximum rate of pressure rise and the maximum cylinder pressure a prerequisite for HCCI high load extension. About the stratified methanol-to-gasoline compared to gasoline HCCI, a 50% increase in the maximum IMEP attained was achieved with an acceptable maximum pressure rise rate of 0.5 MPa/°CA while maintaining a high thermal efficiency [51].

5. Conclusion

CAI/HCCI engines still have not met the level of development and cost that would make a market introduction possible at the moment. The technical challenges facing both gasoline and diesel HCCI combustion are their limited operational range and less optimized combustion phasing, owing to the lack of direct control over the start of ignition and the rate of heat release. HCCI combustion represents a step change in combustion technology and its future research and application should be considered as part of an effort to achieve low-temperature combustion in a wide range of operating conditions in an IC engine. Combustion process in future IC engines converges towards premixed compression ignition combustion, while turbocharging and direct injection become a norm on such engines: it therefore may not remain futuristic but become a realistic possibility that, with more flexible engine hardware and their real-time control, a fully flexible engine could be developed to convert the chemical energy from any type of fuel into mechanical work through premixed auto-ignited low-temperature combustion [1].

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