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Challenging and Future of Homogeneous Charge Compression Ignition Engines; an Advanced and Novel Concepts Review

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Research Article

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Abstract

A thorough examination of past and present combustion chemistry results, emission behaviors, the difficulties this technique faces, and all controlling parameters, including transient states, are presented, along with the potential of HCCI combustion to lower internal combustion engine exhaust emissions, especially NOX and soot emissions, and to define the application range of this technique. According to HCCI combustion chemistry and emissions analysis, the heavy fuels exhibit two-stage heat release or two-stage combustion process involving low temperature oxidation (LTO) stage followed by high temperature oxidation (HTO) stage separated by a time delay between them. This is attributed to the negative temperature coefficient (NTC), the HCCI engine's advantage of reducing NOX emissions diminishes at high load conditions, HC production decreases with increasing engine load, and soot ejection is negligible under all operating conditions. Over a broad range of speed and load, the HCCI combustion characteristics are advancedly controlled by valve timing, compression ratio, intake air temperature, and EGR. The use of EGR in HCCI operation is restricted to EGR-rates of around 70%. At this point, response speeds and ignition timing are significantly decreased and delayed, respectively, which causes misfiring and the generation of HC emissions. To fully realize the benefits of HCCI engines in the future, challenges include homogenizing fuel, air, and recycled burned gases before ignition, controlling ignition and combustion timing, and controlling heat release rates.

Key words: HCCI Combustion, Emission, Lean Burn Fuel Mixture, EGR, Valve Timing, Compression Ratio

Introduction

Nowadays, HCCI combustion concepts are considered as the most important solution for both conventional gasoline and diesel engine processes. This approach theoretically combines the advantages of both conventional processes; low raw emissions and high fuel economy. It can be realized by either liquid or gaseous fuel or modifying the conventional gasoline or the diesel engine. In all cases, the combustion is initiated by auto-ignition of an overall lean and homogeneous fuel-air mixture. Because there is no difference in local and overall equivalence ratio and temperature values in case of perfect HCCI-process, the compression of completely homogeneous mixture leads to a simultaneous auto-ignition of the whole discharge. Compared to the SI engine, where only a small part of the total mixture is part of the flame front and takes part in the combustion process, the entire charge reacts at the same time in the case of HCCI combustion. However, the local temperatures are significantly lower than that of the flame front in the SI engine or that of the stoichiometric zones of the diesel spray combustion, and compared to the SI combustion, the local HCCI combustion releases the heat from the entire charge more slowly and uniformly. Because the whole charge is involved simultaneously in the combustion process, HCCI heat release has a smaller overall duration than the conventional ones.

Classification of HCCI Combustion

Active Thermo-Atmosphere Combustion ATAC (1), Compression-Ignited Homogeneous Charge CIHC (2), Premixed Lean Diesel Combustion PREDIC (3), Premixed Charge Compression Ignition PCCI (4), Activated Radical Combustion AR (5), Controlled Auto-Ignition CAI (6), Homogeneous charge Intelligent Multiple Injection Combustion System HIMICS (7), Uniform Bulky Combustion System UNIBUS (8), or Modulated Kinetics MK (9). However, Thring (10) created the globally recognized term, HCCI. In order to prevent misfire and stabilize the combustion process under partial load, the principles of HCCI combustion were applied to two-stroke engines (1), (11). Additionally, it has been shown that HCCI combustion is only feasible at moderate engine speeds and loads because of incomplete burning at very low loads or high engine speeds and uncontrolled and premature knock-like combustion at higher loads. Conventional diesel spray combustion or gasoline SI-combustion must be utilized for larger loads, particularly at full load. Only applications with very high or extremely low loads find this so-called dual-mode notion (12), (13) appealing.

Chemistry of HCCI Engines

The timing of auto-ignition, the rate of heat release, the reaction intermediates, and the end products of combustion are all determined by the HCCI oxidation chemistry. Heavy fuel combustion using HCCI exhibits a two-stage heat release or two-stage combustion process with a low temperature stage and a high temperature stage. Low temperature kinetic processes (Low Temperature Oxidation, or LTO) are linked to the first stage of heat release, whereas High Temperature Oxidation (HTO) is the second, much stronger stage (primary reaction). The Negative Temperature Coefficient regime (NTC) is responsible for the temporal lag between LTO and HTO (14), (15), (17). 7–10% of the energy is released during LTO (15), with the remaining energy being released during HTO.

The most significant chemical processes that occur throughout olefin's LTO, HTO, and NTC and are used in the chemical kinetic models are described here.

Olefin Low Temperature Oxidations

The LTO reactions generally occur at temperature not exceed 850 ^{O}K [(16)-(18)], further details about LTO are given in [(19)-(23)]. The most important steps during LTO are the following; the reaction (1) starts with the generation of hydrocarbon radicals (R) by hydrogen atom abstraction from the fuel (RH), and then hydrogen atom (H) reacts with oxygen (O_2) in an endothermic reaction.

$$RH + O_2 \rightarrow R + HO_2 \quad \text{.....(1)}$$

This reaction is quite slow, and the following two- step mechanism finally starts the low temperature oxidation.

Step 1: hydrocarbon radical (R) attacked by existent oxygen to form Alkyl-peroxide (RO_2) reaction (2).

$$R + O_2 \rightarrow RO_2 \quad \text{.....(2)}$$

Then, an H-atom abstraction occurs. This reaction would result in a subsequent chain branching and thus could lead to ignition [(24), (25)], reaction (3).

$$RO_2 + H \rightarrow ROOH \quad \text{----------(3)}$$

Step 2: A second O_2-addition, reaction (4), in addition with the subsequent H-atom abstraction finally results in ignition, reaction (5-8):

$$HO_2R + O_2 \rightarrow HO_2RO_2 \quad \text{.....(4)}$$
$$HO_2RO_2 + RH \rightarrow HO_2RO_2H + R \quad \text{.....(5)}$$
$$HO_2RO_2H \rightarrow HO_2RO + OH \quad \text{.....(6)}$$
$$HO_2RO \rightarrow ORO + OH \quad \text{.....(7)}$$
$$RH + OH \rightarrow R + H_2O \quad \text{.....(8)}$$

The (OH) radicals produced by these reactions oxidize the hydrocarbons and the increasing reaction rate of these exothermic reactions (reactions 2 to 8) result in the first heat release (LTO). With increasing temperature, the formation of oxidizer (OH) decreases, leading to the decrease of the first-stage heat release. This mechanism is called degenerate chain branching. It is responsible for the increase of ignition delay with rising temperature, Negative Temperature Coefficient (NTC), and explains the existence of the NTC region between LTO and HTO during the two-stage HCCI combustion.

Negative Temperature Coefficient Region

At temperatures between 800 K and 1000 K, approximately, the fuel radicals feed the following two reactions (9-10),

$$R + O_2 \rightarrow \text{Olefin} + HO_2 \quad \text{.....(9)}$$
$$HO_2 + HO_2 \rightarrow H_2O_2 + O_2 \quad \text{.....(10)}$$

and result in an accumulation of H2O2, which remains relatively inert as long as the temperature is below about approx 1000 K[26].

$$H_2O_2 + M \rightarrow OH + OH + M \quad \text{(11)}$$

While, the temperature increases due to compression, and above 900-1000 K the chain branching reaction (11) quickly produces large numbers of OH radicals and thus initiates the second-stage heat release process via reaction (8).

The temperature at which the reaction occurs is independent of the fuel. But, any engine parameter that makes the in-cylinder charge earlier reach the H2O2 decomposition temperature (e.g. higher compression ratio, high fuel equivalent ratio, and increased intake temperature) will advance the start of main combustion.

High Temperature Oxidation of Olefin

At temperatures above 1000 K, molecules with long chains break up into short one. First, alkyl radicals are formed by abstraction of H-atoms (reaction 12):

$$RH + X \rightarrow R + HX \quad \text{(12)}$$

X represents radicals like (H, O, OH, and HO_2) [27]. At temperature above 1100 K, The most important chain branching reaction (13) that is responsible for the production $H + O_2 \rightarrow O + OH$ **(13)**

The further decomposition results in mainly CH3 and C2H5 radicals, and the subsequent reactions lead to the formation of formaldehyde (CH2O) and to the burning of hydrocarbons (C2H6, C2H5, C2H4, C2H3, C2H2). Detailed reaction paths can be found in [28]-[30]. The HTO leads to the formation of CO, which finally oxidizes to CO_2. This final reaction releases a significant amount of energy and thus is important for high fuel economy. Reaction (14) is only important above 1100 K and below this temperature there is only a partial oxidation of CO. $CO + OH \rightarrow CO_2 + H$ **(14)**

This explains the increased of HCCI-process CO emissions, at which the combustion temperatures are much lower than the conventional diesel process.

Emissions and Efficiency of HCCI Engines

Because HCCI engines can run under ultra lean mixture conditions and low temperature combustion, one of the most significant advantages of HCCI combustion is the massive decrease of NOX raw emissions of 90–98% when compared to traditional combustion (6), (12), (13), (31), and (33). However, the advantage over conventional engines diminishes as the load increases and the peak combustion temperatures climb (32).

The homogenous lean mixture prevents the production of fuel-rich zones and temperatures exceeding 1400 K (34, 35). Fuel deposition on the walls, however, may lead to inadequate mixture preparation and local fuel-rich areas that can create soot due to incomplete combustion. Because fuel penetration is high and the cylinder's internal temperature and pressure are low during early direct injection, this may particularly occur. HCCI combustion often produces far more HC emissions than traditional combustion approaches (35), (36). For instance, compared to normal diesel combustion, Schlotz's HCCI diesel studies (35) revealed almost five times greater HC emissions. This results from either high EGR rates or low combustion temperatures brought on by lean mixes. To prevent excessive heat release rates, EGR is often required to extend the combustion time. Excessive EGR may cause misfiring, which would significantly raise HC emissions. However, EGR may also aid in lowering the quantity of unburned hydrocarbons since it will raise the charge temperature, particularly in the case of internal EGR, and lower the limit of lean mixes, which will naturally boost the combustion and emission characteristics.

Because both HC and CO contain chemical energy that hasn't been released during combustion, their emissions have a direct impact on fuel usage. Because of the high heat release rates and very brief combustion duration, the HCCI process may approach the ideal Otto cycle (combustion at constant volume) if a suitable phasing of combustion relative to the engine cycle is attained. This usually leads to excellent efficiency if there is no

misfiring or incomplete burning (37). The elimination of throttle losses gives the gasoline engine an extra efficiency boost. The spray-guided direct injection approach has the same theoretical ability to reduce fuel consumption at partial load (38), (39). Several writers have obtained HCCI fuel efficiencies that are equivalent to conventional diesel combustion at part load (40), (41). However, due to inadequate mixture preparation, fuel wall impingement, incomplete burning, and poor combustion phasing (premature ignition), this advantageous impact of HCCI combustion may not always be achieved when fuels with low volatility and strong ignitability, such as diesel, are employed.

Engines Parameters and Control of HCCI Combustion

Controlling the combustion process without having direct access to the reaction mechanism is the biggest issue when running an engine in HCCI auto-ignition mode. The thermodynamic circumstances and the chemical makeup of the cylinder charge induce chemical reactions. The most crucial engine factors that affect the beginning of combustion and the ensuing heat release have been thoroughly covered in the sections that follow.

Compression Ratio

The rate of heat release in HCCI combustion processes and its impact on the temperature change inside the combustion chamber are both influenced by the compression ratio. Both the end-of-compression temperature and the rate of low temperature reactions rise with an increase in the compression ratio. This facilitates the igniting process overall and enables

lower-temperature inlet charges to be effectively burned. On the other hand, compression ratios in HCCI engines often need to be lowered in order to postpone the onset of combustion and avoid excessive heat release rates. However, because a lower compression ratio results in lower thermal efficiency, the lowering of compression ratios should be limited. In a diesel-fueled HCCI engine, the impact of compression ratio on the start of combustion (SOC) was investigated (42). This experiment found that if the compression ratio was altered from 8:1 to 12:1, SOC progressed by around 20 crank angle degrees. Additionally, knock-like combustion with high temperatures and elevated NOX emissions occurs when compression ratios are too high. A variable compression ratio via variable valve timing is an efficient but costly solution to this issue. This technique may be used to dual-mode engines to permit lower compression ratios in the event of HCCI combustion and to achieve the high compression ratios required for good thermal efficiency in the case of conventional combustion at full load.

Variable Valve Timing

Non-cooled internal EGR is often used to provide the necessary high temperatures and charge dilution for slowing the release of heat, which can only be effectively supplied by variable valve timing. A cycle-to-cycle and cylinder-specific control of charge composition is beneficial for completely transient operation. A completely variable electro-hydraulic valve actuation system makes this feasible (13), (39), (43). Additionally, these systems allow one to adjust the intake flow (swirl ratio) and the effective compression ratio, both of which are advantageous for traditional operation. For HCCI engines, the strategy of trapping internal residual gas to reduce the thermal requirements for the auto ignition timing across a broad range of speeds and loads is difficult (44)–(48).

Exhaust Gas Recirculation (EGR)

One well-known technique for lowering combustion temperatures and NOX emissions in conventional engines is exhaust gas recirculation. Recycled burned gases replenish a portion of the initial cylinder charge. The external EGR and the internal EGR are the two main EGR concepts in use today. When using an external EGR, some of the manifold's exhaust gases are split off, cooled if needed, and then combined with the fresh air in the suction section. Another way to lower combustion temperatures is by cooling. Whether some of the hot exhaust gases from the previous cycle remain in the cylinder or are drawn back from the manifold in the case of internal EGR depends on the timing of the intake and outlet valves (49)–(54). EGR has four significant impacts on HCCI combustion:

A. Heat capacity effect: The inert burned gases must be heated together with the remaining in-cylinder charge during compression and combustion. End of compression and combustion temperatures, heat release rates, and maximum pressure increase (dp/dt)max are all decreased because the total heat capacity of the charge is larger with burned gases owing to the higher specific heat capacity values of carbon dioxide (CO2) and water vapor (H2O) (51, 54). If a lot of EGR is applied, the heat capacity effect prolongs the combustion time.

B. Charge heating effect: The temperature of the intake charges rises when hot, burned gases are combined with

colder inlet air. Advanced auto-ignition timing is mostly caused by the heating effect, which is significant for non-cooled EGR systems. Additionally, it shortens the combustion duration and raises the maximum pressure rise (dp/dt)max and heat release rate (55, 56).

C-Dilution effect: A portion of the incoming air is replaced by burned gases, which lowers the oxygen content in the cylinder. In the case of Controlled Auto-Ignition CAI combustion, this impact has no influence on auto-ignition timing; nevertheless, if significant levels of EGR are utilized, it prolongs combustion and slows down the rate of heat release (18), (56). Tsurushima et al.'s (17) experiments using diesel-like fuels (two-stage ignition) likewise demonstrate a reduction in oxidation rates; however, the initiation of both LTO and HTO is postponed, and the time gap between both reactions is lengthened.

D-Chemical effect: Lastly, the chemical reactions of the next combustion cycle may include active combustion products found in the burned gases. Since external and cooled EGR often produces chemically inert gases, it has little impact. Internal EGR, however, may result in accelerated auto-ignition due to the presence of short-lived chemically active components in the recycled burned gases (5), (50), and (57).

Lastly, the HCCI combustion is constrained at constant EGR-rate by the knocking combustion at higher loads (richer mixture) and its low load limit (lean mixture). In knocking combustion, excessive heat is generated during rapid combustion, leading to high peak pressures and potentially damaging the engine. Reduced heat release rate, delayed combustion timing, and a drop in combustion temperature are all caused by an increase in EGR. However, EGR also lowers the low load limit in terms of equivalent ratio; a lower combustion temperature results from the presence of more recycled burned gases. These low temperatures may cause partial burning, which raises CO and HC emissions and decreases efficiency. At EGR-rates of around 70%, the HCCI operating area reaches its third limit. At this point, response rates and ignition timing are significantly decreased and delayed, respectively, which causes misfiring and increases HC emissions once again (33).

Inlet Air Temperature

The most often used technique in laboratory tests is adjusting the temperature of the intake air to regulate the ignition start. Combustion begins earlier at higher intake temperatures and vice versa. The findings of Velji et al. (38) demonstrate that for a mixture of 80% methane and 20% diesel at various air temperatures. In addition to a significant reduction in volumetric efficiency when intake air temperatures are high, the range of crank angle that may affect ignition timing is rather small. One of the many consequences of a change in the EGR rate in the case of non-cooled EGR is often a change in the intake temperature.

Air Excess Ratio

Richer mixes result in quicker energy release and advanced ignition, as previously discussed together with the effects of EGR. Depending of the fuel and other boundary conditions, combustion stability deteriorates, knock occurs, and NOX emissions rise at a particular point. Conversely, slower energy release and delayed ignition result from thinner mixes. At low temperatures, very lean mixes cause incomplete combustion, which increases CO and HC emissions..

Engine Speed

Turbulence and burn rates rise as engine speed increases in conventional engines. Since the mixture homogeneity is already achieved before combustion, there is no discernible impact of speed with HCCI systems. This explains why the time available for responses decreases as engine speed increases while reaction rates stay mostly constant.

Fuel Composition

Diesel-like fuels and gasoline-like fuels are the two primary categories of fuels in terms of the HCCI combustion process. fuels similar to diesel (increased
ignitability, high cetane number) exhibit two-stage combustion, while fuels with low cetane numbers, such as gasoline, exhibit single-stage combustion. In addition to the cylinder charge's chemical and thermal makeup, the fuel molecule size and structure also affect the LTO reaction occurrence time and subsequent heat release (18). The fuel composition affects the whole HCCI process since the LTO reaction has a significant impact on the beginning of major combustion. The fuel composition has no discernible effect on the ignition start in a single-stage ignition. The experimental studies of Tsurushima et al. (17), who employed a gas sampling technique to investigate the composition of the in-cylinder charge during LTO and HTO for three diesel-like straight-chain saturated hydrocarbons (n-pentane, n-hexane, and n-heptanes) with varying chain lengths, corroborated these findings. The mixture was completely homogenized in an external vaporizer, and all tests

maintained the same intake temperature (100°C), equivalency ratio, and O2 concentration. With LTO and HTO, all fuels exhibit the typical two-stage combustion. Fuel molecule size affects ignition and combustion behavior; longer carbon chains are more ignitable, the LTO starts sooner, and more energy is produced. Because of this, the combustion chamber's mean temperature rises to around 900 K, which is required to start the HTO.

Fuel-Air Mixture Homogeneity

To avoid soot, HC, and CO emissions during HCCI combustion, the degree of homogeneity of the charge before ignition should be as high as feasible. When it comes to nitric oxides, a significant amount of mixture non-homogeneity may be accepted without increasing the production of NOX (32). When it comes to the controlled auto-ignition CAI gasoline combustion (58), full homogeneity leads to an end-of-compression temperature that is lower than the necessary auto-ignition temperature of around 1000°C under realistic intake air temperature, EGR rate, and temperature.

To reach the necessary end-of-compression temperatures of 1000–1200°C, the temperature of the intake air must be raised to around 150°C. On the other hand, if hot EGR zones are next to fuel-rich reactive zones, partially homogenized mixtures are more likely to ignite earlier. In this instance, there is no heating of the intake air and the ignition takes place at generally lower but adequately high local temperatures. There is no traditional flame combustion since the whole charge is close to the ignition limit, even if ignition starts in a comparatively tiny region. The initial local ignition's energy release sets off many subsequent ignitions over the whole combustion region, producing a homogenous combustion whose energy release rates may be regulated by the degree of homogeneity.

HCCI Engines Transient Behaviors and Control Strategies

Boundary conditions make it more difficult to achieve steady-state conditions in HCCI combustion, which has a big impact on engine performance. As was discussed in the previous section, the variables used to control the HCCI combustion process interact significantly during engine operation. Conventional one input single output control methods cannot give sufficient control in transient situations (39). To control combustion from cycle to cycle and allow a transient change in speed and load in the HCCI working zone, a real-time combustion signal is needed in addition to conventional and relatively slow mass flow, air excess ratio, and temperature sensors.

Among the most challenging tasks are the time at load variations, especially in lean mixture scenarios, and the mode transition between HCCI and conventional combustion. A model-based combustion control system that precisely forecasts the charge composition and thermodynamic conditions is required in this situation as the relevant thermodynamic values fluctuate from cycle to cycle. For example, a dual-mode engine has to transition from SI to HCCI mode when the load is reduced. Due to the typically high exhaust gas temperatures, the first HCCI cycle will have a highly advanced combustion with an unfavorable maximum pressure rise. The sudden change from one mode to another must be smoothed out using model-based transition functionality.

HCCI Engine Challenges

Overcoming many obstacles is necessary to fully realize the advantages of HCCI combustion in modern engines. The two main challenges are

Fuel, air, and recycled burnt gases are homogenized before to ignition.

the timing of combustion and ignition as well as the control of heat release rates. The challenges associated with these issues are described below.

A- Homogeneous Mixture Formation

According to the HCCI engine theory, achieving good fuel efficiency, reducing PM and HC emissions, and preventing oil dilution all depend on understanding mixture generation and minimizing fuel-wall interactions. Fuel deposition on the walls has been shown to negatively affect HC emissions, even for fuels that are somewhat volatile, such as gasoline (59). Exterior mixture formation and interior mixture creation are the two main methods of producing mixtures and homogenizing fuel and air.

External mixture production is the simplest way to get a uniform in-cylinder mixture. The mixture enters the cylinder during the intake stroke. This method is also known as port injection or fumigation. The turbulence created by the intake flow supports further homogeneity. Since the air-fuel mixture is subjected to temperature and pressure development throughout, the port injection is included in the early homogenization concepts. One drawback of this method is that the ignition's beginning cannot be influenced by the injection time. Heavy fuels with low volatility result in poor evaporation, increased wall impingement, increased CO and HC emissions,

increased fuel consumption, and dilution. For very volatile gaseous and liquid fuels, this injection is especially desirable (12), (13).

When internal mixture formation takes place, fuel is injected directly into the cylinder. The two ideas that underpin this strategy are early and late injection. Early injection, which uses a long ignition delay at low temperatures to homogenize the fuel mixture, is the most widely used method for HCCI heavy fuel applications. Interestingly, part or all of the gasoline is injected prior to reaching top dead center. Because of the fuel's poor volatility and the cylinder's low air density, strong fuel injection may result in severe wall dampness. In these circumstances, it is necessary to design new, highly flexible injection systems that can respond to variations in the combustion chamber's shape as well as in-cylinder pressure and temperature during injection. Despite the problems with in-cylinder injection, direct injection is expected to become the most important method for HCCI engines in the future.

Numerous researchers have already worked hard to reduce wall impingement and create low penetration injectors (12), (13), (60), (61). An appropriate injection system must have a high degree of flexibility in order to allow the injection strategy to be modified to the changing boundary circumstances during injection. High-pressure injection with several small nozzle holes is often used to improve spray disintegration and include the whole cylinder charge in the mixture production process while limiting wall film development. It is possible to modify the spray penetration further by splitting the injection event into several pulses of different durations. In the shorter pulse duration of early in-cylinder injection in HCCI engines results in less momentum for the liquid fuel, which lowers penetration. The low gas density at the beginning of injection necessitates short pulses with lower injection velocities and a comparatively lengthy time gap between pulses. As the piston rises, penetration diminishes and the temperature and density of the cylinder increase. The pulse durations may be increased while the intervals between subsequent pulses are decreased. Towards the end of the pulsed injection, the distance between the nozzle and the piston drops significantly, and each pulse's mass must be reduced once again to prevent fuel deposition on the piston (62). Additionally, the spray direction has to be correctly adjusted since the volume between the nozzle and the piston is much larger than with conventional diesel injection at TDC. To provide adequate fuel and gas mixing and prevent fuel deposition on the cold cylinder wall, the angle between the spray plumes must be decreased. The effects of fuel droplet interaction and spray direction angle on the internal flow field of the cylinder have been extensively studied (63)–(65). A narrow spray cone angle injector with a dual injection technique was studied to reduce wall wetness and avoid an out-of-bowl injection when fuel is injected early for HCCI combustion (66). A traditional diesel engine's fuel injection angle was lowered from 156 degrees to 60 degrees. The dual injection method was very effective in reducing NOX emissions, according to the results of an investigation of three alternative injection strategies: early injection, traditional diesel combustion, and dual injection. Additionally, the dual injection approach with narrow angle fuel injection may reduce CO emissions by modifying the first and second injection timings.

B - Control of Ignition and Combustion Timing and Heat Release Rates

The management of ignition time, which establishes the primary combustion phasing and so has a significant impact on efficiency and operating range, is a significant difficulty in addition to the issue of homogenous mixture production. The engine may be destroyed by intense knock-like combustion caused by premature combustion. For lean fuel-air ratios, both with and without significant EGR, a stable combustion may be achieved under low and partial load. Heat release rates and knock-like combustion significantly increase as the load approaches stoichiometric levels. Moreover, the advantages of emissions disappear (32, 40). It is not feasible to directly adjust the ignition timing using the spark or the beginning of injection in conjunction with the very brief and well-known ignition delay, in contrast to spark ignition or typical diesel engines. The low temperature chemistry, which is dependent on the charge's whole temporal temperature history, has a major impact on the beginning of combustion. Ignition timing and phasing have an impact on the governing techniques of homogenous charge compression ignition combustion heat release rates. Exhaust gas recirculation (EGR) rate is a crucial parameter to regulate the HCCI combustion process, according to the findings of Mingfa Yao et al.'s (67) study on these tactics. With a high stated thermal efficiency and little emissions, the ignition time and combustion duration may be controlled within an appropriate range. The operating range of equivalency ratios and engine loads in HCCI combustion may be increased by the efficient ignition control provided by the addition of hydrogen to the fuel (68). This study used chemical kinetics analysis to examine the processes of ignition control. The findings indicate that hydrogen's consumption of OH during low-temperature oxidation is responsible for the delayed ignition. The initiation of the high-temperature

oxidation is delayed and heat release is slowed down as a result of the lower OH content. The auto-ignition characteristics of the fuel, the fuel-air ratio, the fuel's volatility, the EGR rate, the temperature and reactivity of the recycled gas, the mixture homogeneity, the engine's compression ratio, the intake temperature, the heat transfer to the engine, and fuel additives all have an impact on HCCI combustion phasing. There are two primary categories of methods for controlling combustion phasing (32). The first group's goal is to change the mixture's time-temperature history. Fuel injection timing, intake air temperature fluctuation, compression ratio (CR) variation, and variable valve timing are all included. using altering the fuel's characteristics, the fuel-air ratio, or the quantity of oxygen using EGR, the second group aims to regulate the charge's reactivity.

Final comment

The historical categorization of HCCI engines is reviewed and studied, along with all the metrics used to investigate their performance, emissions, and combustion characteristics. And the following is a summary of the analysis that has been done so far.

1. The Negative Temperature Coefficient regime (NTC) is responsible for the time lag between the low temperature oxidation (LTO) and high temperature oxidation (HTO) stages of HCCI combustion of heavy fuels, which exhibits a two-stage heat release or two-stage combustion process. The key chemical processes that occur during olefin fuel's LTO, HTO, and NTC—all of which are used in chemical kinetic models—are examined and clarified.
2. An more fuel-rich homogenous mixture causes the cylinder peak temperatures to rise, which eliminates the benefit of NOX reduction under high load circumstances. Furthermore, the contribution of HCCI engines to the creation of soot is minimal, particularly if there is no fuel deposition on the engine walls, which might happen as a result of improper mixture preparation. Additionally, HCCI engines have comparatively high HC generation, particularly when operating at partial load.
3. When using HCCI combustion, the compression ratio should be kept moderate because a high compression ratio will result in knock-like combustion; on the other hand, a low compression ratio is necessary to delay the start of combustion and prevent excessive heat release rates.
4. There are now two distinct EGR ideas in use: the internal EGR and the external EGR. EGR is used in HCCI engines to raise the intake charge temperature, which is primarily responsible for accelerated auto-ignition timing. It also enhances the rate of heat release, the maximum pressure rise (dp/dt) value, and the length of combustion. A higher EGR results in a lower oxygen to fuel ratio low load limit, which causes late combustion timing and a slower rate of heat release. In HCCI operation, when response rates and ignition timing are significantly lowered and delayed, respectively, the EGR rate is restricted to 70%. This causes misfiring and the generation of HC emissions. Additionally, since it affects the engine's volumetric efficiency, the use of different intake air temperatures in HCCI engines to regulate the ignition start is somewhat restricted.

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