






Review

Performance and Emission Parameters of Homogeneous Charge Compression Ignition (HCCI) Engine: A Review

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Abstract: Strict emission regulations and demand for better fuel economy are driving forces for finding advanced engines that will be able to replace the conventional internal combustion engines in the near future. Homogeneous charge compression ignition (HCCI) engines use a different combustion technique; there are no spark plugs or injectors to assist the combustion. Instead, when the mixtures reach chemical activation energy, combustion auto-ignites in multiple spots. The main objective of this review paper is to study the engine performance and emission characteristics of HCCI engines operating in various conditions. Additionally, the impact of different fuels and additives on HCCI engine performance is also evaluated. The study also introduces a potential guideline to improve engine performance and emission characteristics. Compared to conventional compression ignition and spark ignition combustion methods, the HCCI combustion mode is noticeably faster and also provides better thermal efficiency. Although a wide range of fuels including alternative and renewable fuels can be used in the HCCI mode, there are some limitation/challenges, such as combustion limited operating range, phase control, high level of noise, cold start, preparation of homogeneous charge, etc. In conclusion, the HCCI combustion mode can be achieved in existing spark ignition (SI) engines with minor adjustments, and it results in lower oxides of nitrogen (NO_x) and soot emissions, with practically a similar performance as that of SI combustion. Further improvements are required to permit extensive use of the HCCI mode in future.

Keywords: HCCI; engine performance; emission; fuels; additives

1. Introduction

The negative environmental impact due to greenhouse gas emission as the impact of excessive use of fossil fuels has encouraged scientists to discover a new resource of energy [1]. One of renewable sources of energy is renewable energy that comes from wind, solar, hydro, bioenergy and other sources [2,3]. However, the main problem of most renewable energy is unstable production because of an availability issue. For example, solar energy is only available for specific hours daily and therefore requires an energy storage device to store energy. The most reliable energy storage currently available is a battery however its capacity is quite limited. Due to this reason, some researcher's attempt to

discover a new type of material for energy storage [4,5]. Even though researchers claim these materials are promising, commercial production is not quite possible at this moment due to unavailability. As a result, various industries are still depended on fossil-based fuel-based internal combustion engine (ICE). ICE is a crucial driver of industrial advancement. Without the transportation sector, the living standard of today would not have been possible to attain. There are two types of internal combustion engines: Spark-ignition (SI) and compression ignition (CI). The former offers lower efficiency and a lower compression ratio, which might be due to a higher knock. Whereas the latter produces harmful air pollutants such as submicron aerosols, carbon monoxide (CO), carbon dioxide (CO₂), oxides of nitrogen (NO_x), sulphur dioxide (SO₂) and hydrocarbon (HC); these emissions are responsible for deteriorating air quality [6]. Though the term aerosol is often associated with particulate matter (PM), however, technically it refers to both the condensed phase particles and the gaseous medium in which they are suspended. Incomplete combustion, sea salt and dust are the main sources of the primary PM emission. The condensation of low volatility gases such as sulfuric acid, ammonia and functionalized organic compounds are responsible for the formation of secondary PM in the atmosphere. Many airborne particles comprise both anthropogenic and natural components. The PM components are categorized in terms of aerodynamic diameter size, e.g., 10 µm and smaller (PM₁₀) and 2.5 µm and smaller (PM_{2.5}) [7]. Atmospheric PM emission directly or indirectly contributes to the reduction in visibility associated with poor air quality, which affects atmospheric radiation transmission and hence climate. The emission produced by the Internal Combustion engines is responsible for global warming and also diversely affects human health.

Researchers worldwide are looking for better combustion techniques to increase engine efficiency, lower the fuel consumption and reduce emission levels. One ideal solution can be the introduction of HCCI. When the homogenous charge mixtures reach chemical activation energy, combustion begins in multiple spots. Compared to conventional compression ignition (CI) and spark ignition (SI) combustion methods, HCCI mode is noticeably faster [8].

Table 1 shows comparative parameters related to the combustion processes of the SI, CI and HCCI engine [9]. The conversion of conventional CI and SI engines into the HCCI mode can play a significant role in increasing the thermal efficiency and lower emissions [10,11]. Different types of renewable fuels and additives can also be used in the HCCI mode. In general, the HCCI mode uses a lean air-fuel mixture, which can be auto ignited and then burnt with invisible flame promulgation [12,13].

Table 1. Comparative parameters of spark ignition (SI), compression ignition (CI) and homogenous charge compression ignition (HCCI) engines [9].

Parameters	SI	HCCI	CI
Ignition method	Spark ignition	Auto-ignition	Compression ignition
Charge	Premixed homogeneous before ignition	Premixed homogeneous before ignition	In-cylinder heterogeneous
Ignition point	Single	Multiple	Multiple
Throttle loss	Yes	No	-
Compression ratio	Low	High	-
Speed	High	Low	-
Combustion flame	Flame propagation	Multi-point auto-ignition	Diffusive flame
Fuel economy	Good	Best	Better
Max. efficiency	30%	>40%	40%
Major emissions	HC, CO and NO _x	HC and CO	NO _x , PM and HC
Injection type	Gasoline direct injection	Port and direct injection	Direct injection
Equivalence ratio	1	<1	-

The benefits of using HCCI are: (i) It operates at fuel-lean condition and compression ratios (>15) comparable to those of diesel engines, thus it is possible to achieve higher efficiencies compared to SI engines [14–16]; (ii) a wide range of fuels can be used [15,17–19] and (iii) it is able to produce lower PM and NO_x [20] emissions, which have negative health and environmental effects. On the other

hand, HCCI modes have some disadvantages, which include higher emissions of HC and CO [21–23]. These regulated emissions can be lowered by using an oxidizing catalyst as the exhaust contains enough oxygen.

When HCCI is applied in a gasoline engine, it shows visible improvement in fuel consumption and emissions like NO_x and particulate matter as compared to SI combustion [24–28]. However, the combustion characteristics of HCCI from gasoline engines are slightly different from HCCI and diesel engines. Usually, HCCI gasoline combustion starts at the point of the crank angle position where 10% of charge has burned (CA10) [29]. It is found that exhaust gas recirculation (EGR) dilution has a more significant effect than air dilution on the start of combustion when the EGR rate remains 0%–40% [29]. When the EGR rate exceeds 40%, the beginning of combustion gradually becomes dependent on the relative air-fuel ratio (λ). It is observed in the literature [29] that the combustion duration depends on λ when the EGR rate is 0%–30%. When the EGR rate exceeds 30%, the combustion duration gradually becomes dependent on the EGR rate rather than λ . Among many reasons for this may be that the dilution effect of EGR can slow the reaction for auto-ignition.

To understand the fundamentals of HCCI combustion, it is essential to know about NO_x and soot formation. Figure 1 represents the regions of formation of NO_x and soot with respect to equivalence ratio–temperature (ϕ – T) [9]. NO_x refers to both nitric oxide (NO) and nitrogen dioxide (NO_2). The formation of NO_x in the combustion products largely depends on the combustion temperature [29]. From Figure 1, NO_x is generally produced at high temperature and low equivalence ratios. If the flame temperature is maintained below 2200 K, it is possible to reduce NO_x formation [30]. Contrary to NO_x formation, soot formation occurs at high equivalence ratio and moderate flame temperature. For diesel engine operation, strategies to reduce NO_x formation usually increases soot emissions and vice versa.

By controlling the flame temperature and equivalence, it is possible to reduce both NO_x and soot emissions [31]. HCCI combustion is based upon these criteria; lower flame temperature and sufficient air and fuel mixture to increase the homogeneity of the charge. As a result of the homogeneous in-cylinder mixture, the temperature and pressure will rise throughout the compression stroke and simultaneous auto-ignition will occur across the cylinder. NO_x formation is reduced by maintaining low local temperatures as a result of the absence of a high-temperature flame front [32]. The homogenized lean mixture will result in low equivalence ratio, which in turn will reduce soot formation [33].

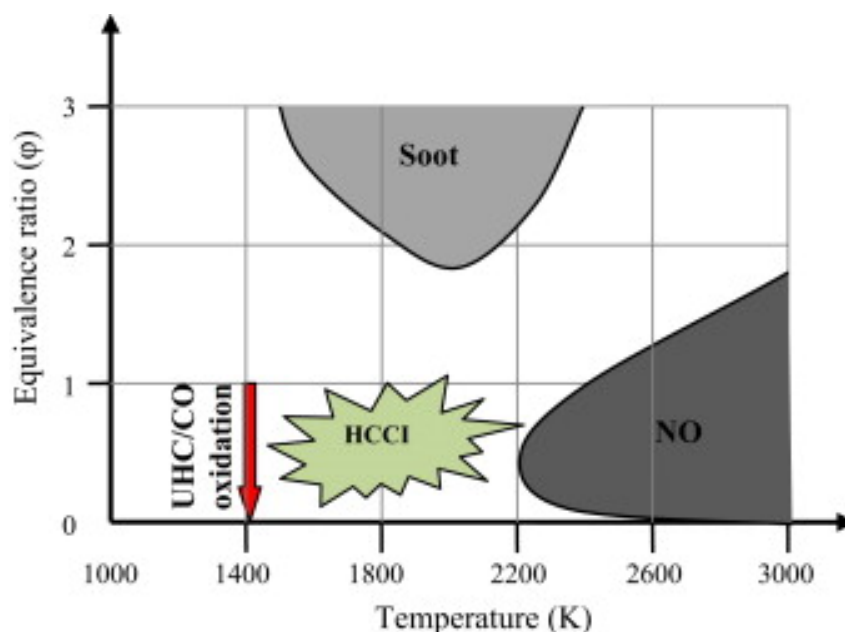


Figure 1. Equivalence ratio versus temperature [9].

As HCCI combustion technology has the potential to replace conventional CI or SI engines, this paper aims to review peer-reviewed articles, which report recent advancements of this technology. This review focuses on the role of the HCCI mode on the performance and emission parameters along with the effects of various fuels, additives on those parameters of the HCCI engine. This article can be used as a good reference to understand the role of HCCI mode and develop the HCCI technology in order to lower emissions.

2. Comparative Performance Analysis of HCCI Engine

Acceptability of engine depends mainly on its performance criteria. Some of the performance criteria include cylinder pressure (CP), brake specific fuel consumption (BSFC), heat release rate (HRR), brake thermal efficiency (BTE), etc. Furthermore, several other factors such as fuel properties, air-fuel mixture, fuel injection pressure and timing, the amount of injected fuel, fuel spray pattern, etc. significantly affect engine performance [34]. This review section compares engine performance between HCCI and SI operation modes focusing on the criteria mentioned above and the results are summarized in Table 2.

2.1. Cylinder Pressure

The CP measurement is considered to be very important because it provides valuable information like the peak pressure, indicated mean effective pressure, P–V diagram, heat release rate, combustion duration, fuel supply effective pressure, ignition delay, etc. [35]. It is the predominant view that compared to SI engines, HCCI engines are able to achieve higher cylinder pressure [25,26,28,29,36–46]. For example, Polovina et al. [26] investigated engine performance, emission and combustion characteristics of the HCCI engine. The authors used gasoline and gasoline–ethanol blends to operate the HCCI engine. They carried out a comparison between the HCCI engine and conventional SI and spark-assisted compression ignition (SACI) engines under various conditions like naturally aspirated, boosted, lean and stoichiometric. For the experiment, they achieved HCCI combustion by modifying a four-cylinder, four-stroke, naturally aspirated, air-cooled and port fuel injected gasoline engine. The authors reported that at partial load, the cylinder pressure of HCCI is quite similar to that of conventional SI engines, but at higher load, the cylinder pressure exceeds the SI one (approx. 40% increase). These results were also supported by Szybist et al. [36]. In their experiment, the authors varied the engine load (350 to 650 kPa), net indicated mean effective pressure ($IMEP_{net}$), intake manifold pressure (98 to 190 kPa) and EGR rate. The authors reported that increasing EGR at constant manifold pressure and increasing manifold pressure at constant EGR, both retarded the combustion. Yap et al. [25] used hydrogen as a fuel to operate the HCCI engine and investigated the combustion and emission characteristics. Their experiment was done in a modified single-cylinder, four-stroke and port fuel injected gasoline engine and also observed higher cylinder pressure than SI engines, which is 3%. However, they concluded this percentage of higher cylinder pressure at a reasonable level. Kobayashi et al. [39] also investigated the combustion, performance and emission characteristics of HCCI engines. Their experiment was conducted on both single- and four-cylinder gasoline engines using gasoline as fuel. However, they observed 6% higher cylinder pressure compared to SI while HCCI operation was done on a turbocharged mode. To mitigate it, they used a blend of natural gas with gasoline. Al-khairi et al. [47] compared the cylinder pressure between HCCI and SI combustion and found no remarkable difference between them. They concluded that HCCI pressure rise was more fast and narrower than SI. To produce 4 N.m torque, SI mode showed a very high pressure rise, which made the engine unstable. However, an exceptional result was obtained by Milovanovic et al. [42]. They investigated the combustion characteristics of an HCCI engine in a modified gasoline engine under constant speed and different loads. The authors concluded that the cylinder pressure could be controlled by varying the amount of trapped residual gas (TRG) in the fresh air/fuel mixture inside the cylinder, which will also enable controlling the auto-ignition timing and heat release rate.

2.2. Brake Specific Fuel Consumption and Brake Thermal Efficiency

BSFC is a measure of fuel efficiency [48], which is used to compare the efficiency of IC engines with a shaft output [49]. Researchers have reported both increased and decreased BSFC in HCCI engines compared to that of conventional SI engines [26,28,29,36,37,39,41,43,44,46,47]. Szybist et al. [36] modified a boosted single-cylinder research engine for HCCI combustion, which was equipped with the direct injection (DI) fuelling system. Their purpose was to apply different control systems to achieve an HCCI high load limit. The authors reported that to increase the load, it is necessary to increase the manifold pressure and external EGR. In their experiment, manifold pressure was varied from 98 kPa to 190 kPa and EGR was varied from 10% to 30%. While observing the BSFC, it was found that at part load the difference is very low. However, at higher load, the difference with SI combustion becomes greater. The maximum 7% more BSFC was found for HCCI operation compared to SI operation. Similarly, Yun et al. [37] investigated a spark-assisted HCCI combustion and double injection strategy to extend the higher operating load limit of an HCCI engine to maximize the fuel economy benefit of HCCI. They found that spark-assisted HCCI combustion was a key factor to reduce combustion noise and improve fuel economy at the high load condition. Alternatively, the double injection strategy could reduce combustion noise slightly, but the BSFC was increased due to incomplete combustion. Al-khairi et al. [47] investigated the performance of an HCCI engine by modifying a gasoline engine and using natural gas as fuel. The authors observed an improvement in BSFC, which is up to 25% while using HCCI operation compared to SI at low load condition.

Some of the researchers [28,41,43,44,46] found the opposite results due to applying specific techniques and appropriate mixture homogeneity as well as complete combustion. For example, Hyvonen et al. [44] investigated the spark-assisted HCCI combustion in a five-cylinder variable compression ratio (VCR) HCCI engine using gasoline as fuel. They found a lower BSFC than in the SI engine. Another experiment was done by Lemberger et al. [46] on an HCCI engine using diethyl ether (DEE) as fuel. Later, they compared the performance result with an SI engine and found a 3% lower BSFC than for SI. However, contradictory to all the studies stated above, Polovina et al. [26] reported no significant difference in BSFC between HCCI and SI engines.

Most of the researchers reported higher BTE of HCCI engines compared to that of diesel engines [26,29,36,38–41,43–47,50–52]. This is a common conclusion of all researchers, that when an HCCI engine is modified from an SI engine it provides higher BTE than conventional SI engines. Gotoh et al. [40] compared the BTE of an HCCI engine and gasoline engine, both operating with gasoline. The authors applied a blow-down supercharging system and varied the intake temperature and pressure ranging from 32 °C to 50 °C and 100 kPa to 200 kPa. The authors concluded that by imposing the blow-down charging system, the operating load limit of the HCCI engine could be increased along with the BTE. An investigation and comparison of combustion, performance and emission characteristics of two-stroke and four-stroke spark ignition and HCCI operations in a DI gasoline engine were conducted by Zhang et al. [41]. They observed that, among all the experiments, when the engine was run in the HCCI mode, it provided the highest BTE. About 5% higher efficiency than SI was achieved through HCCI operation. The authors explained this phenomenon as the pumping loss in the SI mode is the reason for having lower BTE, which can be recovered successfully in HCCI mode. Similar experiments were done by Machrafi et al. [50] and Zhao et al. [51], but they used different types of fuels, i.e., blends of iso-octane and n-heptane as well as gasoline and n-butanol. In all these cases, they found higher BTE.

Table 2. Summary of HCCI engine performance results compared to SI engines using various fuels.

Engine	Test Condition	Fuel	Performance	References
4-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	↑: Cylinder pressure ↓: BSFC	[28]
4-cylinder, 4S, AC, PFI, CR: 10.8:1, RS: 2000 rpm	VS and VL	Gasoline and ethanol	↑: Cylinder pressure -: BSFC ↓: BTE	[26]
1-cylinder, 4S, AC, PFI, CR: 12.5:1 and 15:1, RS: 1500 rpm	CS and VR	Propane	↑: Cylinder pressure	[25]
1-cylinder, 4S, AC, GDI, CR: 11.3:1, RS: 2500 rpm	VS and CL	Gasoline	-----	[53]
1-cylinder, 4S, AC, DI, CR: 11.85:1	CS and VL	Gasoline–ethanol blend	↑: Cylinder pressure, BSFC, BTE	[36]
1-cylinder, 4S, AC, PFI, CR: 12:1, RS: 2500 rpm	VS and VL	Gasoline	↑: Cylinder pressure, BSFC	[37]
1-cylinder, 4S, AC, GDI, CR: 14:1	CS and VL	Gasoline–ethanol blend	↑: Cylinder pressure, BTE	[38]
1-cylinder, 4S, AC, PFI, CR: 14:1, RS: 2100 rpm	CS and VL	Gasoline and CNG	↑: BSFC, BTE	[47]
1-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	↑: Cylinder pressure, BSFC, BTE	[29]
1 and 4-cylinder, 4S, AC, PFI, CR: 17:1, 19:1 and 21:1 RS: 1500 rpm	CS and VR	Gasoline and CNG	↑: Cylinder pressure, BSFC, BTE	[39]
4-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	↑: Cylinder pressure, BTE	[40]
1-cylinder, 4S, AC, GDI, CR: 11.78:1, RS: 1500 rpm	CS and VL	Gasoline	↑: Cylinder pressure, BTE ↓: BSFC	[41]
4-cylinder, 4S, AC, GDI, CR: 4-16:1, RS: 600 rpm	CS and VR	n-heptane and iso-octane	↑: BTE	[50]
1-cylinder, 4S, AC, PFI, CR: 10.66:1, RS: 1500 rpm	VS and CL	Gasoline and n-butanol	↑: BTE	[51]
1-cylinder, 4S, AC, PFI, CR: 10.66:1, RS: 1500 rpm	VS and CL	n-butanol	↑: BTE	[52]
1-cylinder, 4S, WC, PFI, CR: 10.5:1, RS: 2000 rpm	CS and VL	Gasoline	↓: Cylinder pressure	[42]
1-cylinder, 4S, AC, GDI, CR: 12:1, RS: 1500 rpm	VS and CL	Gasoline	↑: Cylinder pressure, BTE ↓: BSFC	[43]
5-cylinder, 4S, AC, PFI, CR: 10-30:1	CS and VL	Gasoline	↑: Cylinder pressure, BTE ↓: BSFC	[44]
1-cylinder, 4S, AC, GDI, CR: 11.85:1, RS: 2000 rpm	CS and VL	Gasoline–ethanol blend	↑: Cylinder pressure, BTE	[45]
1-cylinder, 4S, AC, PFI, CR: 8:1:1, RS: 9000 rpm	VS and CL	Diethyl ether	↑: Cylinder pressure, BTE ↓: BSFC	[46]

Note: 4S = four-stroke, WC = water-cooled, AC = air-cooled, NA = naturally aspirated, PFI = port fuel-injection, GDI = gasoline direct injection, CR = compression ratio, RS = rated speed, ↑: increase, ↓: decrease, -: no change, VS = variable speed, CS = constant speed, VL = variable load, CL = constant load, VR = variable compression ratio.

Aceves et al. [43] proposed a multi-zone model for predicting the combustion, performance and emissions characteristics of an HCCI combustion engine. From their prediction, it was found that BTE was always higher than conventional SI engines in any load condition. However, an exceptional result was obtained by Polovina et al. [26]. They investigated the combustion characteristics of an HCCI engine in a modified gasoline engine under various conditions, including naturally aspirated, boosted, lean and stoichiometric. They reported that at partial load, the BTE of HCCI is quite similar to that of conventional SI engines, but at higher load, the BTE was less than for SI engines.

3. Comparative Emission Analysis of HCCI Engine

Various countries have imposed robust emission control measurements to reduce transport-related emission [54–62]. Emission from HCCI engines includes CO, HC, NO_x and PM. In general, NO_x and PM emission from HCCI engines are quite low [21]. However, HCCI produces high levels of unburned HC and CO [21,63]. All these emissions depend on the engine operating conditions, engine design and fuel quality [64]. This section briefly discusses various emission parameters of HCCI engines using

various fuels and compares with that of SI engine. The comparison of the results is also presented in Table 3.

3.1. CO Emission

Literature reports that CO emission of HCCI engines is higher than that from gasoline engines [25,39,41,43,44,46,47,53]. Kobayashi et al. [39] investigated the engine performance, combustion characteristics and emission parameters of HCCI engines using both single and four-cylinder gasoline engines operating on gasoline. In their experiments, intake temperature was varied from 383 K to 413 K and equivalence ratios from 0.20 to 0.40. A huge amount of CO emission was observed while the engine was running in HCCI mode compared to SI mode. Highest CO was measured as 6000 ppm at 413 K and 0.20 equivalence ratio, which is almost 10 times higher than the CO emission achieved from SI mode. Aceves et al. [43] proposed a multi-zone model for predicting the combustion, performance and emissions characteristics of an HCCI combustion engine. From their prediction, it was found that CO was always higher than in conventional SI engines in any load condition. The combustion efficiency dictates the amount of CO₂ and CO and is defined as the ratio of CO₂ to the total fuel carbon present in the exhaust, including CO, CO₂ and UHC [65]. CO is generally formed in the cylinder crevices, where it is not possible to attain complete combustion due to being too cold [66]. This needs to be solved to reduce CO emission. For complete conversion of CO to CO₂, the temperature needs to be above 1500 K [67]. However, in HCCI combustion mode, the peak gas temperature remains well below the required temperature and thus emits a significant amount of CO. Some researchers were able to reduce the CO emission of HCCI engines by solving the problems mentioned earlier [28,42,50]. For e.g., Kuboyama [28] reported lower CO emission achieved through the reduction of EGR, which can be due to relatively rich fuel concentration. However, this strategy increases NO_x emission, which can be attributed to lesser amount of diluted gas. The author also reported that reducing coolant temperature could reduce the CO emission slightly [28].

3.2. HC Emission

In general, HCCI combustion increases HC emission [68], which is another challenge for the broad application of HCCI engines. Kobayashi et al. [39] investigated the emissions characteristics of an HCCI engine by modifying a gasoline engine and using natural gas as fuel. They found higher HC than in SI engines. Lemberger et al. [46] converted a single-cylinder, air-cooled, 25cc, 4-stroke SI engine to run in HCCI mode by applying various combustion control system. The authors used diethyl ether to operate the engine. Later they compared the emission result with an SI engine and found higher HC than for SI. In their experiments, engine speed was varied from 1000 rpm to 4000 rpm and equivalence ratios from 0.30 to 0.75. A huge amount of HC emission was observed while the engine was running in HCCI mode compared to SI mode. Highest HC was measured as 950 ppm at 3000 rpm and 0.30 equivalence ratio, which is almost 7 times higher than the HC emission achieved from SI mode. Incomplete combustion occurs when the in-cylinder temperature is low and thus increases HC emissions [22,69,70]. HC is mainly formed in the crevices of the cylinder, which are too cold for complete consumption [66]. For example, higher concentrations of natural gas and hydrogen in gasoline engines can reduce UHC and CO emission levels as they reduce the wall-wetting effect on the cylinder liner [71]. By solving the problems mentioned above, some researchers [28,42,47] obtained lower HC in HCCI engines. For e.g., Milovanovic [42] was able to reduce the HC emission by reducing the coolant temperature.

3.3. NO_x Emission

NO_x formation generally occurs through an elevated temperature reaction between nitrogen and oxygen to form NO. Further reaction covers some of this NO into NO₂. The final flame temperature of HCCI engines is usually well below 2000 K as the engine mostly operates at the fuel-lean condition. Therefore, the chemical reactions that are responsible for NO_x emission remain

mostly inactive [72]. As a result, compared to SI engines, NO_x emission from HCCI engines is quite negligible [25,26,36,39,42–44,46,50]. Polovina et al. [26] evaluated engine performance, combustion characteristics, and emission of HCCI engines. To achieve HCCI combustion, the authors modified a four-cylinder, four-stroke, naturally aspirated, air-cooled and port fuel injected gasoline engine. The authors used gasoline and gasoline-ethanol blends at various conditions (which includes naturally aspirated, boosted, lean and stoichiometric). Finally, the authors compared the results with conventional SI and spark-assisted compression ignition (SACI) engines. They reported low level of NO_x (<10 ppm) emission for HCCI engines. Kobayashi et al. [39] observed much lower NO_x in HCCI engine than in SI engines. In every test condition, NO_x was below 2 ppm. Similarly, Lemberger et al. [46] reported similar results. Aceves et al. [43] proposed a multi-zone model for predicting the combustion, performance and emissions characteristics of an HCCI combustion engine and reported very low NO_x emission (<2.5 ppm) compared to conventional SI engines in any test condition. Surprisingly, Yun et al. [37] and Ji et al. [38] found a slightly higher NO_x than with SI engines. Yun et al. [37] reported that NO_x emission increased with the increase of fuel mass injection, which can be attributed to locally available hotspot generated by the increased inhomogeneity. Ji et al. [38] reported that ignition improvers such as 2-Ethylhexyl nitrate (EHN) increase NO_x emission in HCCI mode.

Table 3. Summary of HCCI engine emission results compared to SI engines using various fuels.

Engine	Test Condition	Fuel	Emission	References
4-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	↓: CO, NO _x -; HC	[28]
4-cylinder, 4S, AC, PFI, CR: 10.8:1, RS: 2000 rpm	VS and VL	Gasoline and ethanol	↓: NO _x	[26]
1-cylinder, 4S, AC, PFI, CR: 12.5:1 and 15:1, RS: 1500 rpm	CS and VR	Propane	↑: HC, CO ↓: NO _x	[25]
1-cylinder, 4S, AC, GDI, CR: 11.3:1, RS: 2500 rpm	VS and CL	Gasoline	↑: HC, CO ↓: NO _x , PM	[53]
1-cylinder, 4S, AC, DI, CR: 11.85:1	CS and VL	Gasoline-ethanol blend	↓: NO _x	[36]
1-cylinder, 4S, AC, PFI, CR: 12:1, RS: 2500 rpm	VS and VL	Gasoline	↑: NO _x	[37]
1-cylinder, 4S, AC, GDI, CR: 14:1	CS and VL	Gasoline-ethanol blend	↑: NO _x	[38]
1-cylinder, 4S, AC, PFI, CR: 14:1, RS: 2100 rpm	CS and VL	Gasoline and CNG	↑: CO ↓: HC, NO _x	[47]
1-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	↓: NO _x	[29]
1 and 4-cylinder, 4S, AC, PFI, CR: 17:1, 19:1 and 21:1 RS: 1500 rpm	CS and VR	Gasoline and CNG	↑: HC, CO ↓: NO _x	[39]
4-cylinder, 4S, AC, PFI, CR: 12:1, RS: 1500 rpm	CS and VL	Gasoline	-----	[40]
1-cylinder, 4S, AC, GDI, CR: 11.78:1, RS: 1500 rpm	CS and VL	Gasoline	↑: HC, CO ↓: NO _x	[41]
4-cylinder, 4S, AC, GDI, CR: 4-16:1, RS: 600 rpm	CS and VR	n-heptane and iso-octane	↓: NO _x , CO	[50]
1-cylinder, 4S, AC, PFI, CR: 10.66:1, RS: 1500 rpm	VS and CR	Gasoline and n-butanol	↓: NO _x	[51]
1-cylinder, 4S, AC, PFI, CR: 10.66:1, RS: 1500 rpm	VS and CL	n-butanol	↓: NO _x	[52]
1-cylinder, 4S, WC, PFI, CR: 10.5:1, RS: 2000 rpm	CS and VL	Gasoline	↓: HC, CO, NO _x	[42]
1-cylinder, 4S, AC, GDI, CR: 12:1, RS: 1500 rpm	VS and CL	Gasoline	↑: HC, CO ↓: NO _x	[43]
5-cylinder, 4S, AC, PFI, CR: 10-30:1	CS and VL	Gasoline	↑: HC, CO ↓: NO _x	[44]
1-cylinder, 4S, AC, GDI, CR: 11.85:1, RS: 2000 rpm	CS and VL	Gasoline-ethanol blend	↑: NO _x	[45]
1-cylinder, 4S, AC, PFI, CR: 8;1:1, RS: 9000 rpm	VS and CL	Diethyl ether	↑: HC, CO ↓: NO _x	[46]

Note: 4S = four-stroke, WC = water-cooled, AC = air-cooled, NA = naturally aspirated, PFI = port fuel-injection, GDI = gasoline direct injection, CR = compression ratio, RS = rated speed, ↑: increase, ↓: decrease, -: no change, VS = variable speed, CS = constant speed, VL = variable load, CL = constant load, VR = variable compression ratio.

4. Effects of Fuel Types on HCCI Engine Combustion

It is important to use a specific type of fuel for conventional internal combustion engines, which is governed by the combustion mode and engine set-up. Gasoline is the most suitable fuel for spark-ignition engines as it has good volatility and anti-knock characteristics. Diesel due to its innate high viscosity and lower auto-ignition resistance is most suitable for diesel engines. However, in HCCI combustion mode, it is possible to use any fuel type, which can be vaporized and mixed with air before ignition [73]. As previously described, auto-ignition of the air-fuel mixture is necessary for the occurrence of ignition, fuel selection is a very important factor that significantly affects the engine performance and the control procedures. The key fuel properties for HCCI engines are fuel volatility and auto-ignition qualities. Auto-ignition points vary with fuel types and decrease with the increase of carbon atoms in the HC [73]. A huge number of researchers [71,74–92] have investigated the effects of various fuels on HCCI engines. According to Epping et al. [10], to form a homogeneous charge easily, the fuel must have a high volatility property. Chemically, fuels that have single-stage ignition react less with the change in load and speed. This enables the HCCI engine to be operated over a wide range of conditions. Christensen et al. [93] reported that any liquid fuel could be utilized as a part of an HCCI engine utilizing a variable compression ratio.

Diesel fuel auto-ignites more readily than gasoline. However, the non-premixed diesel engine is becoming a less suitable power production device due to the high oxides of nitrogen (NO_x) and soot emissions. Due to these challenges faced by the traditional diesel engine, diesel HCCI combustion gained a lot of attention in the 1990s. Three different approaches were followed to achieve HCCI combustion with diesel fuel: Premixed HCCI, early direct-injection HCCI and late direct-injection HCCI. Ryan et al. [79], followed by Gray et al. [80], used the premixed HCCI approach in a single-cylinder, four-stroke, variable compression ratio engine. Fuel injectors introduced diesel fuel into the intake air stream and the mixture temperature was increased using an intake air heater. Compression ratio variation along with EGR was used to achieve HCCI. The study highlighted three issues with premixed diesel HCCI. First, when compression ratio was varied premature ignition and knocking occurred. Second, high intake temperatures were required to avoid the accumulation of liquid fuel on the surfaces of the intake system. Third, HC emissions significantly increased compared to diesel due to reduced combustion efficiency. Several others [93–96] worked with premixed diesel HCCI and their results indicate that substantial reductions of NO_x and soot emissions were achieved using this approach. However, it was also clear from these publications that premixed diesel HCCI is not the best strategy for diesel-fuelled HCCI and alternative fuel-delivery or mixing techniques have to be developed. Early direct-injection (i.e., direct fuel injection well before TDC) is a favored diesel HCCI technique for two reasons. First, easy vaporization and better mixing are achieved when fuel is injected in hot compressed air during the compression stroke. Second, with an appropriately designed injector, fuel wall wetting can be minimized, which increases combustion efficiency and reduces emissions. Several publications [97–100] reported using early direct injection to achieve diesel HCCI. Nissan Motor Company developed a process called MK (modulated kinetics) that falls in the regime of late direct-injection diesel HCCI. The principles of this process were described by Mase et al. [81] and Kimura et al. [82]. For this process, a long ignition delay period and rapid mixing was required, which assures premixed combustion and hence low smoke. Large amounts of EGR reduced the combustion temperature, resulting in low NO_x . The ignition delay was also extended by increased EGR and retarded injection (near the top dead centre, TDC). Rapid mixing was achieved by combining a high swirl with improved combustion-bowl geometry. The MK process achieved low emissions with appropriate combustion control.

Gasoline has a lower boiling point and higher octane number. For this reason, using gasoline fuel in HCCI engines is the reason for early combustion, which limits the operating range of HCCI engines [101]. Ultimately, high combustion pressure and a knock are created [101]. Delayed combustion can solve these two major problems. Several approaches have been used by many researchers [51,102,103]. Kitae et al. [103] used liquid petroleum gas with gasoline. Compared to gasoline, LPG has

increased octane number (ON) and lower heating value, which lowers the compression pressure and temperature [104]. As LPG also contains less carbon atoms than the SI or CI engine, it is possible to reduce or diminish carbon dioxide (CO₂) emission when LPG is blended with gasoline or diesel [105]. Some scientists [51,52,102,106] blended n-butanol with gasoline to solve these problems and reported the early start of auto-ignition and shorter combustion duration.

Other fuels besides gasoline and diesel are also used for HCCI engines. Since natural gas is the second most abundant fuel, many researchers have studied the feasibility of using natural gas [107–111] as a fuel in HCCI engines. Due to the high octane rating of natural gas (of the order of methane, which has an octane rating of 107), high compression ratio [93], high intake temperatures or additives promoting auto-ignition like NO_x [112] and dimethyl ether [113] are required in natural-gas-fuelled HCCI engines. Performance aspects of propane-fuelled HCCI engines are discussed in [114,115]. Alcohols [116,117] exhibit good auto-ignition properties and hence are excellent HCCI fuels, with a significantly larger operating range than most other fuels. Hydrogen [118] is also studied as an alternative HCCI fuel. In addition to the above-mentioned neat fuels, many fuel blends and additives like dimethyl ether, diethyl ether, dimethoxy methane or di-tertiary butyl peroxide [119] can be used in HCCI combustion.

5. Effects of Additives on HCCI Engine Combustion

Some chemicals can retard or advance the heat release process of auto-ignition. Thus using an ignition promoter or inhibitor, it is possible to control HCCI auto-ignition. For this reason, the impact of different additives was likewise the centre of interest for different scientists [103,120–126]. The ignition timing and combustion duration can be controlled through water injection. Christensen et al. [93] and Iida et al. [123] reported lower initial gas temperature when water was injected into the fuel. Numerical additives were utilized by Leppard [127] and Westbrook [128] to affect auto-ignition bringing on engine knock, which is about indistinguishable to the kinetics of HCCI ignition. Some researchers have used hydrogen-enriched natural gas mixtures to operate HCCI engines [129,130]. Hydrogen expansion in a natural gas mixture can build in-cylinder peak pressure, reduce ignition temperature and ignition delay time and expand indicated power [8]. It additionally permits the augmentation of the lean limit of the natural gas mixture, without entering the lean misfire region, while accomplishing amazingly low emissions [130]. Hydrogen expansion in ultra-low sulphur diesel (ULSD) advances incompletely premixed compression ignition and results in better performance and decreased emissions [131]. Hydrogen expansion test (in HCCI mode) with a diesel engine showing only small amounts is able to diminish the ignition delay and enhance engine performance [132]. However, to avoid an extreme knock the expansion of hydrogen to diesel should not be more than 15% in the energy ratio. Ryan et al. [79] utilized a heptane/iso-octane mixture in an HCCI engine and found that this mixture is not suitable to control the ignition timing, but at low temperature, the thermal efficiency was quite similar to that of a CI engine. Furthermore, formaldehyde-doped lean butane/air mixtures [17] retarded ignition timing and decreased combustion efficiency.

6. Effects of Engine Parameters on HCCI Engines Combustion

This section discusses the effects of various engine parameters on the ignition timing, combustion rate, performance and emissions to develop an appropriate control method to optimize ignition timing and smooth HRR [133].

6.1. Intake Temperature

Intake air temperature is one of the key controlling parameters of HCCI combustion as it affects the time–temperature history of the mixture. If the intake air temperature is high, it results in the advancement of the start of combustion (SOC); however, it reduces volumetric efficiency [134]. Intake air temperature affects the combustion and emissions formation via two distinctive pathways. Reduced intake temperature extends ignition, which leads to enhanced air/fuel premixing. Additionally, it is

possible to reduce the adiabatic flame temperature of a fuel parcel by mixing at a particular equivalence ratio, which will steer away from the fuel element from the NO_x –soot formation region on the equivalence ratios–temperature map. Lu Chen [133] reported that intake air temperature greatly affects HCCI combustion and when this temperature is increased it advanced the combustion phase and reduced the combustion duration. The author also reported that when intake charge temperature was increased (from 31 °C to 54 °C), NO_x emissions increased linearly (from 10 ppm to 50 ppm) at a fixed fuel delivery rate, engine speed and 30% EGR, this is also supported by another study [135]. However, unburned HC and CO was not affected by the intake temperature [133]. Peak soot luminosity of an HSDI diesel engine significantly reduced when intake charge temperature was decreased from 110 °C to 30 °C under a load condition of 3 bar IMEP [136]. This can be attributed to lower soot temperatures and reduced soot formation. However, it was not possible to completely diminish soot; in-cylinder soot luminosity was detected even at 30 °C.

6.2. Intake Pressure

To extend operating load range, control the combustion and lower the NO_x emission, higher EGR rates turbocharging can be used in HCCI combustion [137–139]. Higher EGR rates may potentially restrict the power loss through turbocharging. It has been reported that using a supercharging in a diesel engine can increase the power and fuel quantity at a lower compression ratio. Similarly, 80 kPa boost pressure was achieved in the premixed CI combustion system using higher EGR rates turbocharging, which is close to the pressure of a conventional full load diesel engine [140]. Up to 16 bar brake mean effective pressure (standard 21 bar) and lower NO_x emission (0.1 g/kW-hr) was also reported as a result of the turbocharging application in a six-cylinder diesel engine [141]. To limit NO_x emission boost needs to be increased compared to conventional engines at the same load. In this case, turbocharger efficiency plays a remarkable role in HCCI to lower the pumping losses. However, two-stage turbocharging with intercooling is a recommended option for the higher load condition and appropriate controlling of the turbocharging system significantly reduces the pumping loss through ‘over boost’.

6.3. Compression Ratio (CR)

To allow complete injection of fuels before ignition, the ignition delay is prolonged by limiting the CR. Reduction of CR is the precondition for premixed combustion and to reduce the gas temperature during the compression stroke of the engine [142]. Reduction of CR of a diesel engine from 18:1 to 16:1 can extend the premixed combustion and lower temperature to increased load condition [143]. Olsson Tunestål [144] demonstrated that by reducing the CR, knock-in HCCI diesel combustion might be controlled. Reduced compression ratio prevents explosive self-ignition from occurring by reducing the temperature rise of the end gas. Peng Zhao [145] investigated the effects of reducing CR on HCCI combustion and reported that reduction of CR from 18:1 to 12:1 reduced the knocking of the engine and increased maximum IMEP from 2.7 bar to 3.5 bar. The NO_x and soot emission in a partially charged compression ignition (PCCI) engine was also reduced due to the reduction of CR from 18.4:1 to 16.0:1 under premixed combustion condition [146]. The reason was explained by the ignition delay, which lowers the gas temperature during the compression stroke and offers better air-fuel mixture; thus reduces the NO_x and soot emissions.

7. HCCI Challenges

Though there are several advantages of HCCI combustion, there are some lacking that need to be overcome to extensively use this method. The current challenges of HCCI combustion are low power output, high HC and CO emission, combustion timing control, homogenous charge mixture preparation and weak cold-start [147]. Combustion phasing control can be achieved by diluting the mixture to retard or advance combustion [148]. Najt and Foster [149] and Christensen and Johansson [150] were able to retard the combustion phase by introducing EGR. The hot gases mix with cooler inlet mixture

of air and fuel and increase the charge temperature. EGR also reduces the oxygen content of the mixture. The burnt gases present in the mixture will take part in the auto-ignition. Christensen and Johansson [150] were able to achieve 16 bars increase of IMEP through 50% EGR rate. Thus, EGR is one of the ways of improving the load capacity of HCCI modes. Atkins and Koch reported that introduction of EGR also increases the overall efficiency [151]. Combustion phasing can also be controlled by mixing two different fuels with a different octane number [152]. However, a high octane number sometimes does not participate in the oxidation reaction, which in turn can increase HC emission. Atkins and Koch also reported that using dual fuel with different octane numbers could increase the HCCI engine operating range [151]. Use of reformer gases (RG) can also control the combustion phasing [153,154]. Dec and Berntsson reported that fuel stratification could be used to control the combustion phasing as it retards ignition timing; however, too much stratification can result in unstable combustion [155,156]. Homogenous mixture preparation can be achieved through fuel injection in a highly turbulent port flow for gaseous and highly volatile fuels [157]. Introduction of a supercharger or turbocharger can boost intake airflow, which will help in improving the load output of the HCCI engine [158,159].

8. Conclusions

The transportation sector plays a crucial role in the development of a nation. However, due to strict emission regulations imposed to cut-down harmful emissions generated by conventional internal combustion engines, it is essential to search for alternative and advanced combustion methods. Homogenous charge compression ignition or the HCCI engine is based upon a hybrid combustion method to resolve the problems associated with conventional IC engines. A thorough review of literature focused on HCCI engines was carried out and it is worth mentioning that the current article comprehensively reviewed and discussed all the aspects as a whole (Table 4). From this study, the following conclusions were derived:

- The chemical kinetics dominates HCCI combustion.
- HCCI combustion can achieve higher thermal efficiency.
- Compared to SI engine, HCCI engine emits much less NO_x and PM emissions and higher HC and CO emissions.
- Various types of fuel can be used in HCCI combustion modes. The fuel choice has a significant impact on engine design and control strategies.
- Auto-ignition of HCCI may be controlled by changing the fuel properties. The addition of an ignition inhibitor in the fuel can make it more chemically reactive.
- Engine parameters have significant influences on HCCI combustion. Better performance from HCCI engine operation is solely dependent on the proper tuning of different engine parameters.

Finally, the HCCI mode of combustion can be incorporated in the conventional SI engines after a few modifications to reduce NO_x and PM emissions, and further improvements are required to overcome the challenges to derive a similar engine performance compared to that of SI combustion.

Table 4. Comparison of the current review article and the review papers published since 2012 on HCCI combustion.

Ref.	Topic Discussed	HCCI Combustion Parameters	HCCI Performance Parameters	HCCI Emission Parameters	Effect of Fuel Choice on HCCI Combustion	Effect of Additive on HCCI Combustion	Effect of Engine Parameter on HCCI Combustion	Challenges of HCCI Combustion	Modelling of HCCI Combustion
This Study		✓	✓	✓	✓	✓	✓	✓	X
[160]	A	X	X	X	✓	✓	X	✓	X
[161]	B	✓	X	✓	✓	X	X	X	X
[162]	C	X	✓	✓	X	X	X	X	X
[163]	D	✓	✓	✓	✓	X	X	X	✓
[164]	E	X	X	X	✓	✓	X	X	X
[165]	F	✓	✓	✓	X	X	X	X	✓
[166]	G	✓	✓	✓	X	X	✓	X	X
[167]	H	✓	✓	✓	X	X	X	X	✓
[168]	I	X	✓	X	✓	X	✓	X	✓
[169]	J	✓	✓	✓	X	X	X	X	✓
[170]	K	X	✓	✓	X	X	✓	X	X

A-Combustion control strategies; B-Effect of bioethanol; C-Use of different piston; D-Effect of alternative fuel; E-Effect of oxygenated fuels; F-Modelling and controller design; G-Effect of engine parameter; H-Heat transfer model; I-Effect of hydrogen and natural gas; J-Combustion modelling; K-Effect of injection strategies.

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