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Medhat Elkelawy, Hagar Alm ElDin Mohamad, Elshenawy Abd elhamid, Mahmoud A. M. El-Gamal

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A Critical Review of the Performance, Combustion, and Emissions Characteristics of PCCI Engine Controlled by Injection Strategy and Fuel Properties

Medhat Elkelawy*[1], Hagar Alm-Eldin Bastawissi [2], E. A. El Shenawy [3], Mahmoud A. M. El-Gamal [4]

[1] Mechanical Power Engineering Departments, Faculty of Engineering, Tanta University, Tanta, Egypt – email: medhatelkelawy@f-eng.tanta.edu.eg
[2] Mechanical Power Engineering Departments, Faculty of Engineering, Tanta University, Tanta, Egypt – email: hagaralmeldin@f-eng.tanta.edu.eg
[3] Mechanical Power Eng. Departments, Faculty of Engineering, Tanta University, Tanta, Egypt – email: el-shinawi.abdulhamid@f-eng.tanta.edu.eg
[4]Mechanical Power Engineering Departments, Faculty of Engineering, Tanta University, Tanta, Egypt – email: mahmoud.elgamal@f-eng.tanta.edu.eg

Abstract: As internal combustion engines (ICEs) produce serious emissions and a big part of greenhouse gases from fuel combustion. Due to the universal concerns about degradation in the ambient environment, limitations on exhaust emissions, depletion of petroleum reserves, and global warming, many strict regulations have been launched on the standard emissions released from engines. These challenges oblige engine researchers worldwide to develop a new strategic balance between engine performance and emissions. Premixed charge compression ignition (PCCI) is a promising technique to overcome these challenges in recent years which can simultaneously reduce NO_x and soot emissions and substantially improve thermal efficiency. The PCCI combustion concept has the advantages of both SI and CI engines, like SI engines as the charge is premixed which produces low emissions and like CI engines the fuel-air mixture is auto-ignited as a result of compression which leads to high thermal efficiency. Normally, PCCI combustion is a single-stage combustion process achieved by employing early injection timing to increase the time available for mixing fuel and air by using single-fuel and split fuel (pilot/main) injection tactics, in which a large fraction of fuel burns in premixed combustion phase resulting in relatively lower in-cylinder temperatures compared to compression ignition (CI) combustion. Thus, the objective of this paper is to provide an inclusive review of the effects of fuel injection timings, ratios, pressure, and fuel properties on the PCCI engine combustion performance improvement and emission reduction, this review has been analyzed extensively based on the published studies to provide and discuss different strategies for the control of PCCI technique of combustion at a wide range of speed and load.

Keywords: PCCI engine; SI and CI engines; NOx emissions; Law Temperature Combustion

I. INTRODUCTION

Nowadays, with the great usage of internal combustion engines (ICEs) to meet the world's energy requirements, automobile applications. As ICEs is the main consumer of fossil fuels and release an enormous amount of harmful pollutants to human health and greenhouse gases. ICEs are producing various emissions and greenhouse gases from fuel combustion[1]. The unburned hydrocarbon (HC), Oxides of nitrogen (NOx), carbon monoxide (CO), carbon dioxide (CO2), smoke opacity, and particulate matter (PM) are main constituents of the exhaust gas[2]. Owing to the high increase of these pollutants, a lot of strict regulations have been launched on the standard emissions released from engines to be acceptable thus oblige engine researchers worldwide to use alternative fuels, exhaust after-treatment devices, and develop clean technologies with lower fuel consumption [3].Compression ignition (CI) engines are the most appropriate device for power generation and transportation applications because of their reliability, durability, high combustion efficiency, and economics due to their high compression ratio and require lower maintenance than gasoline engines[4]. However, there are series concerns about their continual usage due to their undesirable effects on health and the environment as; they release an enormous amount of pollutants and greenhouse gases and their depletion of fossil fuels [5]. In the traditional diesel engine as the ignition delay period is small in addition to the higher viscosity and volatility of diesel fuel a non-homogeneous fuel-air mixture is formed before the start of combustion (SOC), resulting in increased high rates of NOx and soot formation[6]. Low-temperature combustion (LTC) strategies are used to overcome these difficulties which provide lower equivalence ratios and pre-prepared fuel-air mixture which improves the physical processes of fuel and reduces the in-cylinder temperature[7]. Premixed charge compression ignition (PCCI) is one of the most promising LTC strategies for reducing NOx and soot emissions and providing higher thermal efficiency.

This paper reviews important research related to a fundamental understanding the combustion controllability and stability, such as ignition delay, combustion phasing, and heat release rate in PCCI combustion mode. In this regard, this paper aims to provide an inclusive review of the effects of fuel injection timings, ratios, pressure, and fuel properties on the combustion, performance, and emission characteristics of PCCI engines based on the review of published results of mostly experimental works. At a wide range of speeds and loads, the current study aims to provide and discuss different strategies for the control of the PCCI mode of combustion. The controlling strategies are a combination of two or three different techniques like fuel properties like ON and CN, injection timing, ratios and pressure, RPM, rate of EGR, intake air temperature, compression ratio, valve timing, etc. Several injection strategies e.g., injection timing, number of injections and preheating manifold injection, and fuel properties are discussed in detail in this paper. Moreover, this review provides a comprehensive study on the effect of these strategies on engine-out emissions such as NOx, soot, higher HC, and CO emissions, and brake thermal efficiency. Some pivotal articles indicated in this study and the results reported in these articles were used to classify the articles for analyzing the impact of different strategies on PCCI combustion mode as reported in later sections. The main results of selected experimental and some numerical studies were listed in different tables and analyzed to provide an outlook of the suitable strategy to control PCCI mode.

II. COMBUSTION IN CI ENGINE

According to[8], Fig. 1 the combustion phenomenon in CI engines is considered to be taking place in four phases. The four stages are Ignition delay period/Pre-flame combustion, Premixed/rapid combustion, mixing controlled/diffusion combustion, and late combustion. The ignition delay phase is defined as an inactive period between the start of fuel injection to the beginning of combustion [9]. The premixed combustion stage is defined as the period between the beginnings of the combustion to the point of the maximum pressure occurs in a few crank angle degrees. The HRR is maximum during this stage. During the mixing-controlled combustion after the fuel-air mixture built in the ignition delay period has been consumed, the HRR is reliant on the rate mixture which becomes available for burning. While fuel-air mixture requires several physical processes fuel atomization – evaporation – mixing of vapor fuel with air and pre-flame chemical reactions. At the late combustion phase, the unburnt and partially burnt fuel particles left in the combustion start burning as they contact the oxygen. The HRR is minimum during this stage.

As aforementioned, in the CI engine, the NOx and soot emission formation is attributable to the use of a heterogeneous fuel-air mixture[10]. As the CI engine sucks the air alone through the manifold in the suction stroke, then the fuel is injected directly into the combustion chamber at the end of the compression stroke. The injected fuel requires some physical processes (droplet formation - collisions - break up - evaporation) to form a homogenous mixture with air before the start of combustion[11]. The rate of combustion depends on the homogeny of the mixture. As the ignition delay between the fuel injection and the start of combustion is small in the CI engine, so only a fraction of the air and fuel is premixed and burns fast, whereas the remaining fuel requires time for the physical processes than the chemical time scale[12]. Hence, the combustion chamber can be divided into high-temperature zones and fuel-rich zones. In the high-temperature zones, NOx is formed at high rates, and soot is formed in fuel-rich zones.

III. EMISSIONS FORMATION IN COMBUSTION ENGINES

In accordance with the definition, combustion refers to the chemical reaction between hydrocarbons and oxygen to create carbon dioxide, water, and heat. But in practical operation, as oxygen enters with nitrogen to the engine and the incomplete combustion of hydrocarbons, this leads to producing undesirable emissions such as NOx, CO, HC, and soot [13]. The main concern of the more toxic emissions produced by CI engines, which are harmful to human health and the environment. Hence, there is a strong requirement to decrease these emissions as possible[14]. Owing to these aspects, we have to understand deeply the formation of pollutant emissions. Then, we can reduce these emissions by control of the combustion phenomena without the need to use after-treatment devices, to meet the stringent global emission standers.

NOx emission, Nitrogen oxides (NOx) is consists of the sum of nitric oxide (NO) and nitrogen dioxide (NO2), these gases formation is attributable to the interaction between nitrogen and oxygen in the combustion chamber at the in-cylinder local high temperatures as expressed in the equations below. NO2 is a secondary product and is formed by the oxidation of NO in combustion processes. The formation of NOx is a function of the in-cylinder temperature, oxygen concentration, and residence time to complete the reaction[15]. It can be stipulated that to reduce NOx formation, we have to decrease the combustion temperature and make the fuel-air mixture more homogeneous, thus reducing the probability that nitrogen reacts with oxygen atoms.

$$O_2 + \text{heat} \rightarrow O + O$$

$$O + N_2 \rightarrow NO + N$$

$$O_2 + N \rightarrow NO + O$$

Soot formation, soot is not has a clear definition, As is known, solid particles convert to gases when heated, but solid soot forms from gaseous molecules at high temperature. As explained by [16]. Soot formation depends on the properties of the fuel, the structure of the flame, the features of the engine, the combustion temperature, and the operating conditions, under which it is produced but soot can be defined as a solid substance[17].

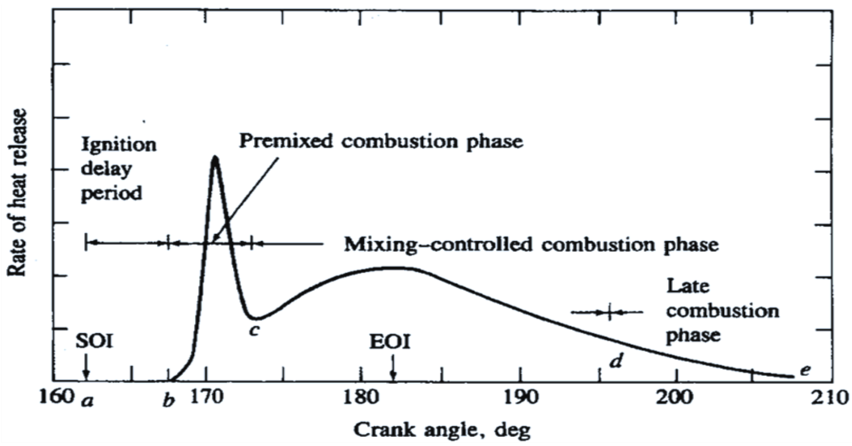

Figure 1. Typical heat release rate diagram identifying different diesel combustion phases

Soot emission is the integrated result of two processes generation and oxidation. Soot generation is formed from the unburned fuel produced through the incomplete combustion of hydrocarbons. Soot results through unique chemical and physical reactions: pyrolysis, nucleation, coalescence, surface growth, and agglomeration, in which vapor-phase hydrocarbons convert into solid soot particles, according to[18], Fig. 2. Soot oxidation is the change of carbon compounds or hydrocarbons to combustion products. The soot oxidation depends on the process and state of the mixture at the time, in other words, when carbon is partially oxidized to CO. These results are preventing the conversion of carbon to soot, even if the carbon is in a rich fuel-air mixture. It has been shown to reduce soot emission, we need to increase the temperature of the late combustion and make the fuel-air mixture more homogeneous which activates the oxidation of carbon.

HC emission, there are many reasons expected to be responsible for the formation of unburnt hydrocarbons. The reasons are (1) Incomplete combustion of over-rich or over-lean mixture in the cylinder due to poor quality of the fuel-air mixture. (2) Fuel droplets in crevice volume due to the higher fuel injection pressure and longer ignition delay period. (3) The In-cylinder wall-wetting due to the flame quenching at the combustion chamber wall. (4) Absorption and desorption of fuel vapor in oil film on cylinder walls and carbon deposits in the chamber [6, 19]. That explains the requirement to increase the amount for the same load. In accordance with the aforementioned to reduce HC emission, we have to make the fuel-air mixture more homogeneous, and select the suitable injection pressure, ignition delay period, and spray angle.

CO emission, Carbon monoxide is a colorless, highly poisonous, odorless, tasteless, flammable gas. However, it is not poisonous, but it prevents the ability to uptake oxygen in the blood, at high concentration it may lead to death, which make it a superior dangerous gas[20]. CO emission is an intermediate combustion product in the burning of hydrocarbon fuels; it results from incomplete combustion. It is produced by the partial oxidation of carbon due to the absence of oxygen through combustion. The conversion process of CO to CO2 is limited by two factors: (1) low combustion temperatures. (2) The lack of oxygen in the reaction zone (over-rich mixtures). Owing to these reasons, the method to reduce Carbon monoxide can be achieved by increasing the quality of combustion to reach the suitable temperature of CO oxidation to CO2. Additionally, make the fuel-air mixture more homogeneous to overcome the lack of oxygen or/and increase A/F so CO finds enough oxygen to convert to CO2.

IV. LOW-TEMPERATURE COMBUSTION:

Nowadays, with increasing concerns about the shortage in fuel sources, increasing environmental pollution, higher energy demand, and stricter emission regulations, these factors are responsible for the requirements of the cleanest and most efficient combustion. So the most important challenge for traditional diesel engines is to meet the world's energy requirements hence, compression ignition engines have to deal with two major challenges improving efficiency and reducing emissions[21]. Meanwhile, a heterogeneous fuel-air mixture is formed before the start of combustion, due to the high viscosity and low volatility of diesel fuel and the fuel-air mixture formation occurs simultaneously.

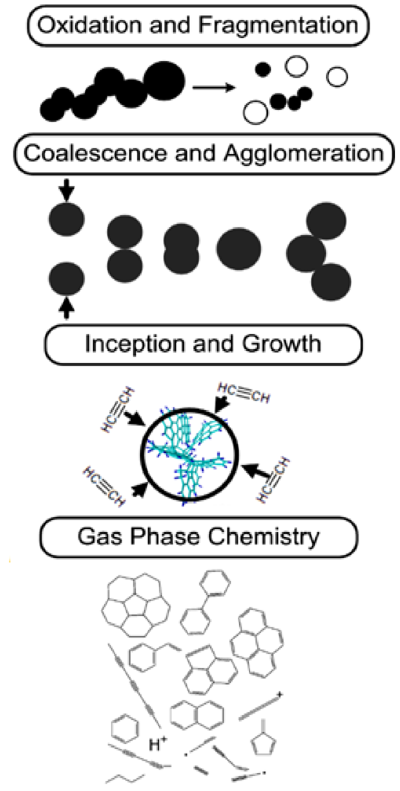

Figure 2. Schematic diagram of soot formation

That leads to high temperature in the cylinder and a high local fuel-to-air equivalence ratio. This is greatly responsible for the higher forms of large amounts of NOx, soot, and pollutant emissions and a waste of useful part of fuel into emissions. However, CI engines have many advantages such as high heat efficiency and low fuel consumption, and easy operation but it produces more toxic emissions which are harmful to human health such as NOx, soot, and PM. Therefore, it is necessary to continue to improve and develop new technologies for CI engines to make them have lower soot and NOx emissions when maintaining high thermal efficiency.

Several improved techniques have been suggested such as after-treatment devices like diesel particulate filter (DPF), selective catalytic reduction (SCR), lean NOx trap (LNT), and diesel oxidation catalyst (DOC). DPF and SCR are used for the soot oxidation and reduction of NOx emissions respectively. But, these devices impose a fuel penalty, require frequent regeneration over a period of time, and increase the total price of the engine, which limits their application for NOx and PM reduction[22]. For these reasons, there is a necessity to focus on different new combustion strategies use of alternative fuels and additives, their blending and composition, new fuel supply system strategies, turbocharger and supercharger, variable valve timing (VVT), exhaust gas recirculation (EGR) and low-temperature combustion (LTC) to reduce the fuel consumption and the formation of exhaust emissions.

Low-temperature strategy (LTC) engines have the ability to reduce NOx and soot emissions to an ultralow level with

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sustaining higher thermal efficiency. LTC strategies are characterized by lower equivalence ratios, the preparation of the fuel-air mixture, and the improvement of fuel atomization. Also, decreases in the peak in-cylinder temperature lead to better energy utilization due to lower radiation losses.

LTC has the flexibility of its fuel (gasoline, mineral diesel, biodiesel, alcohols... etc.)[23]. These LTC advantages make it suitable for the replacement of conventional CI engines. In LTC technology, the in-cylinder temperature is reduced by operating the engine with high EGR or/and operating the engine with an excess air ratio (λ) much higher than 1. However, the control of ignition timing and heat release rate (HRR) are important challenges to achieving LTC technology[24]. LTC strategy can be achieved by various modes like Premixed Charge Compression Ignition (PCCI), Homogenous Charge Compression Ignition (HCCI), Reactivity Controlled Compression Ignition (RCCI), and Stratified Charge Compression Ignition (SCCI). However, the choice of a suitable strategy has shown mixed results in terms of the operating range and emission reduction. Among these strategies used to achieve the LTC, HCCI and PCCI have shown the most promising results, as evident from the Φ-T map[25], Fig. 3.

The combustion phenomenon in LTC mode is considered to be taking place in three phases: (1) Pre-combustion phase depends on charge flow characteristics and changes in species condition. (2) Combustion phase depends on chemical kinetic behavior. (3) Post-combustion phase is affected by chemical and turbulent mixing conditions in as shown in Fig. 4.

The Homogenous Charge Compression Ignition (HCCI) combustion is a combined mixture of SI and CI engine operating cycles. Like SI engines the charge is well mixed producing ultra-low emissions and like CI engines the fuel-air mixture is auto-ignited as a result of compression and has no throttling losses which lead to high thermal efficiency. HCCI can be achieved by premixing the air-fuel mixture before the start of combustion either in the manifold or by early Direct Injection (DI) and the mixture is compressed until the temperature is high enough for auto-ignition to occur near the end of the compression stroke since combustion of charge takes place when the homogenous charge has reached the chemical activation energy spontaneously at multiple sites throughout the charge volume without any diffusion flame and is fully controlled by chemical kinetics[26].

HCCI is governed by three temperatures, at first; we have to reach the auto-ignition temperature of the mixture. Then, the combustion temperature has to increase by at least 1400 K to have high thermal efficiency. Finally, it should not be increased to more than 1800 K to prevent NOx generation. In general, there are many advantages of the HCCI strategy that can be summarized as (1) Higher thermal efficiency due to lower thermal efficiency and lower radiation losses, as combustion uses higher CR, shorter combustion duration, and faster combustion rate that is nearly achieved combustion at constant volume. (2) Saving up to 30% fuel due to using a very lean and dilute mixture and a resulting high specific heat ratio. (3) The ability to operate with low to high octane number fuels like gasoline, diesel, and most alternative fuels. (4) Lower NOx and soot emission due to the absence of flame front, high-temperature regions, and rich mixture. However, this strategy is not without problems and challenges such as (1) High levels of HC and CO emissions. (2) Limited small region of operation due to the occurrence of explosive combustion at high engine loads due to the fast increase of the in-cylinder pressure and misfire at low engine loads. (3) Abnormal pressure noise with noise. (4) The difficulty in combustion phasing control is due to it being difficult to control the ignition timing and combustion rate. (5) A major problem in firing during cold start operation. These challenges make this mode of combustion unsuitable for commercial applications[27].

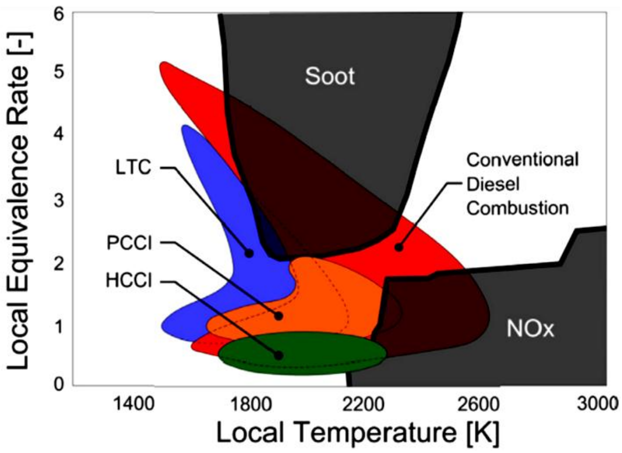

Figure 3. Φ-T diagram of different combustion modes: CI, HCCI, and PCCI, on soot-NOx map

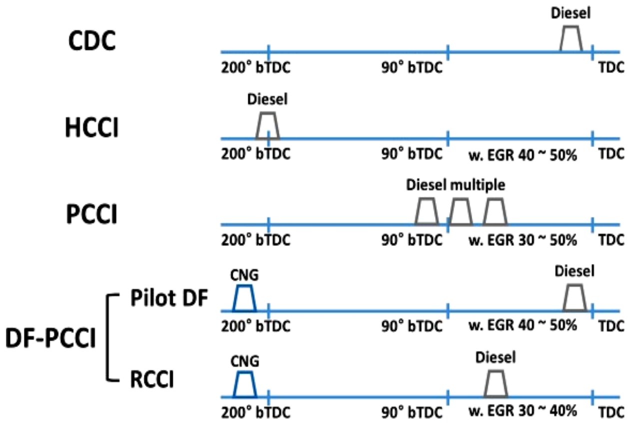

Figure 4. The fuel supply concepts of CI and various advanced combustion technologies

V. Premixed Charge Compression Ignition (PCCI)

Premixed Charge Compression Ignition (PCCI) also known as PPC (partially-premixed compression) or GDCI (gasoline direct-injection compression ignition) has the potential to emerge as a promising combustion strategy that can be used as an alternative to the operational challenges observed in the HCCI strategy. The difference between PCCI and HCCI combustion appears in the air-fuel mixture preparation. As in PCCI, only a part of the fuel is used to provide the homogenous mixture. Hence, the mixing process in PCCI is simpler than in HCCI. Also, the operating range of the PPCI engine is wider than that of the HCCI engine but is still limited at medium loads.

Table 1. Comparison of f basic combustion concepts in CI, SI, and HCCI modes

Mode	SI	CI	HCCI	PCCI
Fuel type	High-octane-like gasoline	High-cetane-like diesel	Flexible fuels	Flexible fuels
air-fuel ratio	Φ ~ 1.0	Φ < 1	Φ ≤ 1	1 ≤ Φ ≤ 2
Ignition type	Spark ignition	Auto-ignition	Auto-ignition	Auto-ignition
Injection strategy	Fuel and air mixed in the Carburetor	injection close to TDC	Early injection	Early injection + TDC injection
Combustion type	Premixed	diffusion	Premixed	Premixed + diffusion
Mechanism control burning rate	Flame propagation speed	injection timing	chemical kinetics	injection timing + chemical kinetics
Thermal efficiency	Low at part load	High	Partially high	Partially high
Emissions	Low	High NOx & soot	Low NOx & soot	Low NOx & soot
Combustion temp.	Low	Partially high	Relatively low	Relatively low

PCCI strategy is a hybrid combustion strategy between HCCI and the traditional CI engines which separates the fuel injection from the start of combustion to provide a sufficiently longer ignition delay leading to enhancing the air-fuel mixture [28]. PCCI partially likes SI engines as the charge is premixed which produces low emissions and like CI engines the fuel-air mixture is auto-ignited as a result of compression which leads to high thermal efficiency under partial-load conditions. So, the PPC strategy has the advantage of both spark ignition and traditional diesel engines with minimal modification of standard engine hardware and software. A comparison of characteristics of SI, CI, HCCI, and PCCI modes is presented in table 1. Initially, in PCCI the premixed charge is obtained by dividing the fuel quantity into two parts [29]. A single-stage direct injection of fuel was used for the small part to prepare the homogenous mixture by advancing its injection timing, while the injection timing of the remaining part was used to control the fuel ignition. But, the more advancing injection timing leads to wall impingements resulting in incomplete combustion with high levels of HC and CO emissions and, consequently, lowers the thermal efficiency so the fuel injection is carried out in multiple or split injection mode.

In PCCI mode, fuel is supplied into the cylinder in three ways, such as port fuel, advanced direct, and late direct injection. [30]suggested that with a suitable selection of injection timing and quantity and EGR conditions for two-stage injection, PCCI can achieve a noticeable improvement in exhaust emissions and performance. Also, a relatively better thermal efficiency with low emission levels and small values of the pressure rise rate can be obtained at high loads, by advancing the first-injection timing and setting the second-injection timing close to TDC. Another study has been done by [31]to understand the combustion and emission characteristics of a diesel PCCI engine with various SOI timings and IVC timings using a full-cycle three-dimensional CFD model coupled with detailed chemical kinetics. They reported that retarding IVC timing and optimized SOI timing produce lower NOx emissions by decreasing the combustion temperature. However, the cases of late IVC timing and early SOI timing increase HC and CO resulting from the serious wall-wetting region due to the low ambient pressure. ISFC is reduced, for the cases with advanced SOI timing around –25 ^{0} CA ATDC and retarded IVC timing, due to using late IVC timing led to delay the ignition timing efficiently TDC which decreased the compression work, as well as SOI timing of –15 ^{0} CA ATDC and early IVC timing, due to the optimization of ignition timing and combustion efficiency. Therefore, the SOI timing and IVC timing must be carefully adjusted. Therefore, the SOI timing and IVC timing must be carefully adjusted.

Additionally, the low volatility of the diesel fuel injected early in the compression stroke, which easily adheres to the cylinder leads to the requirement of high injection pressure with different exhaust gas recirculation (EGR) rates to provide a better air-fuel mixing process, generating sufficient ignition delay to achieve an efficient PCCI combustion. The impact of injection parameters and EGR on the combustion characteristics and exhaust emissions of a PCCI diesel engine with late and early injection timings have been studied [32]. Higher injection pressure led to better indicated thermal efficiency and IMEP with late and early injection timings, as faster combustion leads to enhance heat release rate. Additionally, the higher injection pressure led to lower NOx, smoke, and HC emissions, while CO emissions remained relatively unchanged. The use of EGR increased indicated thermal efficiency and IMEP and reduced soot and NOx emissions but led to higher HC and CO emissions. The injection timing = 20◦ BTDC with EGR was the optimum injection timing that gave simultaneously low soot and NOx emissions, due to the better fuel-air mixing as the spray struck the piston bowl wall at an optimum targeting spot. The effects of fumigated ethanol and EGR on PCCI performance combustion and emissions characteristics were studied also by [33] and compared with pure diesel combustion. The engine was operated with different combinations of ethanol, diesel, and EGR at different loads. They reported that BTE had shown the maximum value at 15% ethanol fumigation. NOx emissions and smoke opacity were reduced at 15% ethanol fumigation compared to pure diesel combustion. The combination of ethanol fumigation and EGR produced has shown higher HC and CO emissions.

Due to knock, misfire, and control of ignition is not easy, the operating range is limited to medium load range only and cannot be practically applied at higher engine load range. Many studies have been carried out to extend the load range by the development of the mode switching concept, where the engine operates in PCCI and CDC modes to switch between them without deterioration in performance and exhaust emissions. Likewise, a work done by [34] using a mode switching between CDC and PCCI combustion modes using an open electronic control unit (ECU) which is programmed to operate the engine in PCCI combustion mode up to medium engine loads and then automatically switch it to CDC mode at higher engine loads. In this study, a two-cylinder, and four-stroke, direct injection fueled by mineral diesel and B20 was used to investigate the performance and emissions characteristics of the mode switching. For both the test fuels PCCI combustion mode showed significantly lower particulate and NOx emissions but slightly higher CO and HC emissions compared to CDC. In CDC the results showed significantly lower CO emission and particulate emissions for B20. Conversely, slightly lower NOx emissions appeared for mineral diesel. The results showed that blending of biodiesel in mineral diesel has similar engine performance but it improved the engine emission characteristics significantly in both modes. This study shows that it is suitable to employ PCCI combustion up to medium loads and conventional CDC at higher loads.

VI. Influence of injection strategy on PCCI engines

Fuel injection strategies (injection timing - number of injections - preheating manifold injection) play a significant role in emission characteristics, in-cylinder combustion process, and engine performance, by affecting fuel atomization, diffusion, evaporation, and fuel-air mixing. The injection timing is defined as the position at which fuel is supplied into the engine and is denoted with respect to TDC. The pilot injection timing strategy is one of the main factors to control the combustion process and the smoothness of the heat release rate, it can be classified into advancing and retarding injection relative to the piston position. The impacts of pilot injection timings have been mapped by many researchers. They concluded that advancing the injection timing can increase ignition delay (the fuel-air mixing time), which is good for the formation of a more homogeneous mixture. The emission of HC and CO increased because of the incomplete combustion as the earlier injection timing leads to lower in-cylinder temperature and pressure, poor fuel evaporation, and wall-wetting, and hence the combustion efficiency has been reduced. While NOx and soot emissions will be declined due to low flame temperatures. Likewise, a work done by [35] studied the impact of pilot injection timings between− 40 and − 24 ◦CA (ATDC) and pilot injection ratio between 0 and 20% on the combustion process, emission characteristics, and fuel economy. The results showed that in PCCI with a pilot injection strategy the BTE is higher with a trade-off of increasing NOx and soot emissions. With retarding pilot injection timing, the heat release peak value of the pilot injection increased, while the main injection shows the opposite trend. The heat release rate curve changed from a single-peak shape to a double-peak shape. NOx and soot emissions increase, while HC and CO emissions decrease. This is because the ignition delay is shortened and the in-cylinder pressure and temperature increase. Additionally, the mixing time of fuel and air declined, and the proportion of premixed combustion declines. With the increase of pilot injection ratio, the peak in-cylinder pressure increases, the ignition delay is shortened and the combustion duration is prolonged. The BTE increases, with the increase of the pilot injection ratio from 0 to 5%, and then BTE decreased when the pilot injection ratio increases. HC, CO, and HCHO emissions increase, this is because more fuel enters the piston crevice. NOx emissions remain almost unchanged.

To further stress another study done by [36] studied the impact of different injection timings at different engine loads on the performance and exhaust emissions of a single-cylinder, direct injection diesel engine when methanol-blended diesel fuel was used. They found that with advancing injection timing, NOx and CO2 emissions increased. Meanwhile, CO and UHC decreased. Advancing injection timing caused an earlier start of combustion relative to the TDC which results in compressing the cylinder charge as the piston moves to the TDC and improving the reaction between fuel and oxygen and hence increasing the maximum cylinder temperature. As a result, NOx emissions start to increase. With advancing injection timing, the ID will be longer and the flame speed will be shorter. These reduce the maximum pressure and output power of the engine. Hence, fuel consumption per output power will increase. On the other hand, retarding injection timing causes later combustion, and hence pressure rises only when the cylinder expands rapidly and reduces the effective pressure to do work. It has been shown that the minimum Brake specific fuel consumption (BSFC) was obtained at the original injection timing for all the fuel blends. The best results of Brake thermal efficiency (BTE) were obtained at the original injection timing. Advanced or retarded advanced injection timing declined BTE values.

To understand the impact of the fluid flow effects on PCCI, [37] explored the effects of different injection timings between -60 and -15 CAD (ATDC) of two different injectors with 5 and 7 holes on fluid flow characteristics of PPC from injection to the end of combustion using high-speed PIV measurements that were performed in a light-duty optical engine to detect the combustion performance. The results revealed that later injection leads to higher mean velocity and turbulence level inside the piston bowl and hence higher mixing efficiency. This behavior is probably attributable to the geometry and the position of the piston bowl relative to the spray targeting. It also affirmed that the 7-hole injector leads to sharper heat release rates, better mixing, higher combustion-driven turbulence, and higher combustion efficiency than the 5-hole injector. Additionally, It was noted that the injection timing of -60 (ATDC) has lower combustion efficiency attributable to a large quantity of fuel trapped in the crevice volumes.

The injection strategy can be classified according to timing and types can be classified into single, double (pilot in advance and main injection close to TDC), and triple injection strategy, the first injection is called pilot injection (injected early before the main injection), the second injection is called the main injection (the one using the largest fuel injection amount and the longest duration angle of injection, and the third injection is also called the post-injection (injected late after the main injection) when the pilot injection is divided into several smaller injections, as shown in Fig. 5. Numerous studies have been done to reveal the effect of the injection quantity-ratios and advanced / retarded fuel injection timing classified based on single, double, and triple injection on performance and exhaust emission characteristics. In this regard, [38] explored the effects of multiple injection strategies on combustion, performance, and exhaust emission characteristics. The experiments were carried out on a medium-duty, single-cylinder diesel engine, with a compression ratio of 18 and with a constant speed of 1500 rpm. The injection timing for single injection strategy was [-25 ◦CA (ATDC) with 100% mass], double injection strategy was [-40 ◦CA (ATDC) and-18 ◦CA (ATDC) with 40% and 60% mass], and triple injection strategy was [-31 ◦CA, -21 ◦CA and -11 ◦CA (ATDC) with 30%, 30% and 40% masses]. The results were compared with a double injection strategy at 12 kg loading conditions. The results revealed that, Compared to the double injection strategy, the single and triple injection strategies had a 67.5% and 41.78% rise in NOx emission respectively. Meanwhile, the single and the triple injection strategy had a 6% reduction and a 25% rise in HC emission respectively. In comparison with the double injection strategy,25 % rise and 8% reduction in CO emission for single and triple injection strategies respectively, due to an increase in charge temperature and a faster oxidation rate. Additionally, 2% rise and a 15.78% reduction in smoke opacity for single and triple injection strategies respectively. The single injection method has the maximum thermal efficiency, which is slightly higher than the double and triple injection strategies.

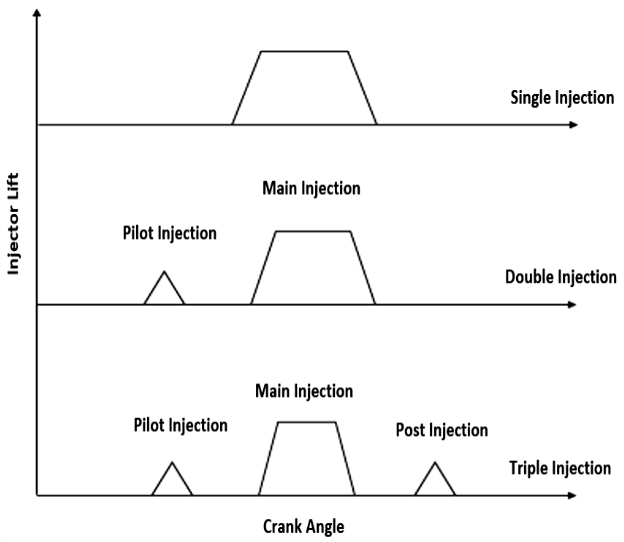

Figure 5. Injection profile for different injection strategies

The experiments were performed on a light-duty 2.2 L four-cylinder compression ignition engine fuelled with G50-Dieseline by [39] they studied the PCCI engine combustion and emission characteristics and compared it with the single-injection strategy characteristics. A two-stage split injection has been used to control the injection quantity ratios and timings. The experiments were performed in two different load groups, 1.37 and 2.97 bar BMEP. For both loads, variations in BTE, CO, THC, and NOx were affected by the first injection timing. BTE increased as the first injection timing was retarded. Less fuel injected leads to less possibility of over-mixing of the air and fuel. CO and THC increased at very early first-injection timings due to wall impingement and wall wetting. Additionally, over-mixing can result in lower combustion temperatures. NOx increased as the first injection timing was advanced, due to an increase in the injected fuel quantity.

The combustion phase was mostly affected by the second injection timing. Smoke variations were dominated by the first and second injection timings. Most of the smoke emitted from the engine may be generated from the second injection part since there is enough time for the first injection part to be mixed well with air. The two-stage split injection produced a decline in NOx and smoke around 50% and 90% respectively compared to the single-injection strategy. On the other contrary, the penalty for using the two-stage split-injection strategy was that the Brake-thermal efficiency decreased due to the increase in THC.

Another simulation study was done by [40] to understand the effects of fuel splitting proportion, injection timing, spray angles, and injection velocity on combustion and emission characteristics. CFD was investigated in a high-speed direct-injection diesel engine by CFD simulation using KIVA-3V code coupled with detailed chemistry. They reported that, for the cases when the main injection is retarded after the start of combustion in the first injection, as the proportion of fuel in the first injection increased it means the mixture becomes more homogeneous and less fuel is left to be oxidized in the second heat release phase. This would result in lower combustion temperature, and thus lower NOx formation. On the contrary, both CO and HC emissions increased with increasing the proportion of fuel in the first injection and when the start of the main injection was retarded. Higher NOx emissions were obtained by increasing the injection velocity because more intensive heat release took place in the cylinder. And higher soot emissions increased for higher injection velocity and with late main injection. Meanwhile, HC emissions did not change much with varying injection velocity. A large spray angle would achieve low soot, CO, and HC emissions, but show some disadvantages in NOx control.

The preheating manifold injection is an effective strategy. Which, Liquid fuel is heated up to its vaporization temperature and sent to the intake manifold to mix with fresh air, forming an external mixture that enters the combustion chamber. Many studies have been implemented to reveal the effect of the manifold injection on the performance and emission characteristics of the PCCI engine. For instance, [41] reported the impact of Hemispherical Combustion Chamber (HCC) geometry and Toroidal Combustion Chamber (TCC) geometry on the PCCI engine characteristics.

A proper fuel vaporizer technology was developed and modified in a single-cylinder DI diesel engine. Which provided an external mixture that enters the combustion chamber as; the fresh air mixes with vaporized fuel. They used different vapor induction rates (2 ml/min and 4 ml/min). The results showed that the PCCI engine has a high possibility of using it in CI engines. It provides the solution for the main danger of the diesel engine, such as NOx and smoke emissions. The decline in NOx and smoke emissions were coupled with decreasing in Brake thermal efficiency (BTE). In PCCI mode, the use of a higher fuel vapor induction leads to lower NOx emissions compared to the low fuel vapor quantity. Additionally, at a rate of 2 ml/min of fuel vapor, the TCC geometry has shown a decline in HC and CO emissions by 8.7 % and 9.5 % vise an increase in NOx and BTE by 5.31% and 6.59% respectively compared to HCC geometry.

Another study done by [42] Studied the effects of manifold injection at various concentrations of n-decanol like 10%, 20%, and 30% as a secondary injection with a primary fuel of NB20 (80% diesel and 20% biodiesel) on the engine characteristics. The obtained results have compared the results with the only injection of diesel and NB20 fuel. The results revealed that for all manifold injections the peak pressure and the rate of heat release rate are improved than diesel and NB20 fuel due to the availability of intrinsic oxygen in n-decanol. The NB20DI30 has the highest in-cylinder pressure due to its superior combustion. NOX emission was reduced for all manifold injections compared to both diesel and NB20 fuel. This is due to the low-temperature combustion achieved because n-decanol has a higher latent heat of vaporization which reduces the in-cylinder temperature.

The HC and CO emissions were increased owing to the low-temperature combustion. Additionally, a significant quantity of homogeneous charge escapes without burning as it enters the crevice volume. The increase in n-decanol proportion resulted in lesser HC emission due to the availability of intrinsic oxygen in it (better combustion). For the n-decanol proportion, BTE increases from 10% to 30%, and the BTE was improved owing to its better combustion. Diesel fuel has higher BTE compared to other tested fuels due to its greater calorific value.

Table 2 represents the summary of the effect of spray angle and fuel injection timings and ratios classified based on single, double, and triple injection on combustion characteristics and emission of the PCCI engines. It has been shown that soot, and NOx emissions decreased with advancing the injection pattern timings due to low flame temperature and improvement of fuel-air mixing. However, HC and CO emissions increased due to wall impingement and wall wetting. Increasing the number of Pilot injections. results in increasing in-cylinder pressure and combustion temperature which leads to an increase in soot, and NOx emissions and reduces HC and CO emissions. Owing to these aspects, we have to find out the appropriate fuel injection timings and ratios, which decrease soot and NOx emissions without increasing HC and CO.

VII. INFLUENCE OF FUEL PROPERTIES ON PCCI ENGINES:

The quality of fuel has a direct impact on ignition delay, heat release rate, pressure, and temperature profile of the engine. This simultaneously affects the combustion, performance, and emission characteristics of the PCCI engine. Fuel ignition quality is affected by numerous factors such as physical properties (density, viscosity, etc.), thermal properties (calorific value, latent heat of evaporation, auto-ignition temperature, etc.), the composition of organic molecules (carbon, hydrogen, oxygen, etc.) and ignition properties (ON, CN) [43]. Density plays a vital role in spray pattern and sprays tip penetration (STP), lower density fuel leads to a wide spray pattern and shorter STP due to the lower droplet's momentum.

Fuel spray penetration and atomization depend on the fuel viscosity, lower viscosity fuel better spray formation and atomization due to the formation of finer fuel droplets which make the fuel injection system easy to discharge the fuel into the cylinder. The calorific value of the fuel increases with the fuels having more carbon and hydrogen molecules, so fossil fuels have higher calorific value compared to renewable fuels. Higher latent heat of vaporization fuels requires higher thermal energy to transform the liquid substance into a vapor/gas; therefore, it can be used only after some engine modifications like external heating devices, higher CR, thermal insulations, and fuel blends. Higher ON fuels such as ethanol and alcohol have higher auto-ignition temperatures. Therefore, they have been considered a suitable blend for the extension in ID that enhances the premixing and spray-driven combustion but could negatively affect the diffusion phase of the combustion. Owing to these aspects, the selection of the fuel type play an important role in enhancing the engine performance and characteristics and so many studies have been done to explore the effects of using different fuels at various condition to increase thermal efficiency and reduce exhaust emissions.

M. Nibin, et.al. [55] investigate the effects of fuel on PCCI combustion, performance, and emission characteristics in a single-cylinder engine at 1500 RPM with neat wheat germ oil (WGO) and various energy share of bioethanol with wheat germ oil and compared the results with diesel fuel. Bioethanol is injected in the inlet port with varying the injected percentage (10, 20, and 30%); subsequently, wheat germ oil is injected directly inside the combustion chamber. The results exhibited that the fuel (WGO) with higher density and viscosity and low volatility which results in poor atomization hence, the mixing of air and fuel is slow resulting in slower heat release, thereby increasing the ignition delay and increasing the exhaust gas temperature. A lower calorific value is resulting in lower peak pressure. These combustion conditions lead to the lowest brake thermal efficiency of WGO at all loads. Additionally, these combustion conditions can help to lower the production of NOx emissions but lead to higher HC, and CO emissions. The early injection of bioethanol provides a more homogenous charge due to its favorite properties before the injection of primary fuel WGO.

The combustion is initiated when the primary fuel is injected at the end of the compression process and the higher flame speed of bioethanol leads to a longer premixed combustion duration which in turn decreases the diffusion combustion duration. The results showed that with the addition of bioethanol, BTE increases as the premixed combustion improves (more heat is released near to TDC which in turn converts more work output) due to the high flame speed of bioethanol, the ignition delay period decreases and combustion duration due to the high latent heat of vaporization of bioethanol. Higher heat release and peak pressure as the viscosity of bioethanol is very low and volatile which helps bioethanol to mix well with air and make a homogenous mixture during the intake and compression process. Besides, with the addition of bioethanol due to these reasons lower HC, CO, NOx, and smoke emissions are produced.

Table 2. Effect fuel injection timings and ratios on PCCI engine characteristics.

Author	Research engine & Fuel type	Injection method	Test condition	Significant remarks on performance and emissions
Kong et al. [44]	Marine engine, Cylinder = 8 CR = 14.5 & Diesel DI fuel	Pilot inj. (°CA BTDC) = 36.5, 32.5, 28.5, 24.5, and 20.5 , The main inj. is injected from - 14.5 to 20∘CA ATDC post inj. = 22 °CA ATDC	RPM= 1350 , Injector hole = 8 Nozzle hole diameter = 0.337 mm Spray angle = 145∘, 150∘, 155∘, and 160∘ P_{inj}. = 87 MPa	↑ NO_X, ↓ Soot with retarding pilot injection timing ↑ NO_X, ↑ Soot with increasing post-injection ratio
Yingying Lu et al. [45]	Heavy-duty engine, Cylinder = 6 CR = 17 & Diesel fuel	Multiple injection timings (°CA BTDC) (90, 75, 60, 45) , (80, 65, 50, 35) , (70, 55, 40, 25)	RPM= 1600 , Injector hole= 8 diameter = 0.217 mm P_{inj}. = 160 MPa	↑ ID, ↓ BTE, ↑ HC, ↑ CO with advancing injection. timing NO_X lowest at − 90 ∘ CA ATDC Soot lowest at − 80 ∘ CA ATDC
Liu et al. [46]	Diesel engine, Cylinder = 4 CR = 17.5 & Diesel and methanol fuel	Pilot inj. = − 26 °CA ATDC Main inj. = − 10 °CA ATDC	RPM = 1800 engine load = 30% Intake air temp. (∘C) = 40	↑ ID, ↑ BTE, ↑ NO_X, ↓ HC, ↓ CO, ↓ Soot with advancing the main inj. timing ↑ NO_X, ↓ HC, ↓ CO, ↑ Soot with increasing pilot inj. quantity ↑ HC, ↓ Soot with advancing pilot inj. timing
Jeong et al. [47]	Diesel engine, Cylinder = 4 CR = 17.3 & ULSD fuel	Single inj. = −5 (°CA ATDC) Double inj. (°CA ATDC) 1st Inj. = −70 ~ −20 2nd Inj. = −5, 5	RPM = 1500 engine load = 50 Nm P_{inj}. = 50 ~ 90 MPa	↑ NO_X, ↑ HC, ↑ CO with advancing first inj. timing lower NO_X, higher Soot at the double injection ↑ NO_X, ↓ Soot with increasing inj. pressure
Mei et al. [48]	Diesel engine, Cylinder = 4 CR = 15.88 & Diesel fuel	1st pilot Inj. (°CA ATDC) = −38 ~ −28 2nd pilot Inj. (°CA ATDC) = −24 ~ −14 Main inj. (°CA ATDC) = −6	RPM= 4000 , Injector orifices = 7 Spray angle = 153°, Intake air temp. = 94∘C EGR = 40% P_{inj}. = 87 MPa	↓ NO_X, ↓ Soot with advancing SOI−P1 timing ↓ ID, ↑ NO_X, ↓ Soot with advancing SOI−P2 timing
Molina et al. [49]	Medium-duty engine, Cylinder = 4 Vd = 5100 cc, CR = 17.5 & Diesel fuel	1st pilot Inj. (°CA BTDC) = 57.5 ~ 45 2nd pilot Inj. (°CA BTDC) = 55 ~ 42.5 Main inj. (°CA BTDC) = 55 ~ − 5	RPM= 1000 , Injector hole = 7 Nozzle hole diameter = 0.177 mm Spray angle = 150∘ EGR = 40% P_{inj}. = 100 ~ 150 MPa IMEP = 3.8 bar	↓ NO_X, ↑ HC, ↑ CO with advancing the inj. pattern timings ↓ NO_X, ↑ HC, ↑ CO with increasing the main inj. fuel mass
Jain et al. [50]	Diesel engine, Cylinder = 1 Vd = 510.7 cc, CR = 17.5 & Diesel fuel	Pilot inj. (°CA BTDC) = (30, 35 & 40) Main inj. (°CA BTDC) = 12 ~ 24	RPM= 4200, Max. power = 6.25 kW EGR = 15 % Pinj. = 40, 70 and 100 MPa	↑ ID, ↓ BTE, ↑ NO_X, ↓ Soot with advancing the main inj. timing ↓ ID, ↓ BTE, ↑ HC, ↓ Soot with increasing FIP Highest NO_X, lowest CO at 40 MPa
Dempsey et al. [51]	Light-duty diesel engine, Cylinder = 4 Vd = 1.9 L CR = 15.1 & low octane gasoline fuel (RON = 68)	Pilot inj. (°CA BTDC) = 91 ~ 324 Pilot inj. ratio = 65% Main inj. (°CA BTDC) = 30 ~ 40	RPM= 2000 , Injector hole = 7, Nozzle hole diameter = 0.14 mm, Spray angle = 148° Pinj. = 40, 55 and 70 MPa IMEP = 4 bar	↑ BTE, ↓ NO_X, ↓ HC with advancing pilot inj. timing ↑ BTE, ↓ NO_X, with increasing FIP
Singh et al. [52]	Diesel engine, Cylinder = 1 Vd = 510.7 cc, CR = 17.1 & B20, B40, and mineral diesel	1st pilot Inj. (°CA BTDC) = 45 2nd pilot Inj. (°CA BTDC) = 35 Main inj. (°CA BTDC) = 12, 16, 20, and 24	RPM= 1500 Pinj. = 40, 70 and 100 MPa EGR = 15 % IMEP = 3 bar	↓ ID, ↓ BTE, ↑ NO_X, ↑ HC, ↑ CO with advancing the main inj. timing ↓ ID, ↑ HC with increasing FIP
Ishiyama et al. [53]	DI diesel engine, Cylinder = 1 V_d = 857 cc CR = 17.8 &	1st stage inj. (°CA BTDC) = 25 ~ 45 2nd stage inj. (°CA BTDC) = 0 ~ 24 Q_1/Q_2 = 2/3, 3/2 and 4/1	RPM= 1200 Pinj. = 80 MPa Injector hole = 6 Nozzle hole diameter = 0.12 mm	↓ NO_X, ↑ CO with advancing 1st inj. timing ↑ BTE, ↓ NO_X with larger first injection quantity at 1st stage inj. = 40∘ BTDC

	Diesel and natural gas		Spray angle = 140° Intake air temp. (◦C) = 40	
Kook et al. [54]	DI diesel engine Cylinder = 1 Vd = 498 cc CR = 18.9, 23 and 27.7 & Diesel	1st stage inj. (˚CA BTDC) = 250, 200, 150, 100 and 50 2nd stage inj. (˚CA BTDC) = 20, 15, 10 and 0	RPM= 800 Pinj. = 30, 120 MPa Injector hole = 5 Nozzle hole diameter = 0.168 mm Spray angle = 150, 100°	↓ NO_X, ↓ CO, ↓ Soot with very early 1st timing ↑ BTE, ↑ HC, ↑ CO, ↑ Soot with higher CR ↓ HC, ↓ CO, ↓ Soot at injection angle 100◦

Elumalai et al. [56] examined the effect of high-volatile n-pentanol percentage on the performance, combustion, and emission characteristics of PCCI engines at 1500 rpm and 17.5 CR. The primary fuel used in the engine was a blend of 80% diesel fuel and 20% waste tire pyrolysis oil with injection timing of 35◦ BTDC, while the secondary fuel n-pentanol was sprayed into the intake manifold with varying percentages, namely 10%, 20%, and 30%. Moreover, 50 ppm of CuO/ZnO (CuZnO) nanoparticles were added to the fuel blend to provide more oxygen which led to better combustion. The results showed that the addition of n-pentanol increased the BTE but was still lower than the diesel fuel and the operating range was limited at 80% load condition due to the appearance of irregular combustion and noisy operation which led to the supply of lean mixture. The addition of n-pentanol helped decline NOx and smoke emissions but led to an increment in CO and HC emissions. It has been shown that using CuZnO nanoparticles results in a similar performance to diesel. Furthermore, the lowest induction rate of n-pentanol provided a significant increase in BTE and a remarkable reduction in NOx and smoke emission as compared to the waste tire pyrolysis oil blend.

The experiments were performed on a single-cylinder diesel engine at 1500 rpm and 17 CR. By Elkelawy et al. [13]. They studied the impact of using blends of biodiesel and diesel fuel to formulate two blends by volume of 20% biodiesel 80% diesel (B20D80) and the other is 40% biodiesel 60% diesel (B40D60) on engine combustion and emission characteristics. While, the premixed fuel blends that vaporized are adjusted at varying percentages, namely, 20%, 25%, and 30%. The results show that the B40-D60 blend of the BTE increased from 19.34% to 29.91%. The CO and HC emissions have the lowest values using B40-D60 due to the adjustable timing of injection for the intake manifold charge and the high oxygen content for B40-D60. NOx and smoke opacity emissions have the lowest values using B40-D60. Hence, the blend with a higher biodiesel ratio recorded a higher BTE and a remarkable reduction in CO, HC, NOx, and smoke emissions.

Hildingsson et al. [57] examined the effects of fuel octane number on a single-cylinder light-duty CI engine at a low load and speed (4 bar IMEP / 1200 rpm) with no EGR as well as at a higher load and speeds (10 bar IMEP / 2000 and 3000 rpm) with EGR. The experiments have been run on four gasoline fuels of RON of 72, 78, 84, and 91 and typical European diesel fuel with a fixed intake temperature of 60°C. They reported that at low load smoke was low for all fuels. Compared to diesel fuel, the high RON gasoline fuels give much lower NOx as the ignition delay is slightly higher which leads to better air-fuel mixing. At high loads, all the fuels showed low NOx below 0.3 g/kWh with sufficient EGR. Meanwhile, smoke was very low for gasoline fuels but high for diesel fuels. CO and HC were higher for gasoline fuels due to incomplete combustion because of lean packets burning at low temperatures, while HC and CO could be decreased for gasoline fuels by reducing the injection pressure. The experiments revealed that small injector holes and high injection pressures are required to overcome the low ignition delay of diesel and provide better mixing rates. Meanwhile, the judicious RON range could be between 75 and 85 with larger injector hole diameters and lower injection pressures.

Another work of [58] examined the impact of RON on the engine performance using three naphtha fuels, with 60, 70, and 80 RON and E10 gasoline "RON 91" was used as the baseline test fuel using a single-cylinder engine running at 15.1 CR. The results showed that at 800 RPM and low loads, the RON 80 naphtha has the lowest SFC and highest ITE of all the fuels due to good combustion phasing and duration, low unburned fuel, and low heat transfer. At 1500 RPM and medium loads, as RON decreased from 80 to 60, higher thermal efficiencies were observed and so RON 60 naphtha exhibited higher ITE similar to E10 gasoline. While, high loads are limited for the naphtha fuels (RON 80 at 14.2 bar IMEP, RON 70 at 11 bar IMEP, RON 60 at 7.6 bar IMEP) due to the requirement of higher EGR rates, which replaced inducted fresh air. So, only E10 gasoline was able to operate at 15 bar IMEP as it has very low SFC and high ITE.

Table 3 represents the summary of the effect of fuel properties on combustion, performance, and emissions characteristics of PCCI engines. Higher octane number fuels like gasoline, ethanol, butanol ...etc. leads to better air-fuel mixing due to longer ignition delay which results in lower soot emissions and lowers NOX due to lower combustion temperature as a result of lower energy content and higher latent heat of vaporization but higher CO and HC due to incomplete combustion. Therefore, lower injection, larger injector holes, and early injection timing might be required for lower reactivity fuels to provide better combustion and lower CO and HC.

While higher viscosity and density fuels like diesel, biodiesel …etc. Required higher injection pressure to provide better atomization and late injection of the pilot with a lower fuel ratio required to reduce the fuel trapping into the crevices. Also, a higher EGR ratio is required for better air-fuel mixing.

Table 3. Effect fuel properties on PCCI engine characteristics.

Author	Research engine	Fuel type	Test condition	Significant remarks on performance and emissions
Singh et al. [52]	Diesel engine, Cylinder = 1 Vd = 510.7 cc, CR = 17.1, IMEP = 3 bar	B20 B40 Mineral diesel	RPM= 1500 , $P_{inj.}$ = 40, 70 and 100 MPa EGR = 15 % 1st pilot Inj. (∘CA BTDC) = 45 2nd pilot Inj. (∘CA BTDC) = 35 Main inj. (∘CA BTDC) = 12, 16, 20, and 24	↑ ID, ↑ NO_X, ↓ HC, ↑ CO at B20 BTE max for B40 at 40 MPa
Viollet et al. [59]	CI engine, Cylinder = 1 V_d = 499 cc CR = 12:14	RON = 40, 68, 93 MON = 40, 63, 84	RPM= 1000~3000 $P_{inj.}$ = 4, 15 MPa Max. load = 10 bar	↑ NO_X, ↓ HC, ↓ CO for RON 40 at 14 CR
Hildingsson et al. [60]	CI engine Cylinder = 1 V_d = 537 cc CR = 16	CN = 56.2, 39.6, 28.6, 24, 22.5, 19, 17	RPM= 1200, 2000, 3000 Injector hole = 7 Nozzle hole diameter = 0.13 mm Spray angle = 153∘ IMEP = 4, 10 bar	↑ ID, ↓ NO_X, ↓ HC, ↓ CO at low CN BTE min for CN 17
Cho et al. [61]	CI engine, Cylinder = 1, V_d = 1.8 L, CR = 15	RON = 80.3, 80.0, 80.0 MON= 77.9, 74.9, 77.5	RPM = 800 with IMEP = 2 bar $P_{inj.}$ = 110 MPa & RPM = 1500 with IMEP = 6 bar $P_{inj.}$ = 170 MPa 1st Inj. (∘CA BTDC) = 45 2nd Inj. (∘CA BTDC) = 35	↓ ID, ↓ NO_X, ↓ CO with MON 74.9 ↓ soot for MON 77.5
Elzahaby et al. [62]	Diesel engine, Cylinder = 1, V_d = 825 cc, CR = 17	Diesel CN 55.6, Ethanol CN 11, ethanol-diesel blends of 0, 10, 20, 30,40 and 50%	RPM= 1500 $P_{inj.}$ = 17.5, 25 MPa Power = 6 kw	↑ BTE, ↓ NO_X, ↑ HC, ↑ CO with the addition of ethanol at a low load ↑ BTE, ↓ NO_X, ↓ HC, ↓ CO with the addition of ethanol at a high load
Zhang et al. [63]	light-duty CI engine, Cylinder = 4, V_d = 2198 cc, CR = 16.6	Gasoline-diesel blends of 0, 20, and 50% CN = 51, 45, 35	RPM = 1800 IMEP = 7.85 bar $P_{inj.}$ = 100 MPa	↓ NO_X, ↑ HC, ↑ CO, ↓ Soot with the addition of gasoline at a low load ↑ NO_X, ↓ HC, ↓ CO, ↓ Soot with the addition of gasoline at a high load
Valentino et al. [64]	Light-duty diesel engine, Cylinder = 1, CR = 10.1	n-butanol-diesel blends of 0, 20, and 40% CN = 52, 44, 36	RPM = 500 Injector hole = 7 Nozzle hole diameter = 0.136 mm Spray angle = 148∘ $P_{inj.}$ = 80 MPa EGR = 0, 50% pilot Inj. (∘CA BTDC) = -5, -2, 1, 4, 7 Main inj. (∘CA BTDC) = 2, 5, 8, 11, 14	↑ ID, ↓ NO_X, ↑ HC, ↓ Soot with the addition of n-butanol
Gowtham et al. [65]	Light-duty diesel engine, Cylinder = 1, CR = 17.5	Diesel, Biodiesel, Ethanol, Ethanol-biodiesel blends of 10, 20, and 30%	RPM = 1500 Rated power = 5.20 kW $P_{inj.}$ = 20 MPa	↓ NO_X, ↑ HC, ↑ CO, ↓ Soot with the addition of ethanol BTE min for B90+E10
Gupta and Krishnasam [66]	Light-duty diesel engine Cylinder = 1 CR = 15 V_d = 662 cc	Diesel Gasoline Butanol	RPM = 1500 Injector hole = 7 IMEP = 1.06 to 4.25 bar $P_{inj.}$ = 30 MPa EGR = 0 to 60%	↑ ID, ↓ NO_X, ↓ HC, ↑ CO, ↓ Soot with blending gasoline or butanol HC min for butanol BTE max for B80+E20
Kaya et al. [67]	Diesel engine Cylinder = 1 CR = 16 Vd = 1498 cc	Diesel B0 Biodiesel B100 B30	RPM = 2000 Power = 0 to 5 kW Pinj. = 65 MPa EGR = 22 to 60 %	↓ ID, ↓ BTE, ↑ NO_X, ↓ and Soot with the addition of biodiesel
Cardone et al. [68]	CI engine Cylinder = 1 CR = 16.5 Vd = 477 cc	Diesel-LPG blends with 20% and 35% by mass	RPM= 2000 Injector hole = 7 Nozzle hole diameter = 0.141 mm Spray angle = 148∘ Pinj. = 75 and 85 MPa IMEP = 2 and 5 bar	↑ ID, ↑ NO_X, ↓ HC, ↓ CO with increasing the LPG fraction Soot has a 95% reduction with the use of LPG

Benajes et al. [69]	Light-duty diesel engine Cylinder = 4 Vd = 1.6 L CR = 18	Gasoline-diesel blends of 0, 25, and 50% CN = 51.2, 36.6, 28.3	RPM= 1500 Intake air temp. (◦C) = 45 Injector hole = 6 Nozzle hole diameter = 0.124 mm Spray angle = 150◦ Pinj. = 80 MPa EGR = 40% IMEP = 3 bar	↓ NO_X, ↓ Soot with the addition of gasoline
Ogawa et al. [70]	Light-duty diesel engine Cylinder = 1 Vd = 1.9 L CR = 14	Diesel N-hexane CN = 50, 50 Boiling point (◦C) = 68.7, 289	RPM= 1200 Injector hole = 4 Nozzle hole diameter = 0.21 mm Spray angle = 40◦ Pinj. = 40, 60, 80, 100 and 120 MPa	↑ BTE, ↑ NO_X, ↓ Soot with n-hexane ↑ NO_X, ↓ HC, ↓ CO with increasing inj. pressure

VIII. SUMMARY AND CONCLUSION

This critical review has focused on the literature related to combustion, performance, and emissions characteristics control by premixed charge compression ignition (PCCI). PCCI is one of the most promising LTC strategies that can help meet goals for reducing NOx and soot emissions and providing higher thermal efficiency through leaner combustion at lower temperatures. PCCI has the advantage of both SI engines as the charge is premixed which produces low emissions and like CI engines as the fuel-air mixture is auto-ignited as a result of compression which leads to high thermal efficiency under partial-load conditions. As in PCCI, only a part of the fuel is used to provide the homogenous mixture. Therefore, the operating range of the PPCI engine is wider than that of the HCCI engine but is still limited at medium loads. However, reliable performance, combustion control, limited operating range, and higher HC and CO emissions are the main challenges that require investigation to find the optimum solution in terms of engine design, operating parameters, and fuel composition of different fuels such as biodiesels, alcohols, ethanol, butanol, hydrogen, CNG, LPG, etc.

A lot of important research related to this potential mode of combustion has been made to extend the operational range and reliability evident in recent years. This article is meant to provide a detailed analysis of the effects of the controlling strategies on the combustion, performance, and emission characteristics based on the review of published results of mostly experimental and numerical studies. Table 2 and Table 3 summarize the further recent extensive research work carried out by different researchers on the fuel injection timings and ratios strategy and fuel properties strategy respectively. The table describes a quick understanding of the effects of PCCI strategies on different engine combustion parameters and emissions. The most significant results of the above PCCI strategies on engine performance and exhaust emissions are summarized below.

1. In PCCI engines, ignition delay duration could be extended by using early fuel injection, using fuels with high RON, and/or using EGR leading to an overall lower combustion temperature as a consequence of retarding the combustion near TDC.
2. For a single injection strategy, advancing fuel injection timing was beneficial in terms of a reduction in NOx and soot emission, but with a negative impact on engine efficiency. Additionally, HC and CO emissions increased due to most of the fuel ends on the cylinder wall, and more percentage is trapped in crevice volume. Therefore, early injection leads to incomplete combustion due to cylinder wall impingement and wall wetting. Thus, using a single injection strategy can provide better combustion and higher efficiency than a double injection, but with a limited operating range. For the double injection strategy, retarding the second injection timing reduced NOx emissions, but soot, HC, and CO emissions increased and engine performance deteriorated due to the shifting of diffusive combustion to later than TDC. In the triple injection case, post-injected fuel does not burn effectively that leading to higher HC emissions but could be beneficial in controlling soot emissions by increasing the temperature of the exhaust gases. From the above discussion of the experimental reported earlier, it can be concluded that a double injection strategy with the properly timed second injection can help achieve better combustion stability and controllability at all speeds and loads.
3. The choice of fuel is an essential strategy to control PCCI engine performance and exhaust emissions as auto-ignition is mainly controlled by the species pools formed during low-temperature combustion. Fuels of high Octane Number are considered suitable for PCCI engines where ignition delay is longer to provide enough time for better mixture preparation which results in lower soot emissions and lower NOX due to lower combustion temperature as a result of lower energy content and higher latent heat of vaporization. But high RON has CO and HC due to incomplete combustion. Additionally, high RON fuel coupled with another control strategy such as early pilot injection might lead to problems of cold starting and misfires at low load, requiring a different control strategy at a low-speed operation like increasing intake temperature or using diesel and other high reactivity fuels which can offer a better control at low load. Lower injection, larger injector holes, and early injection timing might be required for lower reactivity fuels with high volatility to provide better combustion and lower CO and HC. While higher injection pressure and smaller injector holes provide better atomization and late injection of the pilot with a lower fuel ratio required for higher viscosity and density fuels like diesel, biodiesel …etc. to reduce the fuel trapping into the crevices.

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Conflicts of Interest:
The authors do not have any conflict of interest.

NOMENCLATURE & ABBREVIATIONS

ICE	Internal combustion engine	SOI	Start of injection
CI	Compression ignition	CO2	Carbon dioxide
SI	Spark ignition	TDC	Top dead center
PCCI	Partially premixed charge compression ignition	RON	Research octane number
HCCI	Homogeneous charge compression ignition	MON	Motor octane number
HC	Hydrocarbon	PRF	Primary reference fuel
CO	Carbon monoxide	RPM	Revolutions per minute
PM	Particulate matter	Pinj.	Injection pressure
NOx	Nitrogen oxide	1st Inj.	First injection timing
LTC	Low-temperature combustion	2nd Inj.	Second injection timing
EGR	Exhaust gas recirculation	Vd	Displacement volume
COV	Coefficient of variance	CR	Compression ratio
IMEP	Indicated mean effective pressure	SOC	Start of combustion
EOI	End of injection	HRR	Heat release rate
BMEP	Brake mean effective pressure	WGO	Wheat germ oil
BTE	Brake thermal efficiency	ULSD	Ultra-low sulfur diesel
CAD	Crank angle degree	PFI	Port fuel injection
ATDC	After top dead center	GDI	Gasoline direct injection
BTDC	Before top dead center	IVC	Intake valve closing
PPC	Partially premixed combustion	VVT	variable valve timing
SOFI	Start of first injection	SOI−P1	The start of the first pilot injection
SOSI	Start of the second injection	SOI−P2	The start of the second pilot injection

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