

# The Effects of Injection Strategy on Performance and Emission Characteristics of a Premixed Charge Compression Ignition Engine

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## ABSTRACT

This study presents an experimental analysis of the combustion characteristics, performance, and emission behavior of a diesel engine operating with the Premixed Charge Compression Ignition (PCCI) combustion, which was achieved by altering injection parameters, such as timing, pressure, and duration, at an engine speed of 1,800 RPM and a load of 20%. The results show that early injection timing improves mixture preparation while creating a low-temperature heat release (cool flame) and a pressure of 450 bar, which were optimal for stable PCCI operation. Under these conditions, efficiency increased by 10–12% compared to conventional diesel combustion, while the rate of pressure rise remained below the knock threshold. PCCI combustion resulted in substantial reductions in Nitrogen Oxides (NO<sub>x</sub>) and smoke emissions, due to lower peak combustion temperatures and more homogeneous mixture formation. However, unburned Hydrocarbon (HC) and Carbon Monoxide (CO) emissions increased due to reduced oxidation rates and incomplete combustion under low temperatures.

*Keywords-combustion characteristics; emission characteristics; Premixed Charge Compression Ignition (PCCI); injection timing*

## I. INTRODUCTION

Conventional diesel engines use a non-homogeneous air-fuel mixture, forming fuel-rich and poor regions, with high and low combustion temperatures, which increases the formation of NO<sub>x</sub>, resulting in higher emission levels than spark-ignition gasoline engines. In contrast, gasoline engines typically operate with a more homogeneous mixture, leading to lower soot formation and more stable combustion temperatures. Improved mixture homogeneity contributes to cleaner combustion and reduced formation of pollutants [1]. In order to overcome the emission limitations of conventional diesel combustion, Low-Temperature Combustion (LTC) strategies were used, such as PCCI. PCCI has some common characteristics with Homogeneous Charge Compression Ignition (HCCI), in which fuel is injected early to allow sufficient time for mixing with air. This results in a relatively homogeneous mixture before ignition. In HCCI operation, the premixed charge is

compressed during the compression stroke until auto-ignition occurs near the end of compression. This combines the high efficiency of diesel engines with the clean combustion characteristics of gasoline engines [2]. Unlike in conventional diesel combustion, combustion in HCCI and PCCI modes does not start with a spark or a single injection event. Instead, it is initiated by thermochemical conditions, such as intake temperature, equivalence ratio, and the amount of internal or external residual gas. Once the compressed mixture reaches the auto-ignition temperature of the fuel, combustion occurs nearly simultaneously at multiple locations within the cylinder. This distributed combustion process results in lower peak combustion temperatures, effectively suppressing NO<sub>x</sub> formation and inhibiting soot production while offering the potential for improved thermal efficiency [3]. Furthermore, dilution of the mixture using Exhaust Gas Recirculation (EGR) plays an important role in reducing emissions [4]. However, a major drawback of HCCI is the difficulty of controlling the

combustion process under high-load conditions. Additional issues, such as misfire at low loads, an inability to control ignition timing, and elevated HC and CO emissions, make this combustion mode unsuitable for commercial engines [5]. Authors in [6] examined PCCI combustion for hybrid engine applications, using n-heptane as a cetane improver and a secondary port-injected fuel, combined with a directly injected diesel-mahua biodiesel blend. The PCCI strategy promoted a more premixed combustion phase, resulting in a reduced in-cylinder temperature and significant decreases in  $\text{NO}_x$  and smoke emissions, showing that the premix ratio strongly influenced combustion behavior, particularly during the diffusion phase. Peak cylinder pressure and heat release phasing, however, were only marginally affected. Through the use of Response Surface Methodology (RSM) for multi-objective optimization, the optimal PCCI operating conditions were identified, improving brake thermal efficiency and reducing HC, CO,  $\text{NO}_x$ , and smoke emissions compared to conventional combustion. Authors in [7] examined the thermal, emission, and sustainability performance of a dual-fuel PCCI engine fueled with Lemon Seed Oil Methyl Ester (LSOME) biodiesel, diesel pilot fuel, and gaseous fuels (hydrogen and methane), with cerium oxide ( $\text{CeO}_2$ ) nanoparticles added. Optimal performance was observed, with the B20 biodiesel blend enriched with hydrogen and  $\text{CeO}_2$ . Although  $\text{NO}_x$  emissions increased due to elevated in-cylinder temperatures, a multi-criteria Pugh matrix sustainability assessment identified methane- and hydrogen-enriched LSOME blends as viable, economically favorable options. Authors in [8] examined the effects of engine speed extension, intake boost pressure, and multi-pulse injection timing on PCCI combustion and emissions in a heavy-duty diesel engine under medium load conditions. It was indicated that increased intake boost pressure promoted the formation of a leaner, more homogeneous mixture; advanced high-temperature ignition; and improved indicated thermal efficiency, while reducing combustion losses. Increasing engine speed delayed ignition timing and reduced  $\text{NO}_x$ , soot, CO, and UHC emissions, followed by higher thermal efficiency and lower combustion losses despite increased exhaust losses. Delayed multi-pulse injection timing increased the fraction of the over-rich mixture and initial soot formation. However, final soot, CO, and UHC emissions decreased with minimal impact on  $\text{NO}_x$ . Overall, the study shows that extending engine speed and managing air-fuel ratios can effectively expand the PCCI operating range while maintaining high efficiency and low emissions in heavy-duty diesel engines. The PCCI combustion strategy, developed from the HCCI concept, offers improved control over the combustion process [3]. While HCCI relies on a fully homogeneous mixture, PCCI is a hybrid of HCCI and conventional diesel combustion, where diesel fuel is injected early in combination with EGR to enhance mixture uniformity, resulting in partially premixed combustion [9]. Although PCCI engines emit higher levels of  $\text{NO}_x$  and particulate matter than HCCI engines, these emissions remain significantly lower than those of conventional diesel engines. The effects of various parameters, such as injection pressure, timing, fuel quantity, spray angle, and mixture strength, on the combustion process have been evaluated to optimize and expand the operating range of PCCI engines [10]. This study aims to analyze the

influence of injection timing on establishing PCCI combustion in a conventional diesel engine. The results can be used to predict optimal injection timing and guide experimental planning for the effective implementation of the PCCI combustion strategy.

## II. METHODOLOGY

### A. Experimental Setup and Test Engine

Figure 1 presents the layout of the test bench for the investigated engine. It includes experimental equipment, such as an AC dynamometer, an exhaust gas analyzer, a fuel supply system, an engine control system, and various measurement instruments. The test engine is a Changchai 192FA: a single-cylinder, four-stroke engine with a common rail fuel injection system, as presented in Table I.

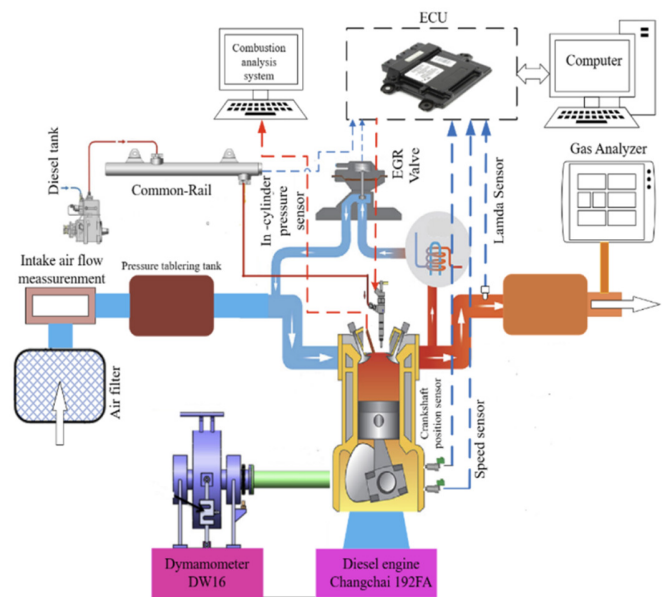


Fig. 1. Layout of the experimental engine on the test bench.

TABLE I. ENGINE SPECIFICATIONS

Parameters	Unit	Value
Cylinder bore: $D$	mm	92
Stroke: $S$	mm	76
Displacement volume: $V_h$	$\text{cm}^3$	504
Compression ratio: $\epsilon$	-	19
Rated power	kW	8.2
Rated speed	rpm	3600

The fuel injection process was controlled using INCA software, which allows the adjustment of key injector parameters, including injection timing, duration, and pressure. The test fuel was diesel, and its physical and chemical properties met the requirements of the TCVN 5689:2023 standard. This study used a comparative experimental approach between two combustion strategies: conventional diesel combustion and PCCI. The experimental procedure was carried out as follows:

- First, the engine was operated at speeds ranging from 1400 RPM to 3200 RPM with 400 RPM increments. The injection pressure was kept constant at 450 bar, and the quantity of injected fuel corresponded to a 20% load.
- At the specified load condition, the injection timing was varied from 35° to 70° Crank Angle (CA) Before Top Dead Center (BTDC). Then, combustion pressure data and engine emission parameters were recorded for each test condition.

### B. Theoretical Basis for Calculating Heat Release Rate

The heat release rate, which is the combustion behavior of the fuel at different points during the engine cycle, can be calculated using the pressure-CA diagram ( $P = f(\alpha)$ ). This method provides a simple yet accurate approach when using a zero-dimensional thermodynamic model. In this model, the temperature within the cylinder is assumed to be constant. Applying the first law of thermodynamics to the working fluid in the cylinder leads to:

$$dQ = dU + pdV + dQ_m + \sum h_i dm_i \quad (1)$$

where  $dQ$  is the heat released from fuel combustion,  $dU$  is the change in internal energy due to the variation in gas temperature,  $pdV$  is the work done by the gas on the piston,  $dQ_m$  is the heat transferred to the cylinder walls,  $h_i$  and  $dm_i$  are the enthalpy and mass of the gases entering or leaving through the clearance volumes  $dm_{kk}$  between the piston and the cylinder.

$$dU = d(m, u) = mdu + udm = mc_v dT - udm_{kk} \quad (2)$$

where  $m$  is the mass of gas in the cylinder,  $u$  is the specific internal energy of the gas,  $c_v$  is the specific heat capacity at constant volume, and  $T$  is the instantaneous in-cylinder gas temperature. It is assumed that the enthalpy of the gas entering or exiting the cylinder is only associated with the blow-by flow:

$$\sum h_i dm_i = \dot{h} dm_{kk} \quad (3)$$

The amount of heat transferred to the cylinder wall is:

$$dQ_m = Ah_c(T - T_w)dt = \frac{1}{\omega} Ah_c(T - T_w)da \quad (4)$$

where  $h_c$  is the convective heat transfer coefficient from the gas to the cylinder wall,  $A$  is the instantaneous heat transfer area,  $T_w$  is the cylinder wall temperature,  $\omega$  is the crankshaft angular speed (rad/s), and  $\alpha$  is the crank angle. Differentiating both sides and substituting into (1) results in the heat release rate:

$$\frac{dQ}{dt} = \frac{\gamma}{\gamma-1} p \frac{dV}{dt} + \frac{1}{\gamma-1} V \frac{dp}{dt} + (\dot{h} - u + c_v T) dm_{kk} + Ah_c(T - T_w) \quad (5)$$

where  $\gamma$  is the ratio of specific heats  $\gamma = \frac{c_p}{c_v}$ ,  $c_p$  is the specific heat capacity at constant pressure, and  $R$  is the gas constant.

## III. RESULTS AND DISCUSSION

### A. Combustion Characteristics of the Engine under Varying Injection Timing

To understand the combustion process in a diesel engine operating under PCCI mode, the engine's combustion characteristics were analyzed. This analysis focused on in-cylinder pressure evolution, pressure rise rate, heat release rate, and in-cylinder temperature. The goal was to identify the PCCI combustion mechanism. The engine operated under the PCCI combustion mode if the heat release rate profile exhibited a cool flame when varying the injection timing. Figure 2 shows that as the injection timing advances from  $-30^\circ$  to  $-45^\circ$  CA After Top Dead Center (ATDC), the in-cylinder pressure increases slightly and the peak pressure shifts gradually toward the top dead center. The maximum in-cylinder pressure increases from approximately 78–80 bar at an injection timing of  $-30^\circ$  CA ATDC to 82–84 bar at  $-40^\circ$  and  $-45^\circ$  CA ATDC. This behavior is due to the longer mixture preparation time associated with early injection, enhancing fuel evaporation, and air–fuel mixing. Thus, the main combustion occurs in a smaller cylinder volume near top dead center. According to thermodynamic principles, this leads to higher pressure. At the same time, the rate of pressure rise increases with advanced injection timing. The peak value increases from approximately 3–4 bar/degree to about 6–8 bar/degree. However, it remains below the knock threshold, indicating smooth engine operation. An additional early peak appears in the pressure rise rate curve as the injection timing is accelerated. This peak corresponds to the onset of low-temperature combustion, or cool flame, in the temperature range of approximately 700 K–900 K. This range is a characteristic feature of the PCCI combustion process. This low-temperature reaction releases some energy before the main combustion event, which moderates the combustion process and improves its stability. Thus, the injection timing range of  $-40$  to  $-45^\circ$  CA ATDC is suitable for establishing the PCCI combustion mode under the investigated operating conditions because it achieves reasonable peak pressure and a moderate pressure rise rate while providing a favorable basis for improving engine efficiency and emission characteristics.

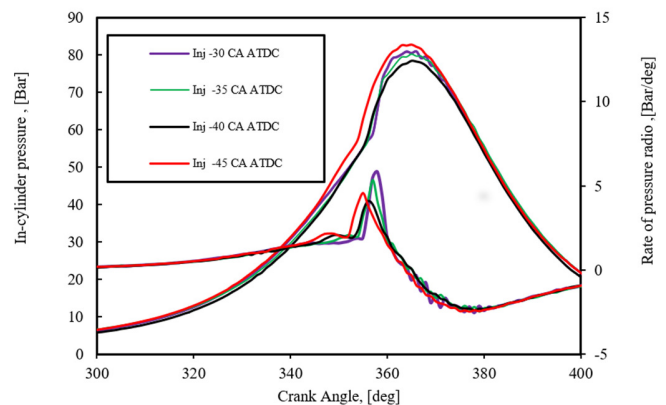


Fig. 2. In-cylinder pressure evolution and pressure rise rate of the engine as the injection timing is varied from 30° to 45° BTDC at 1800 rpm and 20% load.

Figure 3 displays the effects of injection timing on in-cylinder temperature evolution and heat release rate. As the injection timing advances from  $-30^\circ$  to  $-45^\circ$  CA ATDC, the in-cylinder temperature BTDC gradually increases, indicating enhanced low-temperature chemical reactions prior to the main combustion event. The peak in-cylinder temperature occurs slightly earlier and shows a modest increase when the injection timing advances to  $-40^\circ$  and  $-45^\circ$  CA ATDC, reaching approximately 1200–1250 K. This shift is attributed to the longer premixing period, which promotes more homogeneous mixture formation, allowing combustion to occur closer to top dead center, where the cylinder volume is smaller. The heat release rate profiles further highlight the transition toward the PCCI combustion regime. At  $-30^\circ$  CA ATDC, heat release is dominated by a single, sharp peak associated with conventional diesel diffusion combustion. As the timing advances to  $-35^\circ$ ,  $-40^\circ$ , and  $-45^\circ$  CA ATDC, a distinct low-temperature heat release stage emerges before the main heat release peak. This stage corresponds to cool-flame reactions occurring in the temperature range of approximately 700 K–900 K. The magnitude of low-temperature heat release increases with earlier injection timing. Meanwhile, the main heat release peak becomes broader and less abrupt, indicating smoother, more distributed combustion. However, when the injection timing is excessively advanced (at  $-45^\circ$  CA ATDC), increased fluctuations in the main heat release rate occur after the peak. This can be associated with fuel impacting the relatively cold cylinder walls and incomplete evaporation, resulting in non-uniform combustion. Figure 3 shows that earlier injection timing increases LTC activity and helps establish the PCCI combustion mode. However, an optimal injection timing range is needed to balance stable heat release, favorable temperature evolution, and combustion efficiency.

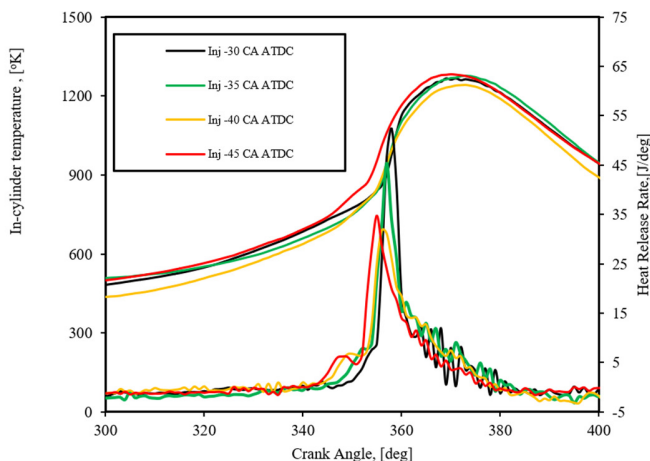


Fig. 3. In-cylinder temperature evolution and pressure rise rate of the engine as the injection timing is varied from  $30^\circ$  to  $45^\circ$  BTDC at 1800 rpm and 20% load.

Cool-flame ignition tends to start earlier as injection timing advances, changing slightly when injection timing occurs before  $45^\circ$  CA BTDC, and the peak magnitude of cool flame remains nearly unchanged, as illustrated in Figure 4. This behavior is attributed to early injection, which allows the

mixture to form more uniformly and be more evenly distributed within the cylinder, resulting in a more stable process. However, as the injection timing is advanced further, the heat release rate during the main combustion phase becomes unstable. This instability occurs because, when the wall temperature is not sufficiently high, part of the fuel burns on the cylinder walls, causing the combustion process to occur in a non-simultaneous manner. Figure 5 shows that the indicated efficiency of the engine operating under PCCI combustion is consistently higher than that of a conventional diesel engine across the full range of injection timings examined. On average, the PCCI engine's indicated efficiency increases by 10–12% compared to the baseline diesel engine, while the conventional diesel engine's indicated efficiency remains nearly constant because its injection timing is unchanged. This improvement is mainly due to PCCI combustion phasing occurring closer to top dead center. This enhances the fraction of useful work produced during the expansion stroke and reduces losses associated with late combustion. As the injection timing is advanced, the PCCI engine's indicated efficiency initially increases, reaching a maximum in the range of approximately  $-45$  to  $-55^\circ$  CA ATDC. Beyond  $-60^\circ$  CA ATDC, the indicated efficiency of the PCCI engine exhibits a slight decreasing trend. This reduction indicates that, from a quantitative perspective, a portion of the heat release occurs while the piston is still moving upward, diminishing the net indicated work and increasing compression losses.

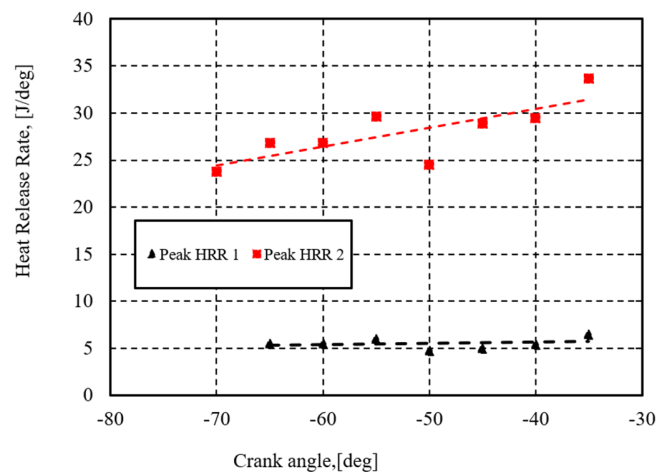


Fig. 4. Start of combustion as a function of injection timing at 1800 rpm and 20% load.

Additionally, excessively early injection can cause fuel to impact the cylinder walls, resulting in poor evaporation and incomplete combustion, reducing the effectiveness of converting energy into work. Regarding mechanical efficiency, a slight increase with advancing injection is indicated. This behavior can be explained by the fact that under PCCI operation, the peak in-cylinder pressure does not increase abruptly, and the rate of pressure rise is well-controlled, reducing mechanical loading and friction losses. However, this improvement is insufficient to offset the decrease in indicated efficiency at very early injection timings. These results suggest that the optimal injection timing range for achieving the highest

overall efficiency under PCCI operation is between  $-45$  and  $-60^\circ$  CA ATDC. At this range, high indicated efficiency is obtained while maintaining stable mechanical efficiency and avoiding excessive losses associated with overly advanced combustion. As presented in Table II, HC emissions increase under PCCI operation compared to a conventional diesel engine, exhibiting a clear rising trend as the injection timing is advanced. Specifically, HC emissions rise from approximately 120 ppm for the baseline diesel engine to 125–130 ppm at an injection timing of  $-45^\circ$  CA ATDC, and reach 150–155 ppm at  $-70^\circ$  CA ATDC. This corresponds to an overall increase of about 25–30%. This behavior is primarily due to the lower combustion temperatures and slower oxidation rates associated with PCCI combustion. Although these characteristics are beneficial for reducing  $\text{NO}_x$  and smoke emissions, they also promote incomplete combustion. Additionally, excessively early injection increases fuel impingement on the relatively cold cylinder walls, leading to fuel film formation and poor oxidized or quenched regions, thereby increasing unburned HC emissions.

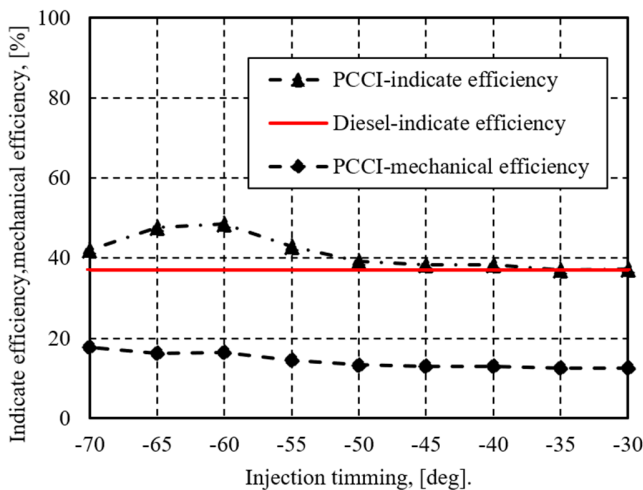


Fig. 5. Indicated efficiency and mechanical efficiency of the engine with single early injection varied from  $30^\circ$  to  $70^\circ$  BTDC, at 1800 rpm and 20% load, with an injection pressure of 450 bar.

TABLE II. COMPARISON OF HC EMISSIONS BETWEEN CONVENTIONAL DIESEL AND PCCI COMBUSTION AT DIFFERENT INJECTION TIMINGS

Operating condition	Injection timing (deg CA ATDC)	HC emission (ppm)
Diesel	45	118
PCCI	45	120
PCCI	50	122
PCCI	55	126
PCCI	60	132
PCCI	65	144
PCCI	70	154

Table III demonstrates how CO emissions vary under conventional diesel operation and PCCI mode with different injection timings. The results indicate that CO emissions are the lowest under conventional diesel operation, at approximately 0.08%. Under PCCI conditions, however, CO

emissions increase significantly and exhibit an upward trend as the injection timing advances. Specifically, CO emissions rise to about 0.18% at an injection timing of  $45^\circ$  CA ATDC, increasing further to 0.32% at  $50^\circ$  CA ATDC and reaching approximately 0.41% at  $55^\circ$  CA ATDC. As the injection timing is further increased, CO emissions rise to approximately 0.50% at  $60^\circ$  CA ATDC, 0.51% at  $65^\circ$  CA ATDC, and reach their maximum value of around 0.57% at  $70^\circ$  CA ATDC. These results suggest that early injection in PCCI mode reduces the average combustion temperature and limits the oxidation of CO into  $\text{CO}_2$ , resulting in significantly higher CO emissions compared to conventional diesel combustion.

TABLE III. COMPARISON OF CO EMISSIONS BETWEEN CONVENTIONAL DIESEL AND PCCI COMBUSTION AT DIFFERENT INJECTION TIMINGS

Operating condition	Injection timing (deg CA ATDC)	CO emission (ppm)
Diesel	45	0.08
PCCI	45	0.18
PCCI	50	0.32
PCCI	55	0.41
PCCI	60	0.5
PCCI	65	0.51
PCCI	70	0.57

Table IV shows how  $\text{NO}_x$  emissions vary under the same operating conditions. The results reveal that  $\text{NO}_x$  emissions are the highest in conventional diesel mode, reaching approximately 360 ppm due to diffusion-controlled combustion, which is characterized by high peak temperatures. Switching to PCCI operation results in a sharp decrease in  $\text{NO}_x$  emissions to approximately 45 ppm at  $45^\circ$  CA ATDC, which is more than an 85% reduction compared to conventional diesel operation. As the injection timing is increased,  $\text{NO}_x$  emissions continue to decrease, dropping to approximately 42 ppm at  $50^\circ$  CA ATDC, 40 ppm at  $55^\circ$  CA ATDC, 35 ppm at  $60^\circ$  CA ATDC, 30 ppm at  $65^\circ$  CA ATDC, and the lowest value of around 28 ppm at  $70^\circ$  CA ATDC. This substantial reduction is attributed to PCCI's LTC characteristics, which reduce the peak combustion temperature and the residence time of gases in high-temperature regions, preventing  $\text{NO}_x$  formation via the Zeldovich mechanism.

TABLE IV. COMPARISON OF  $\text{NO}_x$  EMISSIONS BETWEEN CONVENTIONAL DIESEL AND PCCI COMBUSTION AT DIFFERENT INJECTION TIMINGS

Operating condition	Injection timing (deg CA ATDC)	$\text{NO}_x$ emissions (ppm)
Diesel	45	359
PCCI	45	43
PCCI	50	45
PCCI	55	41
PCCI	60	37
PCCI	65	29
PCCI	70	28

Table V exhibits how smoke emissions vary under conventional diesel operation and PCCI mode with different injection timings. The results show that smoke emissions are the highest under conventional diesel operation, reaching approximately 30%. This outcome reflects the diffusion-

controlled combustion process, which is characterized by locally rich fuel-air regions that promote soot formation. Switching to PCCI operation sharply decreases smoke emissions to very low levels, which continue to decline as the injection timing is advanced. Specifically, smoke emissions are reduced to about 1.0% at an injection timing of 45°CA ATDC, corresponding to a reduction of more than 95% compared to conventional diesel operation. Further advancing the injection timing results in smoke emissions decreasing to approximately 0.8% at 50° CA ATDC, 0.6% at 55° CA ATDC, 0.5% at 60° CA ATDC, and 0.4% at 65° CA ATDC. Smoke emissions reach their lowest value of around 0.3% at 70° CA ATDC, due to the formation of a more homogeneous fuel-air mixture before combustion, preventing the formation of locally rich zones, which are the primary source of soot. Additionally, early injection and combustion under lower-temperature conditions reduce fuel pyrolysis and inhibit soot particle growth. These results highlight PCCI combustion's significant advantage in controlling smoke emissions and complement the substantial NO<sub>x</sub> emission reduction observed in Table IV, despite the increased CO emissions discussed in Table III.

TABLE V. COMPARISON OF SMOKE EMISSIONS BETWEEN CONVENTIONAL DIESEL AND PCCI COMBUSTION AT DIFFERENT INJECTION TIMINGS

Operating condition	Injection timing (deg CA ATDC)	Smoke emission (ppm)
Diesel	45	29.8
PCCI	45	0.8
PCCI	50	0.62
PCCI	55	0.46
PCCI	60	0.4
PCCI	65	0.35
PCCI	70	0.3

#### IV. CONCLUSIONS

This study examined the combustion characteristics, performance, and emission behavior of a diesel engine operating in Premixed Charge Compression Ignition (PCCI) mode with a single early injection at 1,800 RPM and 20% load. Advancing the injection timing from -30° to -45° Crank Angle (CA) After Top Dead Center (ATDC) established the PCCI combustion process, as indicated by the appearance of low-temperature heat release (cool flame) between 700 K and 900 K. Earlier injection improved mixture preparation and shifted the main combustion phase closer to top dead center. This resulted in a moderate increase in peak in-cylinder pressure, from approximately 78 - 80 bar to 82 - 84 bar, while maintaining the pressure rise rate below the knock threshold. Consequently, the indicated efficiency of PCCI combustion improved by about 10%–12% compared to conventional diesel combustion and reached its maximum within the injection timing range of -45 to -60° CA ATDC. PCCI combustion achieved significant reductions in NO<sub>x</sub> and smoke emissions, exceeding 85% and 95%, respectively, due to lower peak combustion temperatures and more homogeneous mixture formation. However, Hydrocarbon (HC) and Carbon Monoxide (CO) emissions increased with advanced injection timing due to reduced oxidation rates and incomplete combustion under low-temperature conditions. Future work should focus on

mitigating these increases while preserving the benefits of PCCI combustion, by optimizing injection strategies (e.g., split or multiple injections), applying Exhaust Gas Recirculation (EGR) and intake boosting, and evaluating alternative fuels or fuel blends with enhanced reactivity. Additionally, extending the investigation to higher engine loads and transient operating conditions, and coupling experimental results with advanced numerical simulations will provide further insights into the practical implementation and control of PCCI combustion in diesel engines.

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