Effects of Compression Ratio and Simulated EGR on Combustion Characteristics and Exhaust Emissions of a Diesel PCCI Engine*

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Abstract
The effects of compression ratio and simulated exhaust-gas recirculation (EGR) on combustion characteristics and exhaust emissions of a diesel PCCI engine were investigated using a single-cylinder test engine. Tests were carried out under constant speed with various compression ratios and EGR rates. Exhaust emissions and in-cylinder pressure were measured for all experimental conditions. Analyses based on engine performance and exhaust emissions were conducted. An optimum compression ratio that provided better indicated thermal efficiency and IMEP while yielding lower emissions of smoke, HC, and CO, and reasonable NOx without EGR was identified. High rates of EGR led to the simultaneous reduction of NOx and soot emissions due to a lower combustion temperature compared with conventional diesel combustion, with a slight penalty in HC and CO.

Key words: Premixed Charge Compression Ignition (PCCI), Engine Performance, Compression Ratio, Exhaust Gas Recirculation (EGR), Emissions

1. Introduction

Diesel engines are becoming increasingly popular for powering light-duty vehicles, because of their superior fuel economy, durability, reliability, and high specific power output compared with SI engines [1]. However, conventional diesel combustion suffers from the tradeoff between particulate matter (PM) and nitrogen oxides (NOx). With growing concerns about environmental protection, global warming, and the limited supply of fossil fuels, new diesel engines should be developed that are able to generate reduced PM and NOx while maintaining or even improving fuel efficiency. To meet strict emission targets set for these two species, new combustion strategies, such as homogeneous charge compression ignition (HCCI), have been investigated. HCCI eliminates locally rich fuel–air mixtures and reduces combustion temperatures, thus achieving lower PM and NOx. However, HCCI faces many challenges, including a lack of combustion phasing control under different operating conditions and fuel with different properties, high pressure-rise rates under high loads, and limitations in creating a homogenous fuel–air mixture [2].

Many strategies have been developed to control the in-cylinder combustion process, in addition to the use of exhaust aftertreatment devices. In-cylinder control parameters include injection pressure, number of injections, shape and timing, EGR, and swirl ratio [3, 4]. The most practical means of implementing a HCCI combustion strategy in a modern direct-injection diesel engine is through partially premixed-charge compression ignition, where fuel and air are not fully homogenous, but the combustion event can be controlled more readily.
In a PCCI combustion strategy, fuel can be introduced into the combustion chamber through port fuel injection, early direct injection, or late direct injection. Port fuel injection and early direct injection often suffer from incomplete fuel evaporation and fuel spray impingement on the cylinder walls, which cause high levels of HC and CO as well as fuel/oil dilution [5-8]. Strategies to reduce fuel-wall impingement explored in the past include the use of narrow spray-cone angle injectors [9-12] and use of uncooled EGR [13]. A late direct-injection approach avoids fuel-wall impingement and provides good control of combustion phasing. A standard injector can be used, thus allowing for transition from PCCI at low and medium load to conventional diesel combustion at higher loads [14-16]. Under PCCI, the desired ignition delay is achieved through a lower compression ratio, enhanced charge motion, higher injection pressure, and relatively large amounts of cooled external exhaust gas recirculation (EGR). PCCI is not fully homogeneous like HCCI, but it makes use of injection timing and EGR to greatly increase the controllability of combustion phasing and the rate of combustion. Much research has been conducted to expand the high-load limits of PCCI using fuel properties [17, 18], split injection [19-22], variable valve timing [23], and fuel–air mixing enhancement [14]. PCCI combustion strategies employing moderately early injection (approx. 25° BTDC) have been widely investigated because they are advantageous in preventing lubricant dilution. Under such cases, high levels of EGR coupled with low compression ratios are used to ensure sufficient air–fuel mixing time, leading to suppression of NOx formation and better combustion phasing. Thus, to achieve low soot and NOx simultaneously under moderately early injection timing it is necessary to optimize the injection pressure, EGR rate, compression ratio, and injection timing [24-27]. Like HCCI, PCCI is prone to high HC and CO emission and a high pressure-rise rate, which results in high combustion noise; additionally, it cannot be used at higher engine loads. Previous research has indicated that simultaneous low NOx and soot can be achieved by initiating combustion at an equivalence ratio below 2 and flame temperatures below 1800 K during combustion [15].

To clearly understand the combustion mechanism inside the combustion chamber in PCCI combustion mode, it is necessary to study the effect of compression ratio and EGR rate in the mixing of the air–fuel mixture, the heat-release process, and the formation of soot, NOx, and other combustion products. Available data are still insufficient to fully understand the relationship among the compression ratio, mixture formation, and the combustion process in PCCI combustion mode.

The present study sought to understand the combustion characteristics and emissions formation in a PCCI engine. Compression ratios of 12, 13, 14, and 15 with a slightly lower fuel injection pressure ($P_{inj}$) of 80 MPa and a slightly narrowed cone angle of 140° were chosen to avoid cylinder-wall wetting. Compression ratios and EGR were varied to establish the best operating conditions for simultaneous low NOx and soot emission and high efficiency.

2. Experimental setup and procedure

A four-stroke single-cylinder direct-injection supercharged diesel engine with a displacement of 781.7 cm$^3$ was used. Table 1 shows the engine operating conditions. Table 2 shows the specification of the fuel used, JIS #2 diesel fuel, which is commonly available in Japan. A schematic of the research engine is shown in Figure 1, and its specifications are shown in Table 3. A common rail injection system capable of developing an injection pressure ($P_{inj}$) of 180 MPa was used. For this work, a slightly narrowed cone-angle 140° injector at 80 MPa injection pressure was selected to readily isolate the effect of compression ratio and EGR rate. The injected mass of fuel ($m_f$) was kept constant ($m_f = 12.2$ mg/cycle, equivalent to $\lambda = 4.5$ with 0% EGR and $\lambda = 3.0$ with 40% EGR where $\lambda$ is excess air ratio). The TDC signals and every half-degree crank angle were detected by
photo interrupters and coupled with a controller to control the injection timing and diesel-fuel-injection duration. The in-cylinder pressure was measured with a piezoelectric pressure transducer (6052C, Kistler) coupled with a charge amplifier (5011B, Kistler). The pressure history was analyzed to obtain the rate of heat release to investigate the combustion characteristics. Exhaust emissions were analyzed using a NOx–CO analyzer (Horiba, PG-240), HC analyzer (Horiba, MEXA-1170HFID), and a smoke meter (Horiba, MEXA-600s). Data for each engine condition were captured when the engine was in equilibrium, where there was almost no change in the emission parameters and exhaust temperature. In this study, simulated EGR based on the mass of the N2 dilution in the intake manifold was used, as indicated in the formula below.

\[
\text{EGR rate} = \frac{N_2}{\text{Air} + N_2}
\]

(1)

![Figure 1. Schematic of single-cylinder test engine with exhaust emission analyzer](image)

<table>
<thead>
<tr>
<th>Table 1. Engine operating conditions</th>
<th>Table 2. Test fuel properties (JIS #2 diesel fuel)</th>
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<tbody>
<tr>
<td>Description</td>
<td>Unit/parameter</td>
</tr>
<tr>
<td>Engine</td>
<td>1000rpm</td>
</tr>
<tr>
<td>Injection</td>
<td>80MPa</td>
</tr>
<tr>
<td>Injection (°BTDC)</td>
<td>2,5,10,15,20,25,30,35,40</td>
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<tr>
<td>EGR rate</td>
<td>0, 40%</td>
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<tr>
<td>Intake</td>
<td>40°C</td>
</tr>
<tr>
<td>Coolant</td>
<td>80°C</td>
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<tr>
<td>Lubricating</td>
<td>80°C</td>
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<tr>
<td>Intake</td>
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<tr>
<td>Cetane number</td>
<td>57-60</td>
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<tr>
<td>Density (°C)</td>
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<tr>
<td>Lower heating value</td>
<td>42.9 kJ/kg</td>
</tr>
<tr>
<td>Sulfur mass</td>
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<tr>
<td>Flash point</td>
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</table>
Real EGR contains CO₂, H₂O, N₂ and O₂ in thermodynamically significant quantities and CO, THC, NOₓ and soot in thermodynamically insignificant quantities. The effect of real EGR can be divided into three namely; dilution effect, thermal effect and chemical effect. Ladommatos et al. [28], Abd-Alla G.H. [29], and Zheng et al. [30], have noted that the dilution effect is substantial compared to the chemical and thermal effect on combustion and exhaust emissions. This has been found to be effective in extension of ignition delay and reduction of NOₓ at the expense of higher particulate and unburned hydrocarbon. The real EGR system would be considered in future work. In this work we considered the effect of nitrogen dilution, which has been studied previously [31] in our laboratory. The baseline tested compression ratio (CR) was 15 with a Derby hat piston-head configuration. A reduction in the compression ratio was obtained using different piston heads with different bowl depths (varying the value of d), while the same bowl shape was maintained, as shown in Figure 2 and Table 4. The Derby hat piston-head configuration was chosen to allow spray to be guided into the piston bowl, thus promoting the mixing of the air and fuel within the piston bowl, avoiding cylinder-wall wetting under moderately early injection timing. Recent research [32] on diesel engines has focused on narrow-angle wall-guided sprays when considering late injection timing. While it is a promising technology, a flexible system that can enable an easier transition to conventional diesel combustion is still needed. For an easier transition from the PCCI regime at moderately early injection timing to conventional diesel combustion, a slightly narrowed angle of 140° with the Derby hat piston geometry was used in this study.
3. Experimental results and discussion

3.1 Effect of compression ratio on diesel PCCI combustion

The aim of reducing the compression ratio is to reduce the in-cylinder temperature and thus the flame temperature during combustion to suppress NO\textsubscript{x} emissions. However, major reductions have not been achieved to date because of the associated negative effects on cold-start capability.

Figure 3 shows the pressure history and the rate of heat release (ROHR) for the different compression ratios (CRs) at an injection timing $\theta_{\text{inj}}$ of 20° BTDC. The in-cylinder pressure decreased as the compression ratio was reduced from 15 to 12. The same trend was observed with the ROHR. This is consistent with the expected trend because the in-cylinder temperatures would decrease as the compression ratio is decreased. Under conditions of CR12, the ignition delay was increased compared with CR15 because of the lower in-cylinder compression pressure and charge temperature. With CR14 and above, a knocking phenomenon and in-cylinder pressure fluctuations were noted. CR13 was observed to have normal combustion with a smooth pressure curve.

To further clarify trends in the PCCI combustion regime, the rate of heat release under CR13 was investigated in detail. Figure 4 shows the rate of heat release at different injection timing $\theta_{\text{inj}}$ for CR13. Late injection timing leads to predominantly premixed combustion,
with a slight proportion of diffusion-controlled combustion. This combustion is slightly different from conventional diesel combustion because a large portion was premixed. Early injection timing of $\theta_{\text{inj}} \geq 20^\circ \text{BTDC}$ exhibited a two-stage ignition, with the first stage representing the low-temperature oxidation (LTO) and the second stage, the high-temperature oxidation (HTO) reaction phenomenon, predominating in PCCI combustion.

To understand the relationship between the compression ratio and engine performance within a diesel PCCI engine, the indicated thermal efficiency, the indicated mean effective pressure (IMEP), and the coefficient of variance of IMEP were analyzed in detail. Figure 5 shows the effect of compression ratio on indicated thermal efficiency and IMEP. As the injection timing was advanced, it was noted that the indicated thermal efficiency and IMEP decreased for all the compression ratios tested, but with greater decreases for higher compression ratios. A compression ratio of 14 and above led to lower indicated thermal efficiency and IMEP under moderately early injection timing. This was thought to be due to poor fuel combustion and higher negative work. Of all the cases investigated, CR13 showed superior values, more so with moderately early injection timing without EGR.

To further understand the relationship between different compression ratios and specific engine emissions in a diesel PCCI engine, a detailed investigation was conducted. Figure 6 shows the specific emissions as function of injection timing for various compression ratios. CO and HC showed similar trends in that as the injection timing was advanced, the emissions increased. This could be attributed to low in-cylinder temperature leading to higher fuel-spray penetration, causing fuel deposits on the cylinder wall and incomplete combustion.

NOx increased drastically as the injection was advanced to a maximum at $20^\circ \text{BTDC}$, and then decreased thereafter. Smoke emission showed lower values under the retarded condition, but increased as the injection was advanced. For all the cases considered, CR12 had the highest emissions of smoke, HC, and CO with the lowest emission of NOx, whereas CR15 had the highest emission of NOx with the lowest emissions of HC, smoke, and CO.
Of all the test conditions considered, CR13 had approximately the same emissions of HC, CO, and smoke as CR15, but with slightly lower emissions of NOx. This indicates that there is an optimum value of compression ratio below which the exhaust emission deteriorates. In this case, CR13 was noted to be the optimum compression ratio.

A tradeoff was observed between smoke and NOx emissions. An increase in NOx was accompanied by a corresponding decrease in smoke. Higher in-cylinder temperatures enhance production of NOx but lead to rapid oxidation of smoke, and the converse is also true. Lower temperatures lead to a longer ignition delay, which gives the air–fuel mixture sufficient time to premix, thus avoiding fuel-rich zones, which burn with soot while at the same time inhibiting the formation of NOx.

CR12 clearly had the highest emissions of smoke, which was considered a limiting factor despite the low NOx emissions, whereas CR14 and above led to knocking and high pressure fluctuations with lower indicated thermal efficiency and IMEP. Of all the investigated compression ratios, CR13 was considered the most appropriate because it had the best indicated thermal efficiency, IMEP, and exhaust emissions. This compression ratio was selected for further analysis of combustion characteristics and exhaust emissions under differing amounts of EGR.

3.2 Effect of exhaust gas recirculation (EGR) on diesel PCCI combustion

EGR has been noted in the literature to improve IMEP and to extend the ignition delay [24, 27]. In conventional diesel combustion, it has been used to control emissions of NOx through charge dilution and lowering the adiabatic flame temperature. In this section, an in-depth analysis of the effects of EGR on performance and emissions will be presented.

The in-cylinder pressure history and ROHR with and without EGR are shown in Figure 7. Late injection timing led to typical diesel combustion, dominated by premixed combustion and a small diffusion-controlled combustion proportion. The injection duration was 9.2° CA, which ended during the premixed-combustion phase. Advancing the injection timing beyond 20° BTDC, the heat-release pattern exhibited PCCI combustion with two-stage heat release. The first was attributed to low-temperature oxidation, and the second to high-temperature oxidation, separated by a short delay due to the negative temperature coefficient (NTC) ignition behavior of the mixture. At 30° BTDC, a decrease in peak in-cylinder pressure and ROHR was noted as a result of the leaner mixture formed because ignition delay was prolonged due to the low charge temperature at the time of injection.

Under EGR conditions, the onset of ignition for both LTO and HTO were greatly affected due to the dilution effect. A greater influence of EGR was seen with \( \theta_{\text{inj}} = 10° \) and 15° BTDC, where the start of combustion was delayed into the expansion stroke where the in-cylinder temperature was rapidly decreasing, and the rate of heat release was slowed down. The combustion characteristic at \( \theta_{\text{inj}} = 15° \) BTDC is typical of the modulated kinetics of combustion proposed by Kimura et al. [14], although we achieved this without the heavy EGR and high swirl they used. For moderately early injection timing, the LTO and HTO phases were clearly seen despite the fact that they were phasing before TDC. With EGR, the ignition delay was prolonged.

To understand the effects of EGR on performance in the diesel PCCI engine, an investigation was carried out. Figure 8 shows the effect of EGR on indicated thermal efficiency, IMEP, and the coefficient of variance of IMEP (COV \(_{\text{IMEP}} (%)\)). As the injection timing was advanced, the indicated thermal efficiency and IMEP gradually decreased with or without EGR. The highest IMEP corresponded to phasing close to the TDC (-2° ATDC) without EGR. With 40% EGR, higher indicated thermal efficiency and IMEP were observed across the injection sweep, apart from late injection, where a marked reduction was observed. This could have been due to over-mixing, leading to an air–fuel mix that was too lean to burn. In conventional diesel combustion, advancing the injection timing leads to
higher pressure-rise rates but drastically reduces it in a PCCI regime. This could be attributed to the fact that in conventional diesel combustion, the premixed part burns very fast, whereas in PCCI, a longer premixing time and a lower in-cylinder temperature results in milder combustion. As the amount of EGR introduced increases, the pressure-rise rate drastically declines because of the dilution effect, leading to extended ignition delay and milder combustion. The combustion phasing was noted to retard with the introduction of EGR, correspondingly leading to improvement in the IMEP because the in-cylinder temperature was reduced, resulting in milder combustion coupled with the fact that negative work was greatly minimized. The COVIMEP was not affected with or without EGR for the injection timings considered. The values were in the range of 1–3%, indicating that the combustion process was stable under these conditions.

In PCCI combustion, it is common to use EGR to lower the adiabatic flame temperature. This reduces NOx at the expense of increased emissions of HC and CO. Figure 9 shows the effect of EGR on smoke, NOx, CO, and HC emissions. In cases without EGR, as the injection timing was advanced, NOx emissions drastically increased, up to a maximum at 20° BTDC, and then declined drastically in more advanced conditions. This could be due to the high temperature, which promotes NOx emissions and a very short time for fuel–air mixing prior to combustion. PCCI conditions with early injection above 30° BTDC and all cases with 40% EGR achieved very low emissions of NOx. This was attributed to the reduced local flame temperature and reduced concentration of oxygen in the intake charge.

Early injection timing, θinj = 30° BTDC and above, without EGR was observed to lead to excessive smoke emissions. This could be attributed to a low in-cylinder temperature that did not allow the soot formed from the rich fuel impinging on the surface of the piston to be oxidized. Smoke emissions below 2% were achieved for θinj = 2 to 30° BTDC with 40% EGR and with no EGR. When the injection timing was advanced, the in-cylinder temperature and pressure decreased, and fuel-spray penetration was enhanced, leading to fuel deposits on the surface of the piston, providing a source of fuel-rich zones and promoting smoke emission. The emission of smoke in conventional diesel combustion is less than that in PCCI combustion regime. The major purpose of PCCI combustion is not to completely eliminate smoke but to reduce it to acceptable levels. It has been noted by many researchers that it is possible to obtain simultaneous reduction of NOx and soot by incorporating a very large amount of EGR (>67%) [33, 15], but this comes at the expense of higher fuel consumption and increased emissions of CO and HC. With 40% EGR, injection timing between θinj = 15 to 25° BTDC was found to achieve less than 0.5% smoke. This was attributed to the fact that the air–fuel mixture could be well premixed without creating fuel-rich pockets, leading to better combustion. The application of EGR increased ignition
delay and offered sufficient mixing time for fuel and air before the start of combustion.

Figure 10 shows the relationship between smoke and NO\textsubscript{x} without and with 40\% EGR. It can be seen that as NO\textsubscript{x} decreased, the smoke increased, but with the introduction of EGR, it was possible to achieve simultaneous reduction of both emissions under a PCCI combustion regime.

**Figure 8.** Effect of EGR on indicated thermal efficiency, IMEP and COV\textsubscript{IMEP} %

**Figure 9.** Effect of EGR on specific emissions

![Figure 8](image_url)

![Figure 9](image_url)
Furthermore, as the injection timing was advanced, CO emissions increased dramatically. Higher EGR led to higher CO. Under a retarded condition with 40% EGR ($\theta_{\text{inj}} = 10^\circ$ BTDC) CO emissions were slightly higher.

This condition could be related to the HC emissions; it could be attributed to the lower temperature and insufficient oxidant for complete combustion. When injection timing is too early, higher emissions could also be associated with piston-bowl wall wetting.

As the injection timing was advanced, a corresponding increase in HC emissions was observed. This trend was also seen with 40% EGR. This could be attributed to the availability of only a limited amount of oxygen to complete combustion. At $\theta_{\text{inj}} = 10^\circ$ BTDC with 40% EGR conditions, the HC emissions were higher than without EGR. This could be attributed to two effects: over-mixing, where the mixture is too lean to burn, and flame quenching in the expansion stroke. This result is consistent with the work of Han et al. [34] and Colban el al. [35]. To mitigate this problem, it would be desirable to increase the available air mass through supercharging while maintaining the EGR levels.

### 4. Conclusions

The main objective of this paper was to analyze the effects of compression ratio and simulated EGR on the combustion characteristics and exhaust emissions in a diesel PCCI engine. A single-cylinder test engine was used. Simulated EGR, consisting of N$_2$ gas, was used to achieve lower NO$_x$. The following conclusions were reached under the test condition of this study.

1. PCCI combustion at all the tested compression ratios was achieved with moderately advanced injection timing without using EGR. This resulted in lower indicated thermal efficiency and IMEP. A compression ratio of 13 was noted to have lowest emissions of smoke, HC, and CO, reasonable amounts of NO$_x$, and better indicated thermal efficiency and IMEP.

2. The introduction of EGR enabled combustion phasing to occur closer to and after TDC with milder reaction rates, thus achieving higher indicated thermal efficiency and IMEP. A 40% EGR rate was found to be sufficient to achieve simultaneously low NO$_x$ and soot emissions with a slight penalty in HC and CO emissions.
References


