

Article

Ethanol Effects on Combustion and Emissions in Partial HCCI Diesel Engines Under Variable Load and Speed Conditions

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Abstract

Ethanol, a cleaner fuel alternative in diesel engines, is used in certain proportions to modify engine performance and improve emissions in diesel engines. In this study, a series of experiments were conducted to determine the effect of different ethanol concentrations on engine performance and emissions in a single-cylinder, water-cooled partial homogenous charge compression ignition (HCCI) diesel engine. Partial HCCI combustion was achieved by port injecting ethanol to form a homogeneous ethanol–air mixture that entered the cylinder, followed by direct diesel injection near the end of the compression stroke to initiate combustion. Experiments were primarily conducted with diesel fuel to achieve performance and emission values of conventional diesel engines. Subsequently, tests were conducted for ethanol contents of 5%, 10%, 15%, and 20%, respectively, under different load and engine speed conditions. Analysis of the experimental results revealed that ethanol fumigation improved combustion and engine performance under all operating conditions. Compared with neat diesel fuel, the E20 blend increased the maximum cylinder pressure by approximately 10% and the peak heat release rate by nearly 38% under full-load conditions. Furthermore, ethanol fumigation improved brake thermal efficiency and reduced brake-specific fuel consumption. In terms of emissions, CO and CO₂ emissions decreased by approximately 40% and 18%, respectively, whereas NO_x and HC emissions increased by nearly 25% and 57%, respectively, with increasing ethanol content. Among the investigated fuels, E20 provided the most favorable overall balance between combustion enhancement, engine performance, and emission characteristics.

Keywords: partial HCCI combustion; ethanol fumigation; combustion characteristics; exhaust emissions



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1. Introduction

One of the most promising approaches developed to improve combustion efficiency and reduce exhaust emissions in internal combustion engines is the homogeneous charge compression ignition (HCCI) combustion concept, which is classified as a low-temperature combustion (LTC) strategy [1–4]. HCCI technology combines the advantages of spark-ignition (SI) and compression-ignition (CI) engines by enabling the formation of a homogeneous fuel–air mixture prior to combustion while maintaining compression-induced autoignition characteristics [1,2,5]. In HCCI operation, the premixed charge is compressed until the temperature and pressure become sufficiently high to initiate spontaneous ignition

at multiple locations within the combustion chamber, resulting in rapid and volumetric combustion [2,5,6]. This combustion mode offers significant potential for improving thermal efficiency while simultaneously reducing particulate matter and nitrogen oxide emissions compared with conventional diesel combustion [3,4]. Recent advances in advanced combustion systems have further highlighted the critical roles of fuel reactivity, ignition delay, combustion phasing, and heat release control in determining combustion efficiency and pollutant formation. Studies have shown that the interaction between fuel properties and combustion development significantly influences ignition characteristics, heat release behavior, combustion stability, and emission formation mechanisms in LTC systems [7,8]. Furthermore, optimization of fuel–air mixture preparation and combustion kinetics has been identified as a key factor for achieving stable and efficient combustion with alternative fuels [7,8]. In this context, oxygenated renewable fuels such as ethanol have attracted considerable attention because of their high oxygen content, high latent heat of vaporization, and potential to improve mixture homogeneity. These characteristics make ethanol a promising candidate for enhancing combustion quality and emission performance in partial HCCI engines. Therefore, further investigations into ethanol-assisted partial HCCI combustion under varying operating conditions are necessary to better understand the interactions between fuel reactivity, combustion development, and emission formation, providing the primary motivation for the present study.

Among the most common advantages of HCCI engines are low NO_x and PM emissions [1,2,9,10]. The absence of localized rich mixture zones in HCCI combustion prevents soot formation, while combustion with lean mixtures at low temperatures significantly reduces temperature-dependent NO_x formation by bringing it below the critical threshold of 1800 K [10–12]. In addition, thermal efficiency is higher compared to conventional engines thanks to combustion occurring at a near-constant volume rate and low combustion temperatures reducing heat losses from the cylinder walls [5,10,13]. However, there are serious disadvantages that limit the commercial use of HCCI engines [13–15]. The most critical challenge is the lack of a direct physical mechanism (such as a spark plug or injector) to control ignition timing and combustion rate [5,6,10]. In addition, these engines have a very narrow operating range due to the risk of severe knocking at high loads and unstable operation problems at low loads [15,16]. High hydrocarbon (HC) and carbon monoxide (CO) emissions resulting from the incomplete oxidation of the fuel due to the low combustion temperature are also among the other important disadvantages [2,3,15,17].

Researchers have developed different LTC strategies such as premixed compression ignition (PCCI) combustion, reactivity controlled compression ignition (RCCI) combustion, and partial HCCI (or partial premixed compression ignition—PPCI/PPCCI) combustion to overcome the limitations of HCCI. Among these strategies, partial HCCI stands out in terms of the advantages it offers [11,14,17–19]. In this technique, a portion of the fuel is injected into the intake manifold or into the cylinder in the early stages of the compression stroke to create a homogeneous mixture, while the remainder is injected directly around top dead center (TDC) to trigger and control the combustion timing. In this way, the combustion rate is made gradual, and emission reductions can be achieved over a wider load range by reducing the risk of knocking [3,18,20].

The literature on LTC strategies covers a wide range of fuels, from neat diesel to various alternative fuels [1,21,22]. Partial premixed lean compression ignition (PPLCCI) experiments conducted by El Shenawy et al. [19] and performed with pure diesel fuel proved that BTE could be increased by 28.79%, BSFC could be decreased by 14.5%, and NO_x could be decreased by 75% at a 30% premix ratio. Duraisamy et al. [23] analyzed the methanol/PODE dual-fuel RCCI combustion mode using polyoxymethylene dimethyl ether (PODE) instead of diesel as a high-reactivity fuel and managed to control brake-

specific hydrocarbon (BSHC) and brake-specific carbon monoxide (BSCO) emissions more effectively while increasing combustion efficiency with this method. Kale and Krishnasamy [24] expanded the narrow load range of diesel-fueled HCCI from 38% to 86% and achieved a 99% improvement in soot emissions compared to conventional combustion by using isopropanol–gasoline mixtures with cetane enhancer additives in HCCI mode. Ali et al. [10] managed to control the combustion phase of the HCCI engine and expand its narrow operating range by using n-butanol and high-reactivity diethyl ether (DEE) fuel pair; in this regard, by increasing the n-butanol ratio, they achieved a high indicated thermal efficiency of 43.2% and zero NO_x and soot emissions. Feroskhan and Gobinath [25] investigated the performance of a compression ignition engine running on biogas and biodiesel by mixing butanol into the secondary fuel under RCCI and HCCI combustion modes; they showed that the RCCI mode, with the addition of butanol, resulted in lower CO emissions, whereas the HCCI mode exhibited very low NO_x and PM emissions.

In partial HCCI applications, alcohol fuels are widely preferred to compensate for the high reactivity and low evaporation characteristics of diesel fuel [2]. Among them, ethanol ($\text{C}_2\text{H}_5\text{OH}$) is one of the most widely used carbon-neutral biofuels because it can be produced from renewable resources such as corn, sugarcane, and agricultural waste. Its high oxygen content (approximately 35%), high octane number (108–116), low cetane number (5–8), and high latent heat of vaporization (840 kJ/kg) promote cleaner combustion, improve combustion phasing by delaying auto-ignition, and reduce cylinder temperature, thereby contributing to lower NO_x and soot emissions [2,26–31]. Ethanol is introduced into diesel engines by blending with diesel fuel, port injection (fumigation), or dual direct injection systems [18,29,31]. Among these methods, port injection enables ethanol to mix homogeneously with the intake air before entering the cylinder, requiring only minor modifications to the existing fuel system while simultaneously reducing NO_x and soot emissions. However, incomplete combustion of the homogeneous mixture may increase HC and CO emissions, particularly under low-load operating conditions [1,3,29,31].

In research on the use of ethanol in diesel engines, Ge et al. [27] found that using up to 15% ethanol with the direct mixing method increased BTE but increased BSFC due to the low heating value. Similarly, Varghese et al. [32] reduced BSFC by 5.5% and the smoke level by 32% by replacing 40% of the diesel fuel with ethanol at full load with manifold injection. El-Seesy et al. [26] reported that the direct mixing method is limited to a 20% ethanol ratio due to phase separation and low cetane number problems, whereas port injection allows the use of up to a 60% ethanol ratio. Studies in the literature on the application of port injection methods show that this technique significantly reduces smoke density and NO_x levels [3,33]. Wu et al. [33] reported that the use of ethanol with port injection is more effective than gasoline in reducing emissions. Similarly, Teoh et al. [3] reported that the combined use of manifold injection and intake air heating effectively controlled combustion timing and reduced NO_x emissions by 14.5%. They also observed that increasing the premixed ratio altered the in-cylinder temperature distribution, leading to higher HC and CO emissions.

Studies on the use of ethanol in diesel engines within the scope of LTC strategies provide critical findings in terms of improving combustion stability and emissions. In this context, Pan et al. [34], by examining ethanol/diesel RCCI combustion in a heavy-duty diesel engine, minimized cyclic variations by optimizing injection timing and premix ratio and determined a premix ratio value of 52% for the most stable operation conditions. Asad et al. [35], in a dual-fuel combustion strategy in which ethanol was injected into the intake manifold and diesel was directly injected, managed to reduce NO_x emissions by 88% and soot emissions by 96% compared to conventional diesel combustion by employing high exhaust gas recirculation (EGR) ratios and optimized ethanol energy ratios (40–60%).

Elbanna et al. [18], in their research on diesel/ethanol advanced combustion strategies, found that the direct dual-fuel stratification (DDFS) mode reduced HC emissions by 52% and CO emissions by 41% and improved combustion control compared to pure diesel PCCI operation. Gawale and Naga [2], in their study using ethanol by converting a diesel engine to dual-fuel HCCI mode, radically reduced NO_x emissions with optimized premix ratios; however, they reported that the specific fuel consumption increased due to the low heating value of ethanol. Wu et al. [33] found that port injection of ethanol and gasoline in partial HCCI mode effectively minimized smoke density. Focusing on the risk of knocking at high loads, Wei et al. [36] numerically demonstrated that increasing the ethanol substitution ratio to 40% in an RCCI engine increased the combustion rate and induced knocking; however, the disadvantages were mitigated through EGR, and a BTE of 42% was achieved for EGR25. In terms of expanding operational limits, Telli et al. [6] slowed down heat dissipation by using a water injection strategy in ethanol HCCI combustion and thus managed to triple the maximum operational load of the engine while keeping NO_x emissions below 0.8 g/kWh. Kale and Krishnasamy [14] expanded the narrow load range of diesel-fueled HCCI from 38% to 86% by adding cetane enhancer additives to ethanol–gasoline mixtures and achieved a 98% improvement in soot emissions and a 76% improvement in NO_x emissions compared to conventional combustion. Tongroon and Chuepeng [37] reported that ethanol premixing reduced soot emissions but increased BSFC compared to gasoline premixing because of its lower heating value.

Although previous studies have demonstrated the potential of ethanol-assisted LTC combustion strategies in reducing emissions and improving combustion characteristics, most investigations have been limited to fixed engine speeds, narrow load ranges, or a single ethanol substitution ratio. Consequently, the combined influence of different ethanol port injection ratios under wide-ranging engine operating conditions has not yet been comprehensively clarified for partial HCCI combustion [38]. Furthermore, comparative experimental data evaluating combustion, performance, and emission characteristics over practical engine operating conditions remain limited.

To address this research gap, the present study experimentally investigates a diesel–ethanol partial HCCI combustion strategy over a wide range of engine speeds and loads using four different ethanol port injection ratios. In this configuration, ethanol was injected into the intake port to form a homogeneous ethanol–air mixture before entering the cylinder, whereas diesel fuel was directly injected near the end of the compression stroke to initiate combustion. The experimental results were systematically compared with those of conventional diesel combustion in terms of combustion characteristics, engine performance, and exhaust emissions. The novelty of this study lies in providing a comprehensive experimental assessment of ethanol port injection under realistic operating conditions covering both variable engine speed and load, thereby offering a broader understanding of the applicability and limitations of ethanol-assisted partial HCCI combustion than previous studies conducted under limited operating conditions.

2. Materials and Methods

2.1. Ethanol Fumigation

The engine was first operated with neat diesel under different load conditions to determine the baseline diesel fuel consumption. Ethanol corresponding to 5%, 10%, 15%, and 20% of the diesel fuel mass flow rate was supplied through a secondary electronically controlled injector mounted on the intake manifold. After port injection, the ethanol was mixed homogeneously with the intake air before entering the cylinder as a premixed charge, while combustion was initiated by direct diesel injection near the end of the compression stroke. The ethanol injector was calibrated using a precision balance to determine the

injection duration and operating voltage required for each ethanol ratio. The ethanol injection timing was fixed at 5 °CA after top dead center (aTDC).

The properties of the fuels used in the experiments are shown in Table 1. The diesel fuel was commercial EN590 diesel fuel obtained from Shell Turkey; the ethanol fuel was supplied by the KİMETSAN company.

Table 1. Fuel properties.

Property	Unit	Diesel	Ethanol
Density (15 °C)	[g/cm ³]	0.82–0.86	0.78–0.79
Cetane number	-	49	11
Kinematic viscosity (40 °C)	[mm ² /s]	2.5–3.5	1.1
Flash point	[°C]	>55	15
Lower heating value	[kJ/kg]	42,640	27,423

2.2. Engine Test Setup

Experimental investigations were conducted on a Super Star make, DI, water-cooled, single-cylinder diesel engine. Test engine technical specifications are given in Table 2. To maintain stable operation conditions, the test engine is run for a while. During experiments, cooling water and lubrication oil operation temperatures were kept constant at 85 °C by additional conditioning systems for water and oil. Loads were applied to the test engine by a 50 kg-capacity hydraulic dynamometer. And engine brake power was measured by an S-type load cell with 0.1 kg precision, mounted on the load arm. All the temperatures were measured by NiCr-Ni-type thermocouples. Prior to the main experimental campaign, a series of preliminary tests was performed to identify the most suitable diesel injection timing for partial HCCI operation. The optimization was based on the combined evaluation of engine performance, combustion characteristics, and exhaust emissions. The results indicated that an injection timing of 29 °CA aTDC achieved the best compromise between brake thermal efficiency, fuel consumption, combustion stability, and emission performance. Consequently, this timing was adopted as the reference injection timing throughout the study. Emissions were measured by a Bosch BEA 060 (Bosch, Bursa, Türkiye) exhaust gas analyzer. Fuel consumption was measured by a volumetric gauge. Air consumption was measured by a mass flowmeter and a surge tank. Experiments were performed at 1400, 1600, 1800, 2000, 2200, and 2400 rpm engine speeds for 40%, 60%, 80%, and full load conditions. Test engine setup is illustrated in Figure 1.

Table 2. Technical specifications of the test engine.

Category	Specification	Value	Unit
General information	Brand/Model	Super Star	-
	Engine type	1-cylinder diesel	-
Dimensions	Cylinder bore	108	[mm]
	Piston Stroke	100	[mm]
	Stroke volume	0.92	[dm ³]
Performance	Maximum power	13	[kW @ 2200 rpm]
	Maximum speed	2500	[rpm]
	Compression ratio	17	-

Table 2. Cont.

Category	Specification	Value	Unit
Fuel system	Fuel type	Diesel	-
	Injection type	Direct injection	-
	Injection timing	29	[°CA]
	Injector opening pressure	225	[bar]
Cooling system	Cooling type	Water cooling	-

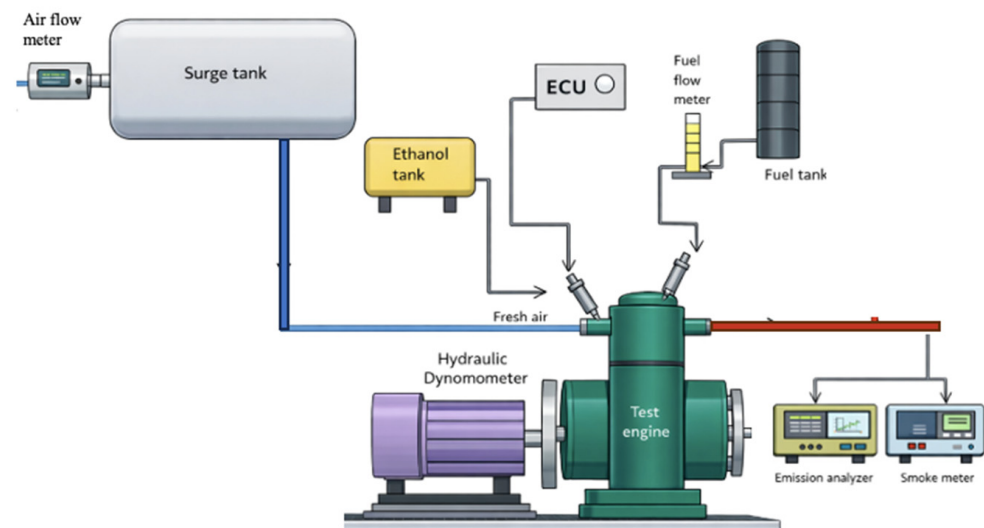


Figure 1. Engine test setup.

2.3. Design of Experiments and Data Analysis

The experimental procedure began with baseline tests using neat diesel fuel to obtain the performance and emission characteristics of the conventional diesel engine. Subsequently, ethanol fumigation experiments were performed at ethanol ratios of 5%, 10%, 15%, and 20%, while the diesel fuel quantity was adjusted to maintain the same engine power as the baseline condition. Full-load tests were conducted at engine speeds of 1400, 1600, 1800, 2000, 2200, and 2400 rpm. Partial-load tests were performed at 1600 and 2400 rpm, corresponding to the engine speeds at which the maximum brake torque (MBT) and maximum brake power (MBP) were achieved. The experiments were carried out following the experimental design, with 3 repetitions for each test condition. The standard deviations of the repeated measurements were small, indicating good repeatability of the experimental results.

Based on the data obtained from the engine experiments, brake torque (BT), brake power (BP), brake-specific fuel consumption (BSFC), and brake thermal efficiency (BTE) values were calculated. Emissions were measured during the experiments using a Bosch BEA 060 emission measurement device. The measurement uncertainties of NO_x , HC, CO, and CO_2 were ± 5 ppm, ± 12 ppm, $\pm 0.055\%$, and $\pm 0.5\%$, respectively, according to the manufacturer's specifications. Before the experiments, all measuring instruments were calibrated in accordance with the manufacturer's instructions. Table 3 summarizes the measuring instruments, measurement ranges, and accuracies used in the experiments. Each operating condition was tested three times under identical conditions, and the average values were used for the analyses to ensure the repeatability and reliability of the experimental results.

Table 3. Measurement systems and ranges.

Measured Parameters	Unit	Measurement Range	Measuring Instrument	Measurement Accuracy
Engine speed	[rpm]	0–9999	Digital tachometer	1
Exhaust temperature	[°C]	0–1000	Ni-Cr-Ni	1
Brake balance force	[kg]	0–100	Water brake	0.1
NO emissions	[ppm]	0–2000	Bosch BEA 060	±5
HC emissions	[ppm]	0–20,000	Bosch BEA 060	±12
CO emissions	[%]	% 0–15.0	Bosch BEA 060	±0.055
CO ₂ emissions	[%]	% 0–20.0	Bosch BEA 060	0.5–5

2.4. Uncertainty Analysis

During the experiments, despite the considerable care taken, systematic errors associated with the instrumentation may occur. Potential sources of error are identified and minimized.

According to the uncertainty analysis, let R be the measured quantity in the system. Let the n independent variables affecting this quantity be denoted as $x_1, x_2, x_3, \dots, x_n$. In this case, R can be expressed by Equation (1).

$$R = R(x_1, x_2, x_3, \dots, x_n) \quad (1)$$

The uncertainty ratios associated with each independent variable are denoted as $w_1, w_2, w_3, \dots, w_n$, while the uncertainty ratio of the quantity R is denoted as w_R , as shown in Equation (2), [39].

$$w_R = \left[\left(\frac{\partial R}{\partial x_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial x_2} w_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial x_n} w_n \right)^2 \right]^{1/2} \quad (2)$$

Uncertainty results of calculated parameters in the experiments are given in Table 4. Uncertainty results are calculated for the engine running under the 40%-E0-1600 rpm condition.

Table 4. Uncertainty analysis results.

Parameter, Calculated	Denoted by	Unit	Value	Uncertainty [%]
Brake torque	BT	[Nm]	0.1658	1
Brake power	BP	[kW]	0.02792	1
Brake-specific fuel consumption	BSFC	[g/kWh]	4.524	1.41
Brake thermal efficiency	BTE	[%]	0.373	1.41

The relative uncertainties of the calculated parameters were found to be between 1.0% and 1.41%, indicating a high level of measurement accuracy and good reliability of the experimental results. These uncertainty levels are sufficiently low for engine performance and emission measurements and are considered acceptable for the present study.

3. Results and Discussion

This study comprehensively investigates the effects of different ethanol ratios (5%, 10%, 15%, and 20%) and varying load conditions (100%, 80%, 60%, and 40%) on engine performance, combustion characteristics, and emissions in a single-cylinder diesel engine operating under partial HCCI combustion mode.

3.1. Effects of Ethanol Fumigation on Engine Parameters

In Figure 2, the variation of BSFC with engine speed is presented for different ethanol ratios in a partially HCCI diesel engine under full-load conditions. It is observed that BSFC values increase with increasing engine speed for all fuel types. For neat diesel fuel, the BSFC reaches its maximum value of approximately 295 g/kWh at 2400 rpm, whereas these maximum values decrease with the addition of ethanol. When the maximum BSFC values obtained at 2400 rpm are examined, they are approximately 278, 273, 270, and 268 g/kWh for E5, E10, E15, and E20 fuels, respectively. Compared to neat diesel fuel, these correspond to reductions of about 5.8%, 7.5%, 8.5%, and 9.2% in maximum BSFC, respectively.

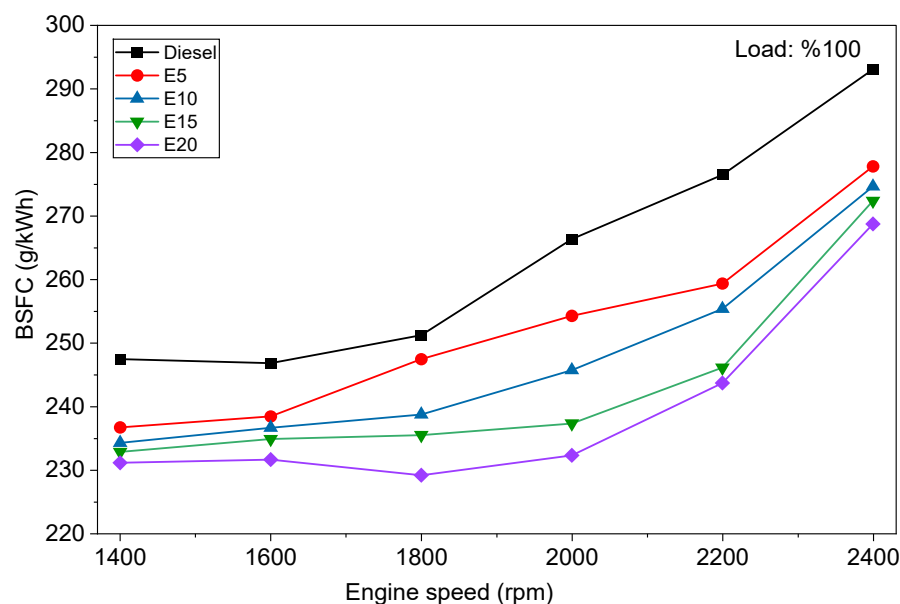


Figure 2. Variation of BSFC under full load conditions for different ethanol ratios in a partial HCCI diesel engine.

The increase in BSFC with rising engine speed is associated with the shortening of combustion duration at higher speeds and the incomplete progression of in-cylinder reactions. The observed reduction in maximum BSFC with ethanol addition can be explained by the oxygen content of ethanol, which enhances combustion efficiency, its high volatility that promotes a more homogeneous air–fuel mixture, and the improvement of the premixed charge in the partial HCCI combustion mode. Furthermore, the more pronounced decrease in maximum BSFC with increasing ethanol ratio indicates a more effective combustion process, particularly at higher ethanol contents. Although the lower heating value of ethanol would theoretically tend to increase fuel consumption, the improved combustion characteristics in this study appear to outweigh this effect.

In Figure 3, the variation of BTE with engine speed is presented for different ethanol ratios in a partially HCCI diesel engine under full-load conditions. The BTE exhibits a peak between 1400 and 1600 rpm for the E5 and E10 fuels, after which it decreases with further increases in engine speed. Similarly, for the E15 and E20 fuels, the BTE reaches its peak between 1400 and 2000 rpm and then declines beyond 2000 rpm. For neat diesel fuel, the maximum BTE is obtained at approximately 1600 rpm with a value of about 34.2%, whereas this maximum value increases significantly with the addition of ethanol. When the maximum BTE values are examined, they reach approximately 36.0%, 38.0%, 39.5%, and 39.8% for E5, E10, E15, and E20 fuels, respectively. Compared to neat diesel fuel, these correspond to increases of about 5.3%, 11.1%, 15.5%, and 16.4% in maximum BTE, respectively.

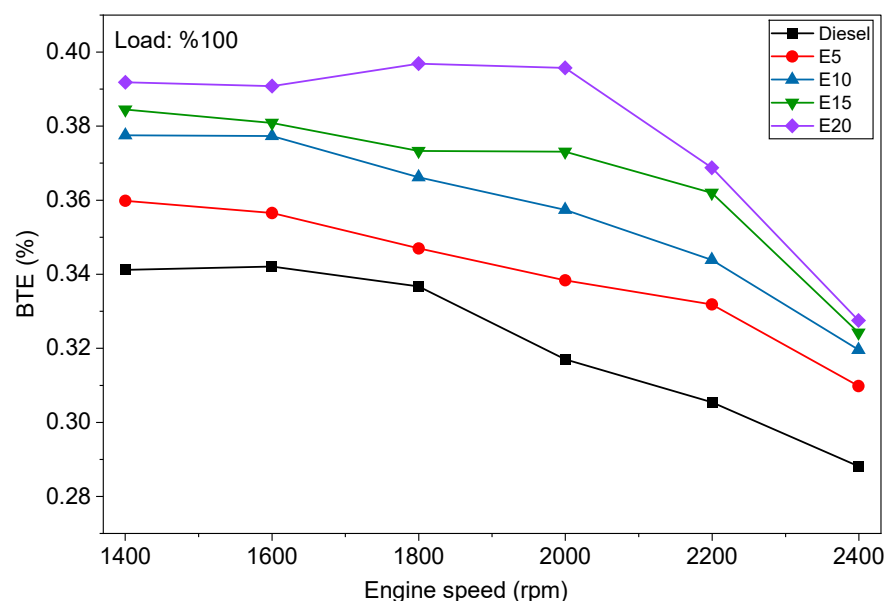


Figure 3. Variation of BTE under full load conditions for different ethanol ratios in a partial HCCI diesel engine.

The decrease in BTE with increasing engine speed is associated with the shortening of combustion duration at higher speeds, increased friction losses, and higher heat transfer losses. The observed increase in maximum BTE with ethanol addition can be attributed to the oxygen content of ethanol, which promotes more efficient combustion, its high volatility that enables the formation of a more homogeneous air–fuel mixture, and the improvement of premixed charge quality in the partial HCCI combustion mode. Furthermore, the more pronounced improvement in maximum BTE with increasing ethanol ratio indicates a faster and more complete combustion process, particularly at higher ethanol contents. However, the reduction of this advantage at higher engine speeds can be explained by limited combustion duration and increased heat losses, which reduce overall combustion efficiency.

3.2. Effects of Ethanol Fumigation on In-Cylinder Pressure and Heat Release Rate

In Figure 4, the variations of in-cylinder pressure and heat release rate (HRR) with crank angle are presented for different ethanol ratios at 1600 rpm and 2400 rpm under 100% load conditions in a partially HCCI diesel engine. At both engine speeds, it is observed that the maximum in-cylinder pressure increases with the addition of ethanol. At 1600 rpm, while the maximum cylinder pressure for neat diesel fuel is around 80 bar, this value rises to approximately 82, 85, 87, and 89 bar for E5, E10, E15, and E20 fuels, respectively. Similarly, at 2400 rpm, the maximum pressure is about 68 bar for neat diesel, increasing to approximately 70, 73, 76, and 79 bar for E5, E10, E15, and E20 fuels, respectively.

When the HRR curves are examined, it is evident that ethanol addition leads to significant increases, particularly during the premixed combustion phase. At 1600 rpm, the maximum HRR is approximately 65 J/°CA for neat diesel fuel, whereas it reaches approximately 75, 85, 95, and 100 J/°CA for E5, E10, E15, and E20 fuels, respectively. At 2400 rpm, the maximum HRR is about 55 J/°CA for neat diesel, increasing to approximately 65, 75, 85, and 92 J/°CA for E5, E10, E15, and E20 fuels, respectively.

These increases in in-cylinder pressure and HRR can be attributed to the high volatility of ethanol and its inherent oxygen content, which facilitate the formation of a more homogeneous air–fuel mixture. In the partial HCCI operating mode, ethanol supplied through the intake port improves the premixed charge, extends the ignition delay, and

allows a greater portion of the fuel to burn in premixed mode, resulting in higher heat release. Consequently, higher peak pressure values are obtained. Furthermore, the sharper HRR curves and higher peak values observed with increasing ethanol ratio indicate an increase in combustion rate and the dominance of premixed combustion characteristics. At higher engine speed (2400 rpm), the combustion event occurs over a narrower crank angle interval, and the curves exhibit a sharper profile; this is associated with increased turbulence and reduced combustion duration.

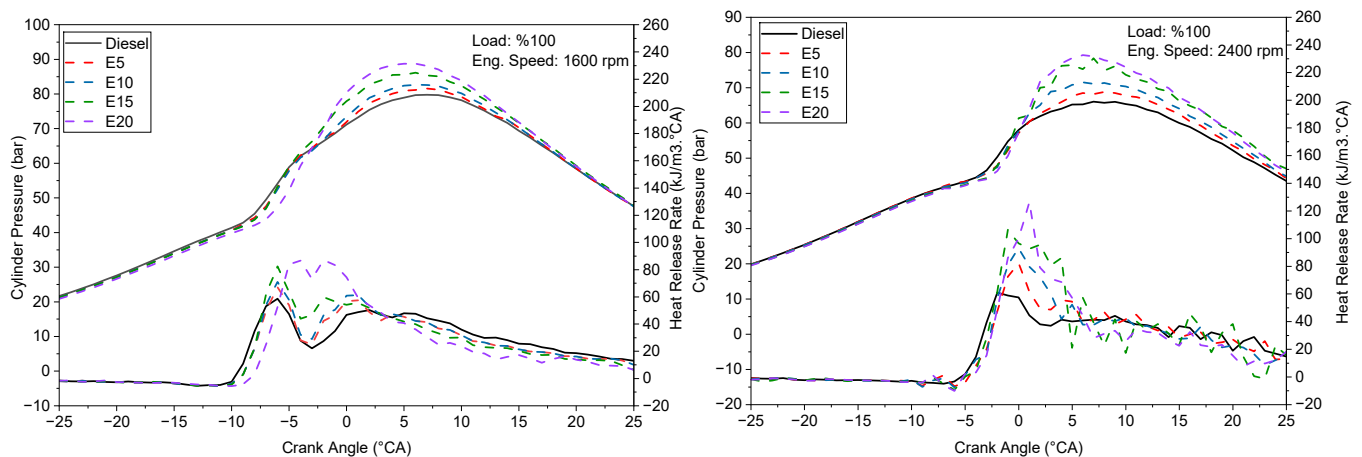


Figure 4. Variation of cylinder pressure and HRR under 100% full load conditions for different ethanol ratios in a partial HCCI diesel engine.

In Figure 5, the variations of in-cylinder pressure and HRR with crank angle are evaluated for different ethanol ratios at 1600 rpm and 2400 rpm under 80% load conditions in a partially HCCI diesel engine. Although the general trend is similar to that observed under 100% load, it is characterized by lower peak values. At 1600 rpm, the maximum in-cylinder pressure for neat diesel fuel occurs at relatively lower levels, while a gradual increase is observed with ethanol addition. For E5, E10, E15, and E20 fuels, the maximum pressure values reach higher levels compared to diesel fuel and tend to shift toward earlier crank angles.

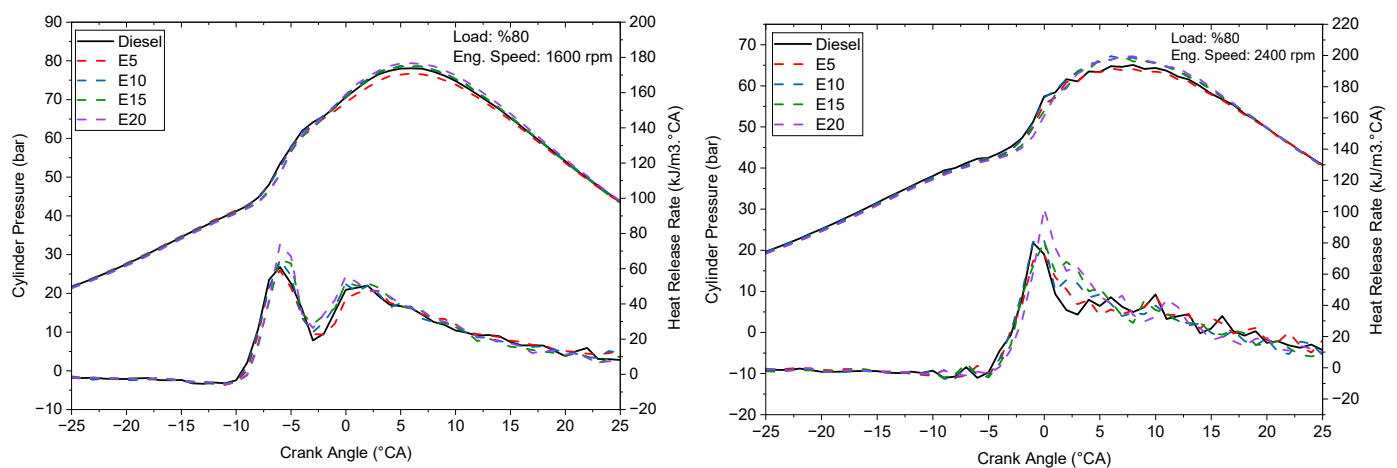


Figure 5. Variation of cylinder pressure and HRR under 80% partial load conditions for different ethanol ratios in a partial HCCI diesel engine.

A similar behavior is observed at 2400 rpm, where the maximum pressure of neat diesel remains lower than that of the ethanol-blended fuels, and the peak pressure increases with increasing ethanol ratio. However, due to the higher engine speed and lower load

condition, the maximum pressure values remain more limited compared to the 100% load case. In addition, at higher engine speed, the combustion event occurs within a narrower crank angle interval, and the pressure curves exhibit a sharper profile.

Examination of the HRR curves indicates that the premixed combustion phase becomes more pronounced with ethanol addition under 80% load conditions. At 1600 rpm, neat diesel fuel exhibits a lower and broader HRR curve, whereas ethanol-containing fuels produce higher and sharper peaks. In particular, E15 and E20 fuels show more distinct HRR profiles, with the maximum heat release shifting toward earlier crank angles. Similarly, at 2400 rpm, the HRR curves occur over a narrower range with sharper characteristics, and the maximum heat release values increase with increasing ethanol ratio.

These increases in in-cylinder pressure and HRR can be explained by the high volatility of ethanol and its inherent oxygen content, which promote the formation of a more homogeneous air–fuel mixture. Under 80% load conditions, the reduction in the total amount of fuel inducted into the cylinder leads to lower combustion temperatures and pressures; however, ethanol fumigation partially compensates for this disadvantage by improving combustion quality. Furthermore, the extension of ignition delay due to ethanol allows a greater fraction of the fuel to burn in premixed mode, which is reflected as more pronounced peak values in the HRR curves. Nevertheless, under low-load conditions, the leaner mixture and the possibility of incomplete combustion in certain regions are considered limiting factors for combustion stability.

In Figure 6, the variations of in-cylinder pressure and HRR with crank angle are presented for different ethanol ratios at 1600 rpm and 2400 rpm under 60% load conditions in a partially HCCI diesel engine. With decreasing load, it is observed that the maximum in-cylinder pressure values at both engine speeds occur at lower levels compared to the previous load conditions. At 1600 rpm, the maximum cylinder pressure for neat diesel fuel is around 76 bar, while it increases to approximately 77, 78, 78.5, and 79 bar for E5, E10, E15, and E20 fuels, respectively. At 2400 rpm, the maximum pressure for neat diesel is about 64 bar, increasing to approximately 65, 66, 67, and 68 bar for E5, E10, E15, and E20 fuels, respectively.

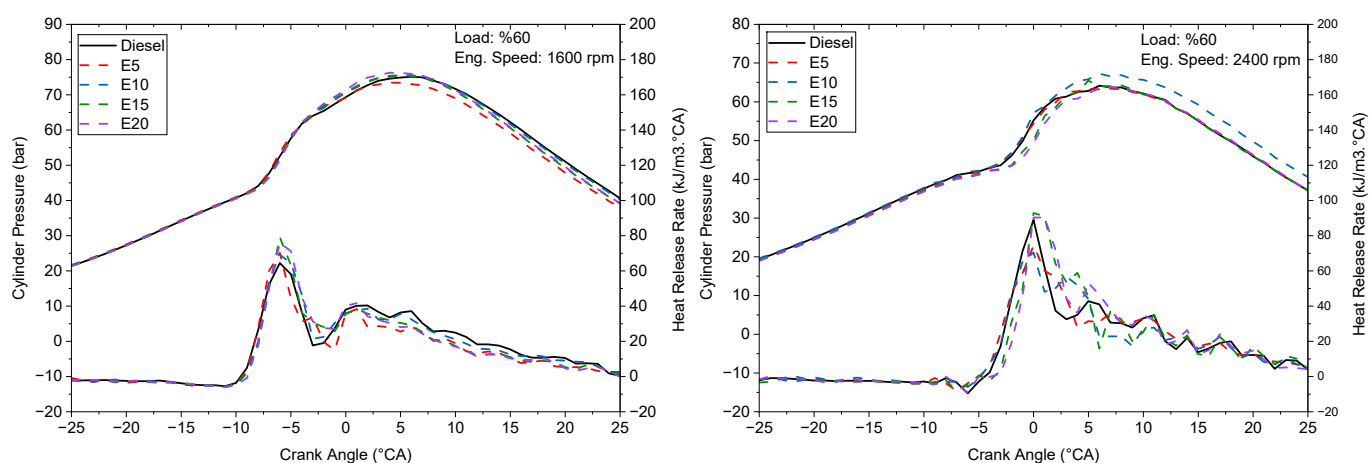


Figure 6. Variation of cylinder pressure and HRR under 60% partial load conditions for different ethanol ratios in a partial HCCI diesel engine.

Examination of the HRR curves indicates that, under 60% load conditions, ethanol addition makes the premixed combustion phase more pronounced, although the peak values remain relatively limited. At 1600 rpm, the maximum HRR for neat diesel fuel is approximately 55 J/°CA, whereas it reaches approximately 60, 65, 70, and 75 J/°CA for E5, E10, E15, and E20 fuels, respectively. At 2400 rpm, the maximum HRR is about 50 J/°CA

for neat diesel, increasing to approximately 55, 60, 65, and 70 J/°CA with ethanol addition. Furthermore, with an increasing ethanol ratio, the HRR curves become sharper, and the maximum heat release shifts toward earlier crank angles.

These increases in in-cylinder pressure and HRR can be explained by the high volatility of ethanol and its inherent oxygen content, which promote the formation of a more homogeneous mixture and increase the proportion of premixed combustion in partial HCCI mode. However, as the load decreases to 60%, the reduction in the amount of fuel inducted into the cylinder leads to lower combustion temperatures and pressures, thereby limiting the maximum pressure and HRR values. In addition, under lower load conditions, the mixture becomes leaner, and the incomplete progression of reactions in localized combustion regions results in increased fluctuations in the HRR curves and relatively reduced combustion stability. Nevertheless, the sharper and higher peak values obtained with ethanol addition indicate that combustion quality is improved even under low-load conditions.

Figure 7 presents the variation of in-cylinder pressure and HRR with crank angle for different ethanol ratios at 1600 rpm and 2400 rpm under 40% load conditions in a partial HCCI diesel engine. With the further reduction in load, it is observed that the maximum in-cylinder pressure values at both engine speeds are lower compared to higher load conditions. At 1600 rpm, the maximum cylinder pressure for neat diesel fuel is approximately 72 bar, while for E5, E10, E15, and E20 fuels, these values increase to about 73, 74, 75, and 76 bar, respectively. Similarly, at 2400 rpm, the maximum pressure for neat diesel is around 62 bar, whereas with ethanol addition, the values rise to approximately 63, 64, 65, and 66 bar for E5, E10, E15, and E20, respectively.

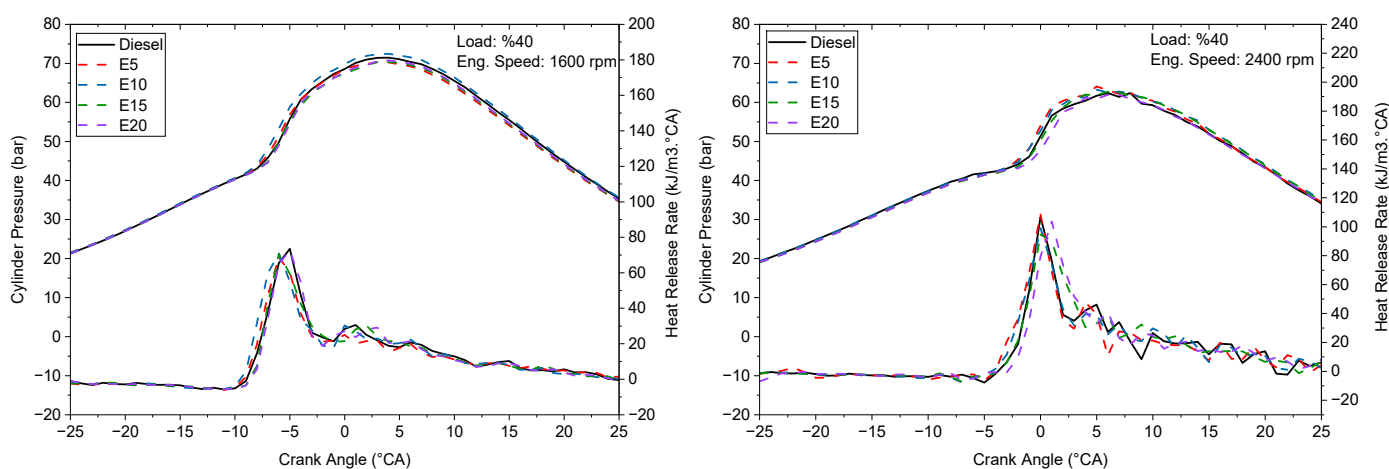


Figure 7. Variation of cylinder pressure and HRR under 40% partial load conditions for different ethanol ratios in a partial HCCI diesel engine.

Analysis of the HRR curves indicates that, under 40% load conditions, the premixed combustion phase becomes more pronounced, although the overall heat release is limited. At 1600 rpm, the maximum HRR value for neat diesel fuel is approximately 50 J/°CA, while for E5, E10, E15, and E20 fuels, these values increase to about 55, 60, 65, and 70 J/°CA, respectively. At 2400 rpm, the maximum HRR for neat diesel is around 45 J/°CA, whereas with ethanol addition, it rises to approximately 50, 55, 60, and 65 J/°CA for the respective blends. Furthermore, as the ethanol ratio increases, the HRR curves become sharper, and the peak heat release shifts closer to top dead center.

These results can be explained by the reduced amount of fuel supplied to the cylinder under low load conditions, leading to lower combustion temperatures and pressures. Ethanol addition, on the other hand, enhances mixture homogeneity due to its high volatility and inherent oxygen content, thereby strengthening the premixed combustion phase.

In partial HCCI operation, the extended ignition delay associated with ethanol allows a larger fraction of the fuel to burn in the premixed mode, resulting in sharper and higher HRR peaks. However, under low load conditions, the formation of very lean mixtures and reduced in-cylinder temperatures limit combustion stability and lead to increased fluctuations in the HRR profiles. Nevertheless, the improvement in combustion characteristics with ethanol addition remains clearly evident even under low load conditions.

When the results obtained under different load conditions (100%, 80%, 60%, and 40%) are evaluated collectively, it is evident that ethanol fumigation has a pronounced and systematic improving effect on the combustion characteristics of a partial HCCI diesel engine. With increasing load, in-cylinder maximum pressure and HRR values increase, while ethanol addition enhances these parameters to higher levels compared to neat diesel under all load conditions. Particularly at higher loads (100% and 80%), the high volatility and inherent oxygen content of ethanol promote the formation of a more homogeneous air–fuel mixture, strengthening the premixed combustion phase and resulting in higher peak pressures and sharper HRR profiles. Under medium and low load conditions (60% and 40%), although maximum pressure and HRR values decrease due to the reduced amount of fuel supplied to the cylinder, ethanol addition continues to improve combustion quality and makes the premixed combustion phase more pronounced.

With an increasing ethanol ratio, the start of combustion shifts closer to TDC across all load conditions, and HRR curves become more concentrated within a narrower crank angle interval. This behavior is attributed to the extended ignition delay caused by ethanol, allowing a larger portion of the fuel to burn in the premixed phase. However, under low load conditions, the formation of leaner mixtures and reduced in-cylinder temperatures limit combustion stability and lead to increased fluctuations in HRR profiles. Despite these limitations, the higher heat release rates and increased in-cylinder pressure values achieved with ethanol fumigation across all load ranges clearly demonstrate that ethanol is an effective additive for enhancing combustion efficiency in partial HCCI mode. In this context, ethanol utilization offers significant advantages in terms of engine performance and combustion effectiveness, particularly under high load conditions.

Figures 4–7 collectively illustrate the influence of ethanol fumigation on in-cylinder pressure and heat release rate (HRR) under different engine loads and engine speeds. In general, increasing engine load resulted in higher peak cylinder pressures and HRR values due to the greater amount of fuel energy released during combustion and the associated increase in in-cylinder temperature. Ethanol addition further intensified these effects by modifying the ignition and combustion processes. Owing to its lower cetane number, ethanol prolonged the ignition delay period, allowing more time for fuel–air mixing before the start of combustion. The extended premixing process increased the amount of combustible mixture available during the premixed combustion phase, leading to higher heat release rates and elevated peak cylinder pressures. As the ethanol ratio increased, the HRR profiles became sharper and more concentrated around top dead center, indicating a more rapid energy release process. These effects were particularly pronounced under high-load conditions (100% and 80%), where higher in-cylinder temperatures promoted efficient combustion and stronger pressure development. Under lower load conditions (60% and 40%), although the overall pressure and HRR levels decreased due to reduced fuel supply and lower combustion temperatures, ethanol continued to enhance premixed combustion and improve combustion phasing. The combined effects of prolonged ignition delay, improved mixture homogeneity, enhanced premixed heat release, and increased oxygen availability contributed to the improved combustion characteristics observed with ethanol fumigation. Overall, the results demonstrate that ethanol fumigation consistently

improves combustion quality in partial HCCI operation, with the most significant benefits observed under medium and high load conditions.

3.3. Effects of Ethanol Fumigation on Emission Characteristics

Figure 8 presents the variation of NO_x and HC emissions with engine speed for different ethanol ratios under full load (100%) conditions in a partial HCCI diesel engine. As observed from the figure, NO_x emissions generally exhibit a decreasing trend with increasing engine speed for all fuel types. For neat diesel fuel, the NO_x emission reaches a maximum value of approximately 1400 ppm at around 1600 rpm and decreases to about 900 ppm at 2400 rpm. With the addition of ethanol, NO_x emissions increase across all engine speeds, with maximum values reaching approximately 1450, 1500, 1550, and 1650 ppm for E5, E10, E15, and E20, respectively. It is evident that the increase in NO_x emissions becomes more pronounced with higher ethanol content.

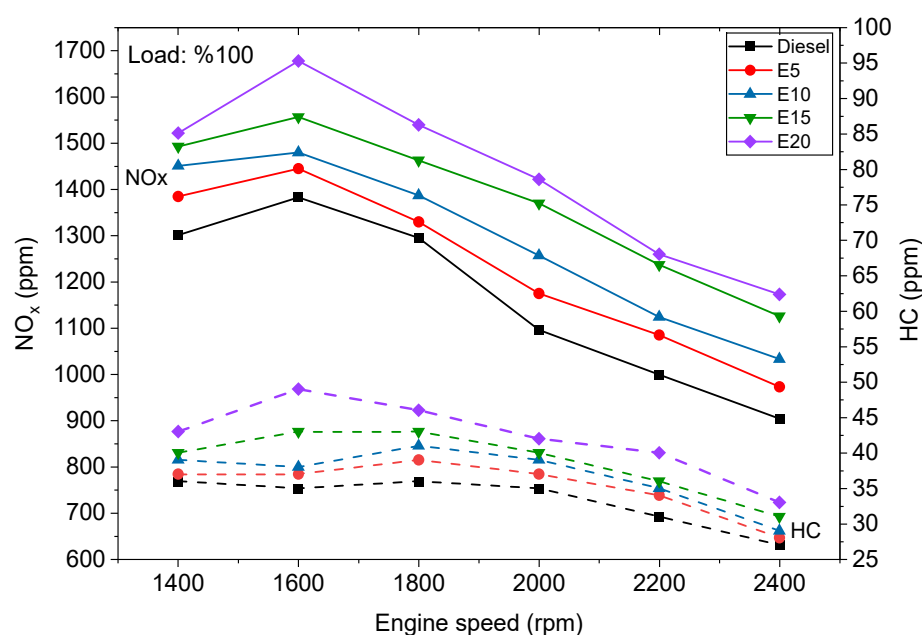


Figure 8. Variation of NO_x and HC emissions under full load conditions for different ethanol ratios in a partial HCCI diesel engine.

For all fuel types, HC emissions exhibit a decreasing trend with increasing engine speed. For neat diesel fuel, HC emissions decrease from approximately 35 ppm at 1400 rpm to about 25 ppm at 2400 rpm. However, with ethanol addition, HC emissions are higher compared to diesel fuel, particularly at low and middle engine speeds. The maximum HC values are observed to be approximately 40, 45, 50, and 55 ppm for E5, E10, E15, and E20 fuels, respectively.

The increase in NO_x emissions with ethanol addition can be attributed to the oxygen content of ethanol and the improved air–fuel mixing, which enhance combustion temperature and reaction rates. In partial HCCI operation, the strengthening of the premixed combustion phase due to ethanol leads to higher in-cylinder peak temperatures, thereby promoting thermal NO_x formation. On the other hand, the decrease in NO_x emissions with increasing engine speed is associated with shorter combustion duration and reduced residence time at high temperatures. The increase in HC emissions is mainly attributed to the high latent heat of vaporization of ethanol, which reduces in-cylinder temperatures and leads to incomplete combustion, especially in low-temperature regions and under low-load conditions. Additionally, the presence of overly lean mixture zones and flame quenching effects inherent in partial HCCI combustion contribute to higher unburned HC emissions.

Nevertheless, increasing engine speed enhances turbulence intensity and improves mixture formation, which, in turn, contributes to the reduction of HC emissions.

Figure 9 illustrates the variation of CO₂ and CO emissions with engine speed for different ethanol ratios under full load (100%) conditions in a partial HCCI diesel engine. As observed from the figure, CO₂ emissions exhibit an increasing trend with rising engine speed for all fuel types. For neat diesel fuel, CO₂ emissions increase from approximately 8.8% at 1400 rpm to about 11% at 2400 rpm. With the addition of ethanol, CO₂ emissions are consistently lower compared to diesel fuel, with values generally ranging between 7.5% and 9% for E5, E10, E15, and E20 blends. It is evident that CO₂ emissions gradually decrease as the ethanol ratio increases.

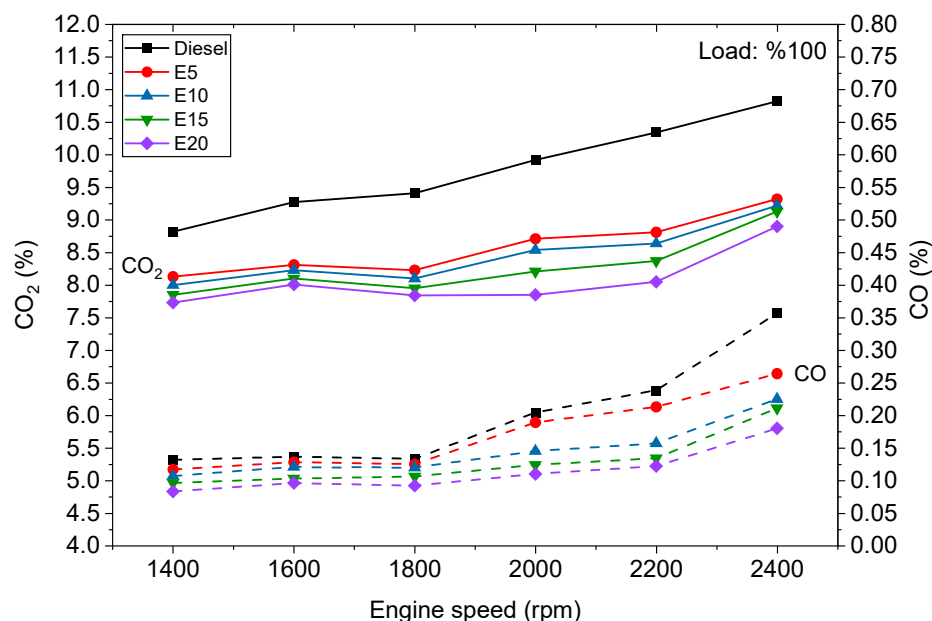


Figure 9. Variation of CO₂ and CO emissions under full load conditions for different ethanol ratios in a partial HCCI diesel engine.

Similarly, CO emissions show a noticeable increase with increasing engine speed for all fuels. For neat diesel fuel, CO emissions rise from approximately 0.10% at 1400 rpm to about 0.35% at 2400 rpm. However, with ethanol addition, CO emissions are lower than those of diesel fuel and progressively decrease with increasing ethanol content. The lowest CO emission levels are typically observed for the E20 blend, reaching approximately 0.20% at 2400 rpm.

The increase in CO₂ emissions with engine speed can be attributed to higher fuel consumption and enhanced oxidation of carbon at elevated operating conditions. On the other hand, the reduction in CO₂ emissions with ethanol addition is primarily due to the lower carbon content and reduced carbon-to-hydrogen ratio of ethanol compared to diesel fuel. Although the oxygen content of ethanol promotes the conversion of carbon to CO₂ rather than CO during combustion, the overall lower carbon availability leads to reduced total CO₂ emissions. The increase in CO emissions at higher engine speeds is associated with shorter combustion duration and incomplete oxidation reactions. Nevertheless, the oxygenated nature of ethanol improves the combustion process and enhances carbon oxidation, thereby contributing to lower CO emissions. The more pronounced reduction in CO emissions with increasing ethanol content indicates improved mixture homogeneity and more efficient combustion characteristics under partial HCCI operating conditions.

Figure 10 presents the variation of NO_x and HC emissions with load conditions for different ethanol ratios at engine speeds of 1600 rpm and 2400 rpm in a partial HCCI diesel

engine. At both engine speeds, NO_x emissions increase significantly with increasing load. At 1600 rpm, the NO_x emission for neat diesel fuel starts at approximately 700 ppm at low load and reaches about 1400 ppm at full load. With ethanol addition, these values increase further, reaching approximately 1500, 1600, 1700, and 1750 ppm at full load for E5, E10, E15, and E20, respectively. Similarly, at 2400 rpm, the NO_x emission for neat diesel is around 600 ppm at low load and rises to approximately 900 ppm at full load, while with ethanol addition, the values increase to about 1000, 1100, 1200, and 1300 ppm for E5, E10, E15, and E20, respectively.

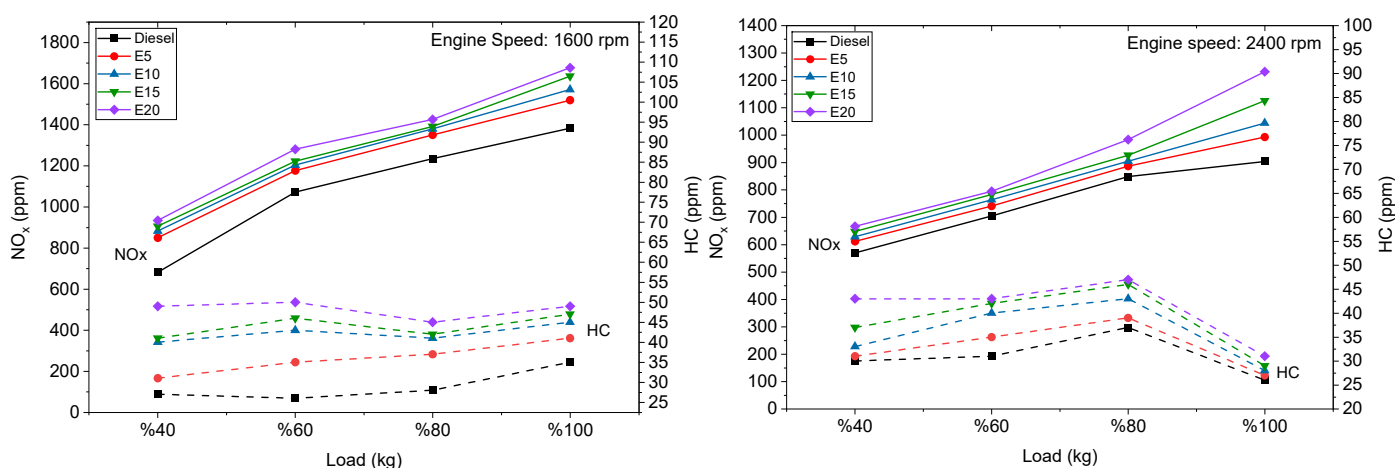


Figure 10. Variation of NO_x and HC emissions under full and partial load conditions for different ethanol ratios in a partial HCCI diesel engine.

In terms of HC emissions, a different trend is observed, where emissions initially increase with load and then decrease under high load conditions at both engine speeds. At 1600 rpm, HC emissions for neat diesel fuel are approximately 30 ppm at low load, increase to about 35–40 ppm at medium loads, and then decrease again to around 30 ppm at full load. With ethanol addition, HC emissions are consistently higher across all load conditions, with the effect being more pronounced at medium loads. The maximum HC values for E5, E10, E15, and E20 are approximately 40, 45, 50, and 55 ppm, respectively. A similar trend is observed at 2400 rpm, where HC emissions peak at medium loads and then decrease noticeably at full load. Additionally, at 2400 rpm, HC emissions are generally lower than those at 1600 rpm for all load conditions as the ethanol ratio increases.

The increase in NO_x emissions with increasing load can be attributed to the higher amount of fuel supplied to the cylinder, leading to elevated combustion temperatures. The oxygen content of ethanol and the improved mixture homogeneity enhance the combustion rate, increasing peak temperatures and thereby promoting NO_x formation, especially under high load conditions. The variation in HC emissions is closely related to load-dependent combustion characteristics. At low and medium loads, leaner mixtures and lower in-cylinder temperatures result in incomplete combustion, whereas at high loads, increased temperatures and improved oxidation conditions lead to a reduction in HC emissions. Additionally, the high latent heat of vaporization and ignition delay effect of ethanol contribute to increased unburned HC formation, particularly at medium load conditions. Overall, it can be concluded that ethanol addition has a significant impact on NO_x and HC emissions, and this effect varies depending on engine load and speed.

Figure 11 presents the variation of CO_2 and CO emissions with load conditions for different ethanol ratios at engine speeds of 1600 rpm and 2400 rpm in a partial HCCI diesel engine. At both engine speeds, CO_2 emissions increase significantly with increasing load for all fuel types. At 1600 rpm, CO_2 emissions for neat diesel fuel start at approximately

4% under low load and reach about 9.5% at full load. With ethanol addition, these values remain lower, reaching approximately 8.5%, 8.2%, 8.0%, and 7.8% at full load for E5, E10, E15, and E20, respectively. Similarly, at 2400 rpm, CO₂ emissions for neat diesel are around 5% at low load and increase to approximately 11% at full load, whereas with ethanol addition, the values decrease to about 9.5%, 9.2%, 9.0%, and 8.8% for E5, E10, E15, and E20, respectively.

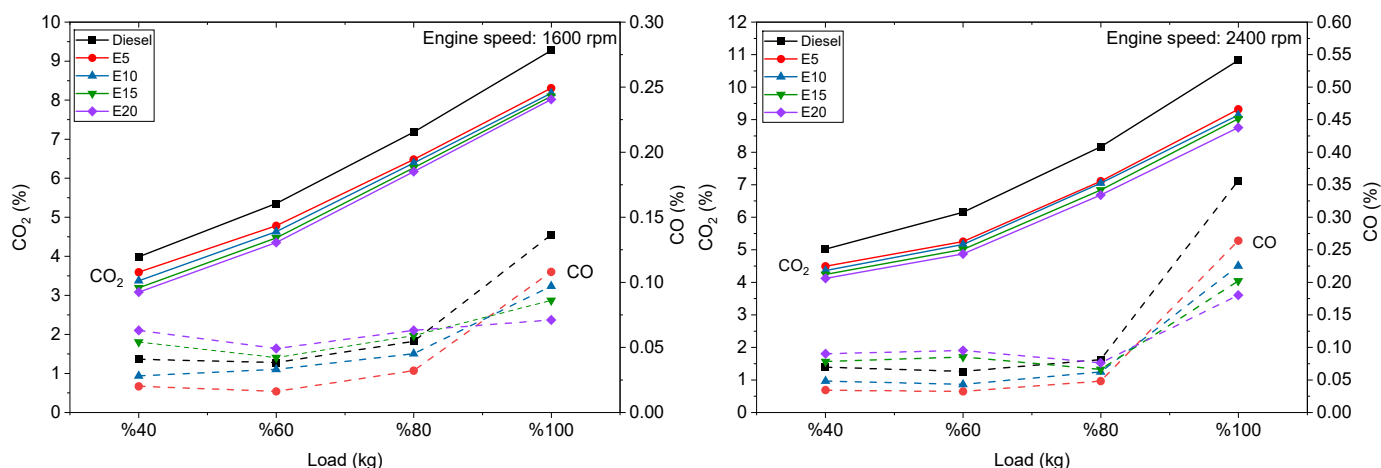


Figure 11. Variation of CO₂ and CO emissions under full and partial load conditions for different ethanol ratios in a partial HCCI diesel engine.

In terms of CO emissions, an overall increasing trend with load is observed at both engine speeds. At 1600 rpm, CO emissions for neat diesel fuel increase from approximately 0.10% at low load to about 0.15% at full load. With ethanol addition, CO emissions are consistently lower across all load conditions, with the difference becoming more pronounced at higher loads. At full load, CO emissions for E5, E10, E15, and E20 are approximately 0.12%, 0.11%, 0.10%, and 0.09%, respectively. At 2400 rpm, CO emissions remain relatively low under low load conditions but increase significantly at full load, reaching approximately 0.35% for neat diesel. With ethanol addition, these values decrease to about 0.25%, 0.22%, 0.20%, and 0.18% for E5, E10, E15, and E20, respectively.

The increase in CO₂ emissions with load is attributed to the higher amount of fuel supplied to the cylinder, resulting in greater total carbon oxidation during combustion. However, the reduction in CO₂ emissions with ethanol addition can be explained by the lower carbon content and reduced carbon-to-hydrogen ratio of ethanol compared to diesel fuel. Although the oxygen content of ethanol enhances combustion and promotes more complete oxidation of carbon, the overall lower carbon availability limits total CO₂ formation. The increase in CO emissions with load, particularly at high load and high engine speed conditions, is associated with shortened combustion duration and incomplete oxidation in certain regions. Nevertheless, the oxygenated nature of ethanol and its ability to promote more homogeneous mixture formation improve combustion efficiency and reduce CO formation. The more pronounced decrease in CO emissions with increasing ethanol ratio indicates improved mixture quality and enhanced combustion efficiency under partial HCCI operating conditions.

Although CO and HC emissions are both generally associated with incomplete combustion, their formation mechanisms differ considerably under partial HCCI operation. The reduction in CO emissions observed with ethanol addition indicates improved oxidation of carbon-containing intermediate species due to the oxygenated nature of ethanol and enhanced mixture preparation. In contrast, HC emissions are strongly influenced by local combustion conditions. The high latent heat of vaporization of ethanol can reduce local

temperatures and create over-lean regions where oxidation reactions cannot be completed. In addition, fuel trapped in crevice volumes and near-wall quenching zones may escape complete combustion and contribute to increased HC emissions. Therefore, lower CO emissions do not necessarily imply lower HC emissions, and the simultaneous decrease in CO and increase in HC emissions observed in the present study are consistent with the combustion characteristics of partial HCCI engines reported in the literature.

The results obtained in the present study are generally consistent with recently published studies on ethanol fumigation and partial HCCI combustion. Previous researchers have reported that ethanol addition improves premixed combustion, increases heat release rates, and enhances thermal efficiency due to its oxygenated nature and improved mixture homogeneity. Similar trends were observed in the current study, where ethanol fumigation increased in-cylinder pressure and HRR while reducing BSFC and improving BTE. Furthermore, the reduction in CO and CO₂ emissions observed in the present work agrees with earlier findings reported for ethanol-fueled HCCI and dual-fuel combustion systems. However, similar to previous studies, increases in NO_x and HC emissions were also observed due to higher combustion temperatures and the formation of locally lean or low-temperature combustion zones. Compared with the available literature, the present study provides a comprehensive evaluation of ethanol fumigation under a wide range of load conditions (40–100%) and engine speeds (1400–2400 rpm), offering additional insights into the combustion and emission behavior of partial HCCI diesel engines.

4. Conclusions

This study experimentally investigated the effects of ethanol fumigation on the performance, combustion characteristics, and emission behavior of a partial HCCI diesel engine under different engine speeds and load conditions. Based on the experimental findings, the following conclusions can be drawn:

- Ethanol fumigation improved engine performance under all operating conditions. The lowest BSFC and highest BTE values were obtained with the E20 blend, particularly under high-load conditions, indicating improved fuel utilization and combustion efficiency.
- The addition of ethanol enhanced the combustion process by increasing both in-cylinder pressure and heat release rate (HRR). At full load and 1600 rpm, the maximum cylinder pressure increased from approximately 80 bar for neat diesel to nearly 88 bar for E20, while the peak HRR increased from about 65 J/°CA to approximately 90 J/°CA.
- Increasing ethanol content promoted premixed combustion and shifted the main heat release phase closer to top dead center. As a result, combustion became more rapid and efficient, especially under medium and high load conditions.
- NO_x emissions increased with ethanol addition due to higher combustion temperatures and enhanced oxygen availability. At full load and 1600 rpm, NO_x emissions increased from approximately 1400 ppm for diesel fuel to around 1750 ppm for E20.
- HC emissions were generally higher for ethanol-containing fuels, particularly at low and medium loads. This increase was attributed to lower local temperatures, over-lean mixture regions, and incomplete oxidation processes associated with partial HCCI combustion.
- In contrast, ethanol fumigation reduced both CO and CO₂ emissions. At full load, CO emissions decreased from approximately 0.15% for diesel to 0.09% for E20 at 1600 rpm, while CO₂ emissions decreased from about 9.5% to 7.8%. Similar trends were observed at 2400 rpm.

- Among the investigated fuels, E20 provided the most favorable overall balance between combustion enhancement, thermal efficiency improvement, and emission reduction.

Overall, the findings demonstrate that ethanol fumigation is an effective strategy for improving combustion efficiency and engine performance in partial HCCI diesel engines. Among the investigated blends, E20 provided the most favorable overall performance, increasing brake thermal efficiency by approximately 12–15%, increasing peak cylinder pressure by nearly 10%, and reducing CO and CO₂ emissions by up to 40% and 20%, respectively, compared with neat diesel operation. These results highlight the practical potential of ethanol fumigation as a low-cost and easily applicable approach for improving combustion quality and reducing carbon-related emissions in existing diesel engines without major hardware modifications. However, the benefits of ethanol addition are not expected to increase indefinitely with increasing ethanol concentration. Excessively high ethanol fractions may result in prolonged ignition delay, combustion phasing retardation, and combustion instability, which could adversely affect engine efficiency and operational reliability. Furthermore, the present study was limited to ethanol ratios up to 20% and did not consider long-term engine durability, cold-start performance, cyclic combustion variations, or optimization of injection strategies. Therefore, future studies should focus on identifying the optimum ethanol ratio under a wider range of operating conditions and developing advanced combustion control techniques to maximize the benefits of ethanol-fueled partial HCCI combustion while mitigating NO_x and HC emissions.

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