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Experimental Investigation of Combustion, Performance, and Emissions Characteristics of Gasoline and Ethanol in a Spray-Guided Gasoline Direct-Injection Engine

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Abstract- The increasing demand for cleaner and more sustainable transportation fuels has intensified interest in renewable oxygenated fuels capable of improving the combustion and emission characteristics of gasoline direct-injection (GDI) spark-ignition engines. This study presents a comparative experimental investigation of the combustion behaviour, engine performance, and gaseous emissions of commercial unleaded gasoline (ULG95) and absolute ethanol in a spray-guided GDI engine. Experiments were conducted under stoichiometric operating conditions ($\lambda = 1$) at an engine speed of 1500 rpm over an indicated mean effective pressure (IMEP) range of 3.5–8.5 bar using optimized spark timing. Combustion characteristics were evaluated through in-cylinder pressure analysis, coefficient of variation of IMEP (COVIMEP), combustion phasing (MFB50), combustion duration, combustion efficiency, indicated thermal efficiency, and indicated specific fuel consumption. Regulated gaseous emissions comprising carbon monoxide (CO), unburned hydrocarbons (HC), and nitrogen oxides (NO_x) were also measured and compared. The results demonstrate that ethanol produced higher peak cylinder pressures, shorter combustion durations, improved combustion stability, and higher combustion and indicated thermal efficiencies than gasoline throughout the investigated load range. The superior combustion characteristics of ethanol were attributed to its inherent oxygen content, higher octane rating, and greater resistance to knock, which enabled optimum combustion phasing without knock-limited spark retard. Ethanol also produced substantially lower CO and HC emissions than gasoline, although differences in NO_x emissions reflected variations in combustion temperature and ignition characteristics. Overall, the study demonstrates that ethanol offers significant advantages in combustion quality and exhaust emission reduction, highlighting its potential as a sustainable fuel for future high-efficiency GDI spark-ignition engines.

Keywords- Gasoline direct-injection engine; Ethanol; Unleaded gasoline; Combustion characteristics; In-cylinder pressure; Combustion phasing; Thermal efficiency; Gaseous emissions; Renewable oxygenated fuels; Spark-ignition engine.

I. INTRODUCTION

The transportation sector remains one of the largest consumers of petroleum-derived fuels and a major contributor to global greenhouse gas (GHG) emissions. Although the rapid growth of battery electric vehicles (BEVs) and hybrid electric vehicles (HEVs) is gradually transforming the automotive industry, gasoline-powered spark-ignition (SI) engines are expected to remain an integral part of passenger transportation for the foreseeable future, particularly in regions where charging infrastructure, vehicle affordability, and energy security continue to favor conventional internal combustion engines [1], [2]. Consequently, improving the efficiency and reducing the environmental impact of gasoline engines remain important priorities for both researchers and automotive manufacturers.

Among modern spark-ignition technologies, gasoline direct-injection (GDI) engines have become increasingly popular because of their superior fuel economy, improved volumetric efficiency, higher specific power output, and enhanced transient response compared with conventional port fuel injection (PFI) engines [3], [4]. By injecting fuel directly into the combustion chamber, GDI engines provide improved charge cooling, greater flexibility in combustion control, and increased resistance to knock, thereby enabling higher compression ratios and engine downsizing. However, the direct injection process also introduces challenges, including fuel spray-wall impingement, mixture stratification, locally fuel-rich combustion zones, and increased particulate emissions [4], [5]. These issues have intensified the search for cleaner and more sustainable fuels capable of maintaining engine performance while reducing regulated exhaust emissions. Renewable oxygenated fuels have emerged as attractive alternatives to conventional gasoline because their chemically bound oxygen promotes more complete combustion and reduces the formation of incomplete combustion products such as carbon monoxide (CO) and unburned hydrocarbons (HC) [6], [7].

Among these fuels, ethanol has received the greatest attention owing to its renewable origin, high octane number, and widespread commercial availability. Ethanol possesses a research octane number exceeding 108, a high latent heat of vaporization, and significant oxygen content, characteristics that improve knock resistance and permit more advanced ignition timing in spark-ignition engines [8], [9]. Numerous experimental investigations have demonstrated that ethanol improves combustion efficiency and thermal efficiency while substantially reducing CO and HC emissions compared with conventional gasoline [10], [11]. However, ethanol also exhibits a lower heating value than gasoline, leading to increased fuel consumption on a mass and volumetric basis. In addition, its high latent heat of vaporization may adversely affect cold-start operation under low ambient temperatures, while its hygroscopic nature presents challenges during storage and transportation [8], [12].

Previous investigations have shown that ethanol generally produces faster flame propagation and shorter combustion duration than gasoline because of its higher laminar flame speed and oxygenated molecular structure [10], [13]. These characteristics improve combustion stability and allow combustion to occur closer to the optimum combustion phasing, resulting in higher indicated thermal efficiency. Furthermore, the higher knock resistance of ethanol enables operation at minimum spark advance for best torque (MBT) over a wider operating range than gasoline [9], [13]. Nevertheless, the improved combustion characteristics of ethanol may also influence combustion temperature and consequently alter nitrogen oxide (NO_x) formation. The combined effects of fuel chemistry, combustion phasing, and engine operating conditions therefore determine the overall environmental and performance benefits of ethanol-fuelled GDI engines [14].

Although numerous studies have compared gasoline and ethanol in spark-ignition engines, reported results vary considerably because of differences in engine design, fuel injection systems, compression ratio, spark timing strategy, engine load, and operating conditions [9], [10], [15]. Many published investigations have focused primarily on brake performance or regulated emissions, while comparatively fewer studies have simultaneously examined detailed combustion characteristics—including peak cylinder pressure, combustion phasing, combustion duration, cyclic combustion variation, combustion efficiency, and gaseous emissions—under identical operating conditions in a spray-guided gasoline direct-injection engine. Such comprehensive experimental analyses are important because they provide a clearer understanding of the mechanisms responsible for the observed differences in combustion behaviour and pollutant formation.

The present study addresses this need through a detailed experimental comparison of commercial unleaded gasoline (ULG95) and absolute ethanol in a spray-guided gasoline direct-injection spark-ignition engine. Experiments were conducted under stoichiometric conditions at an engine speed of 1500 rpm over an indicated mean effective pressure (IMEP) range of 3.5–8.5 bar using optimized spark timing for each fuel. Combustion behaviour was evaluated using in-cylinder pressure measurements, coefficient of variation of IMEP (COVIMEP), combustion phasing (MFB50), combustion duration, combustion efficiency, indicated thermal efficiency, and indicated specific fuel consumption. Regulated gaseous emissions comprising carbon monoxide (CO), unburned hydrocarbons (HC), and nitrogen oxides (NO_x) were also quantified and compared. The principal contribution of this work is the integration of combustion analysis, engine performance, and emission characteristics within a single experimental investigation under identical operating conditions.

By relating combustion behaviour directly to engine efficiency and pollutant formation, the study provides a comprehensive assessment of the advantages and limitations of ethanol relative to conventional gasoline in a modern GDI engine. The findings contribute to a better understanding of the combustion processes governing oxygenated fuels and provide valuable guidance for the optimization of renewable fuels in future high-efficiency spark-ignition engines.

II. EXPERIMENTAL METHODOLOGY

Experimental Test Engine

The experimental investigation was conducted using a single-cylinder, four-stroke, spray-guided gasoline direct-injection (GDI) spark-ignition research engine. The engine was specifically designed for combustion and emissions studies and equipped with a centrally mounted high-pressure fuel injector to facilitate direct fuel injection into the combustion chamber. The spray-guided combustion concept was selected because it provides improved mixture preparation and ignition stability while minimizing wall wetting and fuel impingement compared with wall-guided combustion systems. The engine was coupled to an eddy-current dynamometer capable of maintaining steady-state operating conditions throughout the experimental programme. Engine speed and load were controlled electronically through the dynamometer control system to ensure repeatable operating conditions for all tests.

The experiments were performed at a constant engine speed of 1500 rpm, representing a typical medium-speed operating condition commonly used for combustion investigations and comparative fuel evaluation. A production-type electronic engine management system was employed to regulate fuel injection and spark timing. The ignition timing for each fuel was adjusted to achieve the minimum spark advance for best torque (MBT) whenever knock was absent. Under operating conditions where knock occurred with gasoline, the spark timing was limited by the knock-limited spark advance (KLSA), whereas ethanol was operated at MBT throughout the investigated load range due to its superior knock resistance.

Cooling water temperature and lubricating oil temperature were maintained within their prescribed operating ranges throughout the experiments to eliminate the influence of thermal fluctuations on combustion performance and exhaust emissions. Before each experimental run, the engine was allowed to attain steady-state operating conditions to ensure repeatable combustion measurements. The principal specifications of the research engine are summarized in Table 1.

Table 1: Specifications of the Spray-Guided GDI Research Engine

Parameter	**Specification**
Engine type	Single-cylinder, four-stroke SI engine
Combustion system	Spray-guided GDI
Fuel injection	High-pressure direct injection
Bore	XX mm
Stroke	XX mm
Swept volume	XX cm^3
Compression ratio	XX:1
Number of valves	XX
Maximum injection pressure	XX MPa
Cooling system	Water cooled
Ignition system	Electronic spark ignition
Engine speed	150rpm

Test Fuels

Two fuels were investigated in this study: are the Commercial Unleaded Gasoline (ULG95) and the Absolute Ethanol (≥99.8%) Commercial unleaded gasoline served as the baseline reference fuel, while absolute ethanol was selected because of its high oxygen content, renewable origin, and superior anti-knock characteristics. Unlike ethanol–gasoline blends, neat ethanol permits the direct assessment of the influence of fuel molecular structure on combustion and emission characteristics without interference from blend composition. Ethanol possesses several physicochemical properties that distinguish it from conventional gasoline. These include a significantly higher octane number, greater latent heat of vaporization, lower heating value, lower stoichiometric air–fuel ratio, and approximately 35 wt.% oxygen within its molecular structure. These characteristics influence mixture preparation, combustion phasing, flame propagation, thermal efficiency, and pollutant formation. The principal physicochemical properties of the investigated fuels are presented in Table 2.

Table 2 : Typical Physicochemical Properties of Gasoline and Absolute Ethanol

Property	**Gasoline**	**Ethanol**
Chemical formula	Variable hydrocarbons	C_2H_5OH
Density at 15°C (kg/m^3)	~740	~789
Lower heating value (MJ/kg)	42.8–43.5	26.8
Research octane number (RON)	95	108–109

Motor octane number (MON)	85	89–90
Stoichiometric AFR	14.7	9.0
Oxygen content (wt.%)	0	34.8
Latent heat of vaporization (kJ/kg)	~350	~840
Auto-ignition temperature (°C)	~280	~365

The higher latent heat of vaporization of ethanol promotes significant charge cooling during the intake process, reducing in-cylinder mixture temperature and suppressing knock. Furthermore, its chemically bound oxygen enhances combustion completeness, thereby reducing incomplete combustion products such as carbon monoxide and unburned hydrocarbons. Conversely, the lower heating value of ethanol necessitates greater fuel mass to produce the same engine output, resulting in higher indicated specific fuel consumption.

Experimental Facility and Instrumentation

The experimental facility consisted of the GDI research engine, an eddy-current dynamometer, a high-pressure fuel supply system, combustion pressure acquisition system, fuel consumption measurement system, intake air.

Fig. 2. 1. Experimental test facility and Test grid

measurement system, and a Horiba exhaust gas analyzer for gaseous emission measurements. Cylinder pressure was measured using a water-cooled piezoelectric pressure transducer mounted flush with the combustion chamber. The pressure signal was synchronized with crank-angle position using a high-resolution crank-angle encoder installed on the crankshaft. The encoder provided crank-angle resolution sufficient for accurate combustion analysis, allowing determination of heat release rate, combustion phasing, combustion duration, and peak pressure location. Fuel consumption was measured using a precision gravimetric fuel measurement system, while intake air flow was monitored using calibrated air-flow instrumentation. Cooling water and lubricating oil temperatures were continuously monitored to ensure stable thermal operating conditions. Regulated gaseous emissions were measured using a Horiba exhaust gas analyzer. Carbon monoxide (CO), total hydrocarbons (THC), and nitrogen oxides (NO_x) were measured after the engine had stabilized at each operating condition. Prior to each experimental session, the analyzer was calibrated using certified span and zero gases in accordance with the manufacturer's operating procedures to ensure measurement accuracy.

Experimental Procedure

All experiments were performed under naturally aspirated, stoichiometric operating conditions ($\lambda = 1.0$) at a constant engine speed of 1500 rpm. Engine load was varied by adjusting the throttle opening to obtain indicated mean effective pressures (IMEP) between 3.5 and 8.5 bar. For each operating condition, spark timing was optimized to achieve MBT whenever knock was absent. For gasoline operation, spark timing was limited by knock whenever knock occurred at higher loads. Ethanol, owing to its superior octane quality, operated at MBT over the entire investigated load range.

Before data acquisition, the engine was allowed to stabilize until coolant temperature, lubricating oil temperature, intake air temperature, and exhaust gas temperatures remained constant. Once steady-state operation was achieved, combustion pressure data were recorded over multiple consecutive engine cycles to minimize the influence of cycle-to-cycle variability. Simultaneously, fuel consumption and gaseous emissions were measured under identical operating conditions. The experimental programme ensured that all combustion, performance, and emission measurements were obtained under identical engine operating conditions, thereby enabling direct comparison between gasoline and ethanol.

Combustion Analysis

Combustion analysis was performed using the measured in-cylinder pressure traces combined with crank-angle-resolved cylinder volume calculations. Heat release analysis was subsequently conducted using the first law of thermodynamics to determine combustion phasing and combustion duration.

Peak cylinder pressure provides an indication of combustion intensity and energy release. Combustion phasing was evaluated using MFB50, which represents the crank angle at which 50% of the cumulative heat release has occurred. Combustion duration was defined as the crank-angle interval between 10% and 90% mass fraction burned. Cycle-to-cycle combustion stability was quantified using the coefficient of variation of IMEP (COVIMEP):

$$\mathrm{COV}_{IMEP} = \frac{\sigma_{IMEP}}{\overline{IMEP}} \quad (1)$$

where σ_IMEPis the standard deviation of IMEP and ‾IMEPis the mean indicated mean effective pressure.

Cycle-to-cycle combustion stability was quantified using the coefficient of variation of the indicated mean effective pressure (COVIMEP), which is widely employed as an indicator of combustion stability in spark-ignition engines. Lower COVIMEP values indicate more stable combustion, while values exceeding approximately 5% are generally associated with noticeable deterioration in engine stability, increased cyclic variability, and degraded drivability [16]. Combustion efficiency and indicated thermal efficiency were determined from the measured fuel consumption and indicated work output using standard engine performance equations. These parameters provided an assessment of the effectiveness with which the chemical energy of each fuel was converted into useful engine work.

III. RESULTS AND DISCUSSION

Combustion Characteristics

This section presents a comparative analysis of the combustion characteristics of commercial unleaded gasoline (ULG95) and absolute ethanol in the spray-guided gasoline direct-injection (GDI) engine under stoichiometric operating conditions. The combustion process was evaluated over the IMEP range of 3.5–8.5 bar at a constant engine speed of 1500 rpm using optimized spark timing for each fuel. The analysis focuses on the in-cylinder pressure development, combustion stability, combustion phasing, combustion duration, and combustion efficiency, which collectively provide insight into the influence of fuel physicochemical properties on engine performance and combustion quality.

The combustion behaviour of spark-ignition engines is governed by several interacting factors, including fuel molecular structure, octane rating, oxygen content, latent heat of vaporization, laminar flame speed, and ignition characteristics. Ethanol differs significantly from gasoline in these properties. Its higher octane number permits operation closer to the minimum spark advance for best torque (MBT), while its higher latent heat of vaporization provides charge-cooling effects that suppress knock and improve volumetric efficiency. Furthermore, the oxygen contained within the ethanol molecule enhances oxidation during combustion, promoting more complete combustion and reducing incomplete combustion products such as carbon monoxide and unburned hydrocarbons. These characteristics collectively influence pressure development, heat release, combustion duration, thermal efficiency, and exhaust emissions [19]–[22].

Recent investigations have reported that ethanol generally produces higher peak cylinder pressures, shorter combustion durations, improved combustion stability, and higher indicated thermal efficiencies than conventional gasoline under stoichiometric GDI operation. These improvements have been attributed primarily to the combined effects of advanced combustion phasing, enhanced flame propagation, and increased resistance to knock [20], [21], [23]. However, the magnitude of these improvements depends strongly on engine operating conditions, spark timing strategy, fuel injection characteristics, and engine load. Therefore, detailed experimental evaluation under identical operating conditions remains essential for understanding the combustion behaviour of different fuels. The following subsections discuss the influence of fuel type on the principal combustion parameters measured during the experimental programme.

Peak Cylinder Pressure and Combustion Stability

The variation of maximum in-cylinder pressure for gasoline and ethanol over the investigated IMEP range is presented in Figure 1, while the corresponding coefficient of variation of IMEP (COVIMEP) is shown in Figure 2. Peak cylinder pressure is one of the most important combustion parameters because it reflects the rate of energy release and the effectiveness of converting the chemical energy of the fuel into useful work. Under optimum combustion conditions, higher peak cylinder pressures generally indicate improved combustion efficiency and greater work extraction, provided that excessive pressure rise rates and knock are avoided [3], [16].

As shown in Figure 1, the peak cylinder pressure increased progressively with engine load for both fuels, reflecting the increase in fuel mass supplied to the engine and the corresponding increase in heat release during combustion. This trend is consistent with the expected thermodynamic behaviour of spark-ignition engines operating under stoichiometric conditions and has been reported in numerous experimental studies involving gasoline and oxygenated fuels [17]-[19]. At the lowest investigated load (3.5 bar IMEP), ethanol produced only a modest increase in peak cylinder pressure compared with gasoline. However, as engine load increased, the difference became progressively more pronounced. At 8.5 bar IMEP, ethanol generated a peak cylinder pressure approximately 12 bar higher than gasoline, representing an increase of approximately 33% over the gasoline baseline.

Similar observations have been reported in recent GDI investigations, where ethanol consistently produced higher peak combustion pressures because its superior knock resistance permitted operation closer to MBT while its oxygenated molecular structure promoted more rapid and efficient combustion [17], [20]. Several factors contribute to the superior pressure development observed with ethanol. First, the higher research octane number of ethanol significantly increases knock resistance, allowing more advanced ignition timing without abnormal combustion. Second, ethanol possesses a higher laminar flame speed than gasoline under stoichiometric conditions, accelerating flame propagation and causing a greater proportion of the fuel energy to be released near top dead centre.

Third, the cooling effect associated with its high latent heat of vaporization increases charge density and volumetric efficiency, further enhancing combustion performance. The combined influence of these characteristics results in higher peak cylinder pressures and improved thermal efficiency. Combustion stability was evaluated using the coefficient of variation of indicated mean effective pressure (COVIMEP), which provides a quantitative measure of cycle-to-cycle combustion variability. The COVIMEP values obtained for both fuels remained below 3% throughout the investigated load range (Figure 2), indicating highly stable combustion. It is generally accepted that COVIMEP values below 5% correspond to smooth engine operation with negligible deterioration in combustion stability [3]. Although both fuels exhibited excellent combustion stability, ethanol consistently produced slightly lower COVIMEP values than gasoline over most of the investigated operating range.

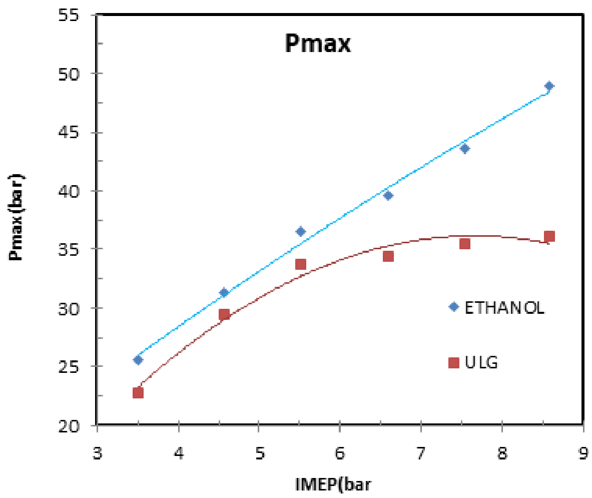

Fig.3.1. Maximum cylinder pressure (Pmax) for Ethanol and ULG at 3.5 -8.5barIMEP

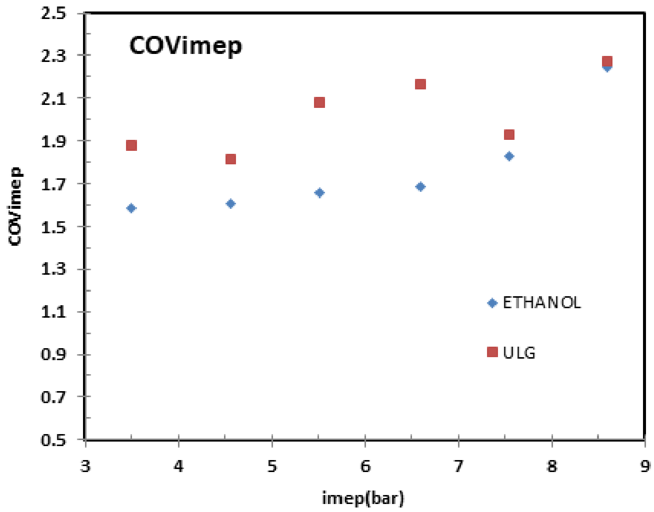

Fig 3.2. COV of IMEP (COVIMEP) for Ethanol and ULG at 3.5-8.5 barIMEP

The improved stability may be attributed to its higher flame propagation rate and the enhanced combustion completeness resulting from its chemically bound oxygen. These findings agree well with recent experimental studies on ethanol-fuelled GDI engines, which reported reduced cyclic variability due to faster flame development and more repeatable combustion events [21], [19]. As engine load increased, a slight increase in

COVIMEP was observed for both fuels. This behaviour is expected because higher cylinder pressures increase combustion sensitivity to small variations in mixture formation and ignition timing. Nevertheless, the observed COVIMEP values remained well within the range associated with stable combustion, confirming the suitability of both fuels for GDI engine operation under the investigated conditions. Overall, the results demonstrate that ethanol provides superior combustion quality compared with gasoline by simultaneously increasing peak cylinder pressure and improving combustion stability. These characteristics contribute directly to enhanced thermal efficiency and improved engine performance, which are discussed in the subsequent sections,

Combustion Phasing

The combustion phasing characteristics of gasoline and absolute ethanol were evaluated using the crank-angle location corresponding to 50% mass fraction burned (MFB50). MFB50 is widely recognized as one of the most reliable combustion descriptors because it represents the midpoint of the combustion process and provides a direct indication of the effectiveness of spark timing relative to top dead centre (TDC). For spark-ignition engines operating under stoichiometric conditions, optimum combustion efficiency is generally achieved when MFB50 occurs between 8° and 10° after top dead centre (ATDC), allowing the maximum proportion of the released chemical energy to be converted into useful expansion work while minimizing heat losses and knock tendency [16], [21], [22].

Figure 3 illustrates the variation of MFB50 with engine load for gasoline and ethanol over the investigated IMEP range. At low engine loads, both fuels exhibited similar combustion phasing because the combustion process remained relatively slow and knock was absent. However, as engine load increased, noticeable differences in combustion phasing developed between the two fuels. Ethanol consistently maintained MFB50 within the optimum combustion window throughout the investigated operating range, whereas gasoline exhibited progressively delayed combustion phasing at higher loads due to spark retard imposed by knock-limited spark advance (KLSA).

The divergence in combustion phasing became particularly evident above 6.5 bar IMEP, where gasoline operation was increasingly constrained by knock. At the maximum investigated load of 8.5 bar IMEP, ethanol achieved an MFB50 approximately 8 crank-angle degrees earlier than gasoline. This earlier combustion phasing resulted in a greater proportion of the fuel energy being released near TDC, thereby increasing expansion work and contributing directly to the higher peak cylinder pressures and improved indicated thermal efficiency observed for ethanol.

The superior combustion phasing exhibited by ethanol can be attributed primarily to its favourable physicochemical properties. Ethanol possesses a significantly higher research octane number than gasoline, enabling combustion to occur at MBT throughout the entire load range without abnormal combustion. In contrast, gasoline required ignition retard at higher loads to suppress knock, delaying the combustion process and reducing thermodynamic efficiency. Similar behaviour has been reported in recent GDI engine investigations, where ethanol's higher knock resistance permitted earlier combustion phasing and improved combustion efficiency compared with conventional gasoline [20], [21], [26]. Another important factor influencing combustion phasing is the higher laminar flame speed of ethanol. Faster flame propagation accelerates the development of the flame kernel and shortens the time required for combustion to reach its midpoint.

Consequently, ethanol reaches MFB50 earlier than gasoline even when both fuels are operated under identical air–fuel ratios and engine speeds. Furthermore, the oxygen incorporated within the ethanol molecule promotes more homogeneous oxidation reactions during combustion, thereby improving flame propagation and combustion completeness. Recent optical combustion studies have similarly reported that oxygenated fuels exhibit more rapid flame development and shorter flame travel times than hydrocarbon gasoline, particularly under stoichiometric operating conditions [19], [24]. The results obtained in the present investigation are consistent with those reported by Wang et al., who observed that ethanol-based fuels maintained combustion phasing closer to the optimum MBT location because of their superior anti-knock characteristics.

Likewise, Zhang et al. demonstrated that ethanol consistently advanced the combustion centroid and improved indicated thermal efficiency in a gasoline direct-injection engine operating under medium and high loads [18], [22]. These observations further confirm that combustion phasing remains one of the principal mechanisms through which ethanol enhances engine efficiency. The influence of combustion phasing on engine performance is substantial because even small deviations from the optimum MFB50 location can produce measurable reductions in thermal efficiency. Delayed combustion shifts the pressure rise further into the expansion stroke, thereby reducing the useful work extracted from combustion.

Conversely, excessively advanced combustion may increase negative compression work and elevate knock tendency. The ability of ethanol to maintain MFB50 within the optimum combustion window therefore provides a significant thermodynamic advantage over gasoline under high-load operation. Overall, the combustion phasing results demonstrate that ethanol enables combustion to occur closer to the optimum crank-angle location than gasoline over the entire investigated load range. This favourable combustion behaviour explains the higher peak cylinder pressures, improved combustion efficiency, and enhanced indicated thermal efficiency observed throughout the experimental programme. The results further confirm that combustion phasing represents one of the primary factors governing the superior combustion performance of ethanol in spray-guided GDI engines.

Combustion Duration

Combustion duration is one of the principal parameters governing the thermodynamic performance and combustion quality of spark-ignition engines. It defines the crank-angle interval required for the combustion process to progress between specified mass fraction burned (MFB) limits and provides valuable information on flame development and flame propagation characteristics. In combustion studies, the 10–90% mass fraction burned (MFB10–90) interval is widely accepted as the most representative measure of the main combustion period because it excludes the initial flame kernel development and the final slow-burning phase while accurately representing the rapid energy release stage of combustion [16], [25], [26].

A shorter combustion duration generally results in a larger proportion of the released chemical energy being concentrated around top dead centre (TDC), thereby increasing the peak cylinder pressure and improving the conversion of thermal energy into useful work. Conversely, prolonged combustion shifts heat release further into the expansion stroke, reducing pressure development, lowering indicated thermal efficiency, and increasing cycle-to-cycle combustion variability. Consequently, combustion duration is closely associated with engine efficiency, combustion stability, fuel economy, and exhaust emissions [3], [16], [27]. The variation of combustion duration (defined by MFB10–90) with engine load for gasoline and ethanol is presented in Figure 4. At the lowest investigated load (3.5 bar IMEP), both fuels exhibited similar combustion durations because the combustion process was relatively slow and knock-free under these operating conditions.

However, as engine load increased, the combustion duration of ethanol became progressively shorter than that of gasoline. The difference between the two fuels increased continuously with load, with the largest deviation observed at 8.5 bar IMEP. The consistently shorter combustion duration of ethanol can be attributed primarily to its favourable combustion characteristics. Ethanol possesses a higher laminar flame speed than gasoline under stoichiometric conditions, enabling the flame front to propagate more rapidly through the combustible mixture. Furthermore, the chemically bound oxygen within the ethanol molecule promotes more complete oxidation reactions during combustion, enhancing flame propagation and accelerating the overall combustion process.

These characteristics enable a greater proportion of the fuel energy to be released within a shorter crank-angle interval, thereby increasing combustion intensity and pressure development. Another important factor contributing to the shorter combustion duration is ethanol's superior knock resistance. Because ethanol possesses a significantly higher research octane number than gasoline, the engine could be operated at MBT throughout the investigated load range without requiring spark retard. In contrast, gasoline operation at higher loads was constrained by knock-limited spark advance (KLSA), delaying the initiation of combustion and consequently extending the overall combustion duration. This behaviour is clearly reflected in the progressive divergence of the combustion duration curves with increasing engine load.

The shorter combustion duration observed for ethanol provides a direct explanation for the higher peak cylinder pressures presented in Figure 1 and the earlier combustion phasing illustrated in Figure 3. Faster combustion concentrates heat release closer to TDC, producing a steeper pressure rise and allowing a larger fraction of the released energy to contribute to useful expansion work. Consequently, ethanol consistently achieved higher combustion efficiency and improved indicated thermal efficiency compared with gasoline throughout the investigated operating range. The present findings agree well with recent experimental investigations on ethanol-fuelled GDI engines. Li et al. reported that ethanol reduced combustion duration because of its higher laminar flame speed and enhanced flame propagation characteristics, resulting in improved pressure development and thermal efficiency [17].

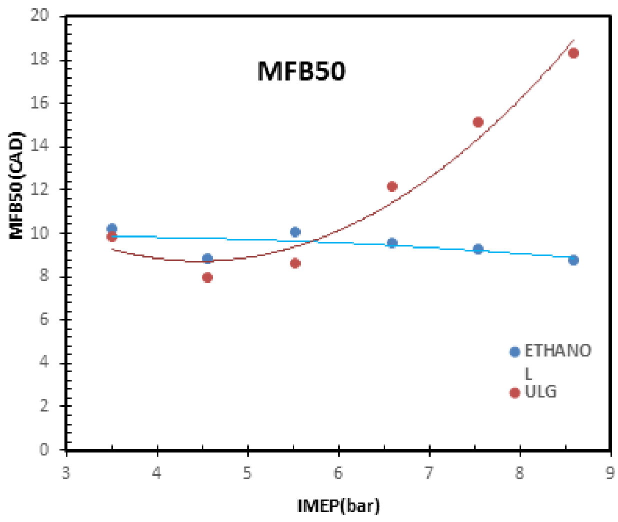

Fig.3.3. Combustion duration (defines by 50%MFB) for ULG and Ethanol at 3.5-8.5bar IMEP.

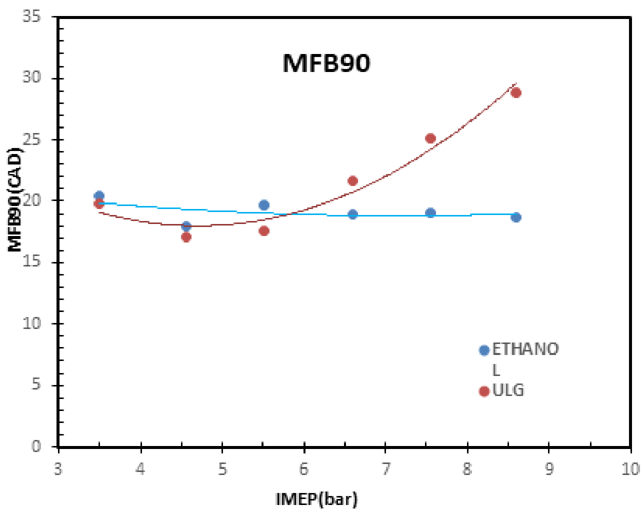

Fig.3.4. Combustion duration (defined by 10%-90%MFB) for ULG and Ethanol 3.5-8.5barIMEP

Similarly, Zhang et al. observed that oxygenated fuels consistently exhibited shorter combustion durations than gasoline under stoichiometric GDI operation, particularly at medium and high engine loads, owing to improved combustion kinetics and combustion stability [18]. Anselmi et al. further demonstrated that renewable oxygenated fuels with favourable combustion characteristics generally achieve faster heat release and improved combustion efficiency, making them attractive candidates for future high-efficiency spark-ignition engines [19]. Combustion duration also influences exhaust emission formation. Faster combustion generally improves oxidation of intermediate combustion products, thereby reducing carbon monoxide and unburned hydrocarbon emissions.

However, concentrating heat release within a shorter crank-angle interval may locally increase combustion temperatures, potentially influencing thermal NO_x formation. The overall effect therefore depends on the balance between combustion duration, combustion phasing, in-cylinder temperature distribution, and the cooling effect associated with ethanol's high latent heat of vaporization. These interactions are examined in detail in the emissions analysis presented in Section 3.2. From an engine design perspective, reducing combustion duration remains an important strategy for improving thermal efficiency without increasing fuel consumption. The present experimental results demonstrate that ethanol naturally provides this advantage because of its inherent fuel properties rather than through modifications to engine hardware.

This characteristic makes ethanol particularly attractive for modern downsized and highly boosted gasoline direct-injection engines, where rapid combustion, knock suppression, and improved thermal efficiency are essential for achieving stringent fuel economy and emission targets. Overall, the combustion duration results clearly demonstrate the superior combustion behaviour of ethanol relative to gasoline. The shorter combustion duration observed throughout the investigated load range contributed directly to the earlier combustion phasing, higher peak cylinder pressures, improved combustion stability, and enhanced indicated thermal efficiency reported in the present study. These findings further confirm that the intrinsic physicochemical properties of ethanol significantly improve the combustion process in spray-guided gasoline direct-injection engines.

Combustion Efficiency

Combustion efficiency is an important performance parameter that quantifies the effectiveness with which the chemical energy stored in a fuel is released during the combustion process. It is generally defined as the ratio of the actual energy liberated during combustion to the total chemical energy

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supplied by the fuel. In spark-ignition engines, combustion efficiency is strongly influenced by combustion completeness, flame propagation characteristics, mixture preparation, ignition quality, and fuel molecular composition. Under stable stoichiometric operation, combustion efficiencies typically exceed 95%, although slight reductions may occur because of incomplete oxidation of carbon monoxide (CO), unburned hydrocarbons (HC), fuel trapped within piston ring crevices, flame quenching near combustion chamber walls, and cyclic combustion variations [3], [16], [28].

The variation of combustion efficiency with engine load for gasoline and absolute ethanol is presented in Figure 5. Both fuels exhibited relatively high combustion efficiencies throughout the investigated IMEP range of 3.5–8.5 bar, indicating stable combustion under stoichiometric operating conditions. However, ethanol consistently achieved higher combustion efficiencies than gasoline across the entire load range. The combustion efficiency of ethanol remained close to 97%, whereas gasoline generally exhibited values between approximately 93% and 94%, demonstrating a clear advantage for the oxygenated fuel. The superior combustion efficiency observed for ethanol can be attributed primarily to its molecular oxygen content. Unlike conventional gasoline, ethanol contains approximately 35 wt.% oxygen, which promotes more complete oxidation of hydrocarbon intermediates during combustion.

This additional oxygen enhances flame propagation, reduces locally fuel-rich regions within the combustion chamber, and facilitates the conversion of carbon monoxide and partially oxidized hydrocarbons into carbon dioxide and water. Consequently, ethanol experiences fewer incomplete combustion losses, leading to improved combustion efficiency. Similar observations have been reported in recent investigations of oxygenated fuels in gasoline direct-injection engines, where ethanol consistently demonstrated higher combustion completeness because of its inherent oxygen availability and improved oxidation chemistry [17], [18], [28]. Another important factor contributing to the improved combustion efficiency is the shorter combustion duration discussed in the previous section.

Faster combustion enables a greater proportion of the released chemical energy to be liberated within the optimum crank-angle interval, thereby reducing heat transfer losses to the combustion chamber walls and minimizing the probability of incomplete oxidation during the expansion stroke. The earlier combustion phasing (MFB50) achieved by ethanol further concentrates heat release near top dead centre (TDC), allowing more efficient conversion of thermal energy into useful indicated work. These observations demonstrate the close relationship between combustion duration, combustion phasing, and combustion efficiency.

The high latent heat of vaporization of ethanol also influences combustion efficiency. During the intake and compression processes, ethanol absorbs a considerable amount of heat as it vaporizes, producing a cooling effect that increases charge density and improves volumetric efficiency. Although this cooling effect reduces the initial mixture temperature, the improved air utilization and more homogeneous mixture formation generally outweigh the reduction in temperature, resulting in more stable combustion and enhanced combustion completeness. Recent optical combustion studies have similarly reported that improved charge preparation and mixture homogeneity contribute significantly to the superior combustion efficiency of oxygenated fuels under stoichiometric GDI operation [22], [29], [30].

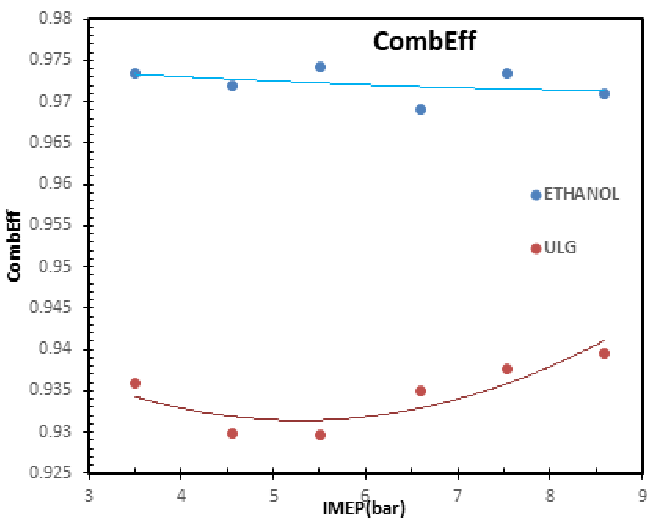


Fig.3.5. Combustion efficiency for ethanol and ULG 3.5-8.5barIMEP

In contrast, gasoline exhibited slightly lower combustion efficiency throughout the investigated operating range. The absence of chemically bound oxygen within gasoline increases the dependence of combustion on oxygen supplied solely through the intake air. Under direct-injection operation, localized fuel-rich regions and spray-wall interactions may develop within the combustion chamber, particularly as engine load increases.

These locally rich mixtures are more susceptible to incomplete combustion and flame quenching, leading to higher emissions of carbon monoxide and unburned hydrocarbons and consequently reducing overall combustion efficiency. This behaviour is consistent with previous experimental studies on gasoline direct-injection engines, which have shown that mixture stratification and wall wetting remain important contributors to incomplete combustion under certain operating conditions [28], [31]. Although ethanol consistently produced higher combustion efficiency, it should be recognized that combustion efficiency alone does not fully characterize engine performance. A fuel may exhibit excellent combustion efficiency while simultaneously requiring greater fuel consumption because of its lower heating value.

Consequently, combustion efficiency must be interpreted together with thermal efficiency and specific fuel consumption to provide a comprehensive assessment of fuel performance. This distinction is particularly important when comparing gasoline with ethanol, since ethanol's lower energy density necessitates a greater mass of fuel to produce equivalent engine output despite its superior combustion characteristics. The present experimental observations agree well with recent studies on renewable oxygenated fuels. Li et al. reported that ethanol improved combustion completeness through enhanced oxidation reactions and faster flame propagation, resulting in consistently higher combustion efficiencies than gasoline under stoichiometric GDI operation [17]. Likewise, Zhang et al. observed that ethanol achieved more stable combustion and reduced incomplete combustion losses, thereby improving combustion efficiency across a wide range of engine loads [18].

Anselmi et al. further concluded that renewable oxygenated fuels generally exhibit superior combustion efficiencies because of the combined effects of molecular oxygen, improved combustion kinetics, and enhanced flame propagation characteristics [19]. Overall, the combustion efficiency results confirm that ethanol provides a more complete and stable combustion process than gasoline throughout the investigated operating conditions. The higher combustion efficiencies observed in the present study are consistent with the earlier combustion phasing, shorter combustion duration, and higher peak cylinder pressures previously discussed. These characteristics collectively contribute to the improved indicated thermal efficiency presented in the following section.

Inicated Thermal Efficiency

Indicated thermal efficiency (ITE) is a fundamental performance parameter used to evaluate the effectiveness with which the chemical energy supplied by the fuel is converted into useful work within the engine cylinder. Unlike combustion efficiency, which quantifies the completeness of the combustion process, indicated thermal efficiency reflects the overall thermodynamic efficiency of energy conversion by accounting for the work produced during the engine cycle relative to the fuel energy supplied. Consequently, ITE is strongly influenced by combustion phasing, combustion duration, heat transfer losses, compression ratio, knock characteristics, and the physicochemical properties of the fuel [16], [32], [33].

The indicated thermal efficiency for the present study was determined from the ratio of the indicated work to the rate of chemical energy supplied by the fuel, as defined in Section 2.5. Figure 6 presents the variation of indicated thermal efficiency with engine load for gasoline and absolute ethanol under stoichiometric operation at 1500 rpm. As shown in Figure 6, the indicated thermal efficiency increased progressively with engine load for both fuels throughout the investigated IMEP range of 3.5–8.5 bar. This trend is characteristic of spark-ignition engines because the relative influence of frictional losses, pumping losses, and heat transfer losses decreases as engine load increases. Consequently, a larger proportion of the chemical energy released during combustion is converted into useful indicated work at higher loads [16], [32].

Although both fuels exhibited increasing thermal efficiency with load, ethanol consistently achieved higher indicated thermal efficiencies than gasoline across the entire operating range. The improvement became increasingly pronounced as engine load increased, with the largest difference observed at 8.5 bar IMEP. The superior thermal efficiency of ethanol can be attributed to the combined influence of several favourable combustion characteristics demonstrated in the preceding sections. First, ethanol maintained combustion phasing closer to the optimum MFB50 location throughout the investigated operating range. Earlier combustion phasing allowed a greater proportion of the heat release to occur shortly after top dead centre (TDC), thereby maximizing cylinder pressure during the power stroke and increasing the useful expansion work. In contrast, gasoline required spark retard at higher loads because of knock limitations, delaying combustion and reducing thermodynamic efficiency. Second, ethanol exhibited consistently shorter combustion durations than gasoline.

Faster combustion concentrated energy release within a narrower crank-angle interval, increasing the average cylinder pressure during expansion while reducing the duration of high-temperature heat transfer to the combustion chamber walls. This reduction in heat losses improved the fraction of fuel energy converted into indicated work, thereby increasing thermal efficiency. Third, the higher combustion efficiency achieved by ethanol contributed directly to the improvement in indicated thermal efficiency. More complete oxidation of the fuel reduced incomplete combustion losses associated with carbon monoxide and unburned hydrocarbons, enabling a larger proportion of the available chemical energy to participate in useful work production. The combination of higher combustion efficiency, earlier combustion phasing, and shorter combustion duration therefore explains the consistently higher indicated thermal efficiencies measured for ethanol.

An additional factor contributing to ethanol's superior thermal efficiency is its high latent heat of vaporization. The substantial charge-cooling effect produced during fuel evaporation increased the density of the inducted air, thereby improving volumetric efficiency and increasing the mass of fresh charge entering the cylinder. Improved volumetric efficiency enhanced combustion quality and pressure development, particularly at medium and high engine loads. Furthermore, the cooling effect suppressed end-gas auto-ignition, allowing operation at minimum spark advance for best torque (MBT) throughout the investigated load range without the occurrence of knock.

This operating advantage is particularly important for modern gasoline direct-injection engines, where knock often limits thermal efficiency at high loads [17], [18], [34]-[37]. The present findings are consistent with recent experimental investigations on ethanol-fuelled gasoline direct-injection engines. Li et al. reported that ethanol produced higher indicated thermal efficiencies because of its superior knock resistance, faster flame propagation, and improved combustion completeness [17]. Similarly, Zhang et al. observed that oxygenated fuels consistently increased thermal efficiency through improved combustion phasing and reduced combustion duration under stoichiometric operating conditions [18]. Anselmi et al. further demonstrated that renewable oxygenated fuels enhance engine efficiency by simultaneously improving combustion quality and suppressing knock, thereby enabling operation closer to optimum ignition timing [19].

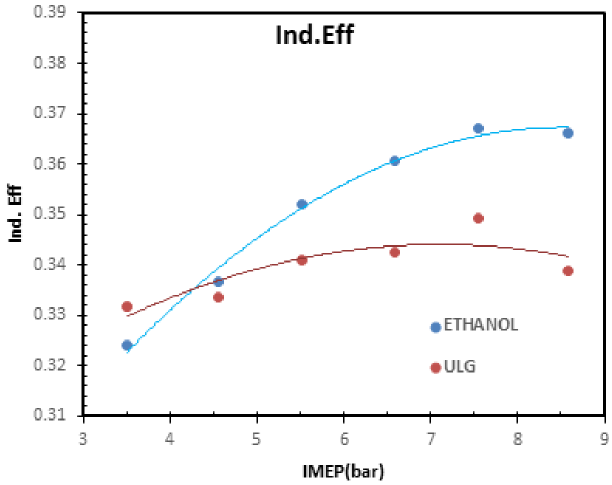


Fig 3.6. Indicated thermal efficiency for ethanol and ULG 3.5bar-8.5bar

Although ethanol possesses a lower lower heating value (LHV) than gasoline, this disadvantage did not prevent improvements in indicated thermal efficiency. Thermal efficiency is fundamentally a measure of how effectively the supplied chemical energy is converted into useful work rather than the absolute energy content of the fuel. The present results therefore demonstrate that the superior combustion characteristics of ethanol more than compensate for its lower energy density under the investigated operating conditions. It should also be noted that indicated thermal efficiency differs from brake thermal efficiency because it excludes mechanical friction losses within the engine.

Consequently, the improvements observed in the present study primarily reflect enhanced in-cylinder combustion processes rather than changes in mechanical efficiency. Nevertheless, improvements in indicated thermal efficiency generally translate into corresponding improvements in brake thermal efficiency, provided that engine mechanical losses remain unchanged. Overall, the indicated thermal efficiency results confirm that ethanol provides superior thermodynamic performance compared with gasoline in the spray-guided gasoline direct-injection engine investigated. The improvements are principally attributed to optimized combustion phasing, shorter combustion duration, higher combustion efficiency, and enhanced knock resistance. These combustion advantages enable a greater proportion of the fuel's chemical energy to be converted into useful indicated work, demonstrating the considerable potential of ethanol as a renewable fuel for future high-efficiency spark-ignition engines.

Indicated Specific Fuel Consumption

Indicated specific fuel consumption (ISFC) is a widely used performance parameter that quantifies the mass of fuel required to produce a unit of indicated work. It provides a direct measure of fuel utilization efficiency and is commonly employed to compare the fuel economy of different fuels operating under identical engine conditions. Unlike indicated thermal efficiency, which measures the effectiveness of converting the supplied chemical energy into useful work, ISFC reflects the actual fuel mass consumed to generate a given engine output. Consequently, ISFC is influenced by fuel heating value, combustion efficiency, combustion phasing, engine operating conditions, and thermodynamic efficiency [16], [32], [36]. Figure 7 illustrates the variation of ISFC with engine load for gasoline and absolute ethanol at 1500 rpm under stoichiometric conditions.

For both fuels, ISFC decreased progressively with increasing IMEP throughout the investigated operating range. This behaviour is typical of spark-ignition engines because the proportion of frictional losses, pumping losses, and auxiliary power consumption decreases as engine load increases. Consequently, a larger fraction of the fuel energy is converted into useful indicated work at higher loads, resulting in lower specific fuel consumption [16], [32]. Although both fuels exhibited decreasing ISFC with increasing load, ethanol consistently recorded higher ISFC values than gasoline over the entire IMEP range. The difference remained evident even though ethanol demonstrated superior combustion efficiency, earlier combustion phasing, shorter combustion duration, and higher indicated thermal efficiency. This apparent contradiction highlights the importance of distinguishing between fuel economy expressed on a mass basis and engine efficiency expressed on an energy basis.

The principal reason for the higher ISFC of ethanol is its significantly lower lower heating value (LHV). Absolute ethanol has an LHV of approximately 26.8 MJ kg^{-1}, whereas commercial gasoline typically possesses an LHV of approximately 42–43 MJ kg^{-1}. Consequently, a greater mass of ethanol must be supplied to the engine to provide the same amount of chemical energy required to produce an equivalent indicated power output. Even though ethanol converts the supplied energy more efficiently, its lower energy density necessitates higher fuel consumption when expressed in kilograms per unit work [16], [32]. In addition to its lower heating value, ethanol requires a richer stoichiometric fuel-air ratio than gasoline. The stoichiometric air-fuel ratio of ethanol is approximately 9.0:1, compared with 14.7:1 for gasoline. Therefore, under stoichiometric operation, a larger mass of ethanol must be injected to maintain the required equivalence ratio. This characteristic further contributes to the higher mass-based fuel consumption observed throughout the experimental programme.

Despite the increase in fuel mass consumption, the higher ISFC of ethanol should not be interpreted as inferior engine performance. When fuel consumption is evaluated on an energy basis, the superior combustion characteristics of ethanol largely compensate for its lower heating value through improvements in combustion efficiency and indicated thermal efficiency. Earlier combustion phasing, faster flame propagation, and improved combustion completeness enable a greater proportion of the supplied chemical energy to be converted into useful work. Consequently, the increase in ISFC primarily reflects the lower energy content of ethanol rather than poor combustion performance. The observed reduction in ISFC with increasing engine load can also be explained by improvements in combustion quality.

As engine load increased, combustion became more stable, peak cylinder pressure increased, combustion duration decreased, and thermal efficiency improved. These changes collectively enhanced the conversion of chemical energy into indicated work, thereby reducing the amount of fuel required per unit output. The trend observed in the present investigation is consistent with established thermodynamic behaviour for naturally aspirated spark-ignition engines operating under stoichiometric conditions [16]. The present findings are in good agreement with recent investigations involving ethanol-fuelled gasoline direct-injection engines. Li et al. reported that although ethanol increased indicated thermal efficiency, its lower heating value resulted in consistently higher specific fuel consumption compared with gasoline [17]. Similarly, Zhang et al. observed that ethanol required a greater injected fuel mass to achieve equivalent engine loads despite exhibiting superior combustion performance and improved thermal efficiency [18].

Anselmi et al. likewise concluded that renewable oxygenated fuels generally display higher mass-based fuel consumption because of their lower energy density, even when combustion quality and engine efficiency are significantly improved [19]. From an engineering perspective, the higher ISFC associated with ethanol should be considered together with its environmental and combustion advantages. Ethanol is produced from renewable biomass resources and contributes to reduced life-cycle greenhouse gas emissions. Furthermore, its high-octane number permits operation at higher compression ratios and more advanced ignition timing, offering opportunities for future engine designs to recover much of the apparent fuel consumption penalty. Modern high-compression GDI engines specifically optimized for ethanol have demonstrated substantial improvements in overall efficiency while maintaining acceptable fuel economy on an energy basis [34], [38]-[42].

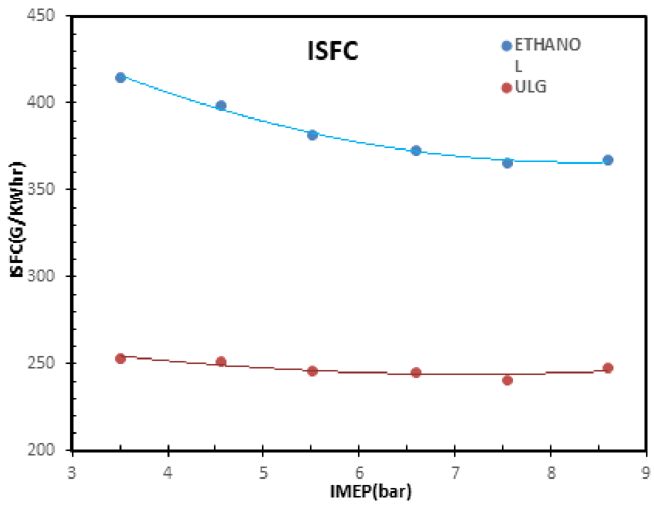


Fig.3.7 Indicated Specific Fuel Consumption for ethanol and ULG 3.5bar-8.5bar

Overall, the ISFC results demonstrate that ethanol requires a greater fuel mass than gasoline to produce equivalent indicated work because of its lower heating value and lower stoichiometric air-fuel ratio. Nevertheless, the superior combustion characteristics of ethanol—including higher combustion efficiency, earlier combustion phasing, shorter combustion duration, and increased indicated thermal efficiency—confirm that the observed increase in ISFC is fundamentally an energy-density effect rather than a limitation of the combustion process. These findings reinforce the potential of ethanol as an effective renewable fuel for high-efficiency gasoline direct-injection engines when evaluated using both energy-based efficiency and environmental performance criteria.

Gaseous Emissions Characteristics

The combustion process in spark-ignition engines directly influences the formation of gaseous exhaust emissions. The principal regulated gaseous emissions produced during gasoline direct-injection (GDI) engine operation include carbon monoxide (CO), nitrogen oxides (NO_x), and unburned hydrocarbons (HC). The concentrations of these pollutants are governed by combustion temperature, air-fuel ratio, combustion completeness, flame propagation, in-cylinder mixture distribution, and fuel chemical composition. Consequently, variations in combustion phasing, combustion duration, and combustion efficiency, as discussed in Section 3.1, are reflected in the measured exhaust emissions [16], [41], [24].

The oxygenated molecular structure of ethanol significantly alters the combustion chemistry compared with conventional gasoline. Ethanol contains approximately 35 wt.% oxygen, which promotes more complete oxidation of intermediate combustion products and improves combustion completeness. Furthermore, its high latent heat of vaporization produces substantial charge cooling, influencing peak combustion temperature and consequently affecting thermal NO_x formation. The combined influence of these properties results in distinct emission characteristics that differ from those of gasoline [17], [18], [43]. The following subsections examine the effects of fuel type on carbon monoxide, nitrogen oxides, and unburned hydrocarbon emissions over the investigated operating range of 3.5–8.5 bar IMEP at a constant engine speed of 1500 rpm.

Carbon Monoxide (CO) Emissions

Carbon monoxide is primarily produced during incomplete combustion when insufficient oxygen is available for the complete oxidation of carbon to carbon dioxide. In gasoline direct-injection engines, CO formation is strongly influenced by locally rich fuel-air mixtures, flame quenching near combustion chamber walls, spray-wall impingement, and incomplete oxidation during the expansion process [16], [41].

Figure 8 presents the variation of carbon monoxide emissions with engine load for gasoline and absolute ethanol. For both fuels, CO emissions increased gradually with increasing IMEP. This increase is expected because higher engine loads require greater fuel delivery, producing richer local mixture regions within the combustion chamber and increasing the likelihood of incomplete oxidation during combustion. Although both fuels exhibited increasing CO emissions with load, ethanol consistently produced significantly lower CO concentrations than gasoline throughout the investigated operating range.

The reduction was particularly evident at medium and high engine loads, where ethanol maintained superior combustion efficiency despite increased fuel delivery. The lower CO emissions observed with ethanol are primarily attributed to its chemically bound oxygen, which promotes more complete oxidation of carbon monoxide into carbon dioxide during the later stages of combustion. The additional oxygen improves oxidation kinetics within locally fuel-rich regions where gasoline combustion may remain incomplete. Furthermore, the shorter combustion duration and earlier combustion phasing achieved by ethanol increase combustion completeness before exhaust valve opening, thereby reducing residual carbon monoxide in the exhaust gases.

The improved combustion stability observed in Section 3.1.1 also contributed to the reduction in CO emissions. Lower cyclic variability results in more repeatable combustion events, reducing the occurrence of incomplete combustion cycles that contribute disproportionately to carbon monoxide formation. Similarly, the higher combustion efficiencies reported in Section 3.1.4 indicate that a larger fraction of the supplied fuel energy was released through complete oxidation, further explaining the consistently lower CO emissions associated with ethanol. These findings are consistent with recent experimental investigations involving oxygenated fuels in GDI engines. Li et al. reported that ethanol significantly reduced CO emissions because its oxygen content enhanced oxidation reactions during combustion [17].

Zhang et al. similarly observed lower CO emissions for pure ethanol under stoichiometric operating conditions, attributing the reductions to improved combustion completeness and faster flame propagation [18]. Anselmi et al. further demonstrated that renewable oxygenated fuels consistently decrease CO emissions through enhanced oxidation chemistry and improved mixture utilization [19]. Overall, the CO emission results demonstrate that ethanol substantially improves combustion completeness compared with gasoline, confirming the combustion efficiency trends discussed earlier and highlighting one of the principal environmental advantages of oxygenated renewable fuels.

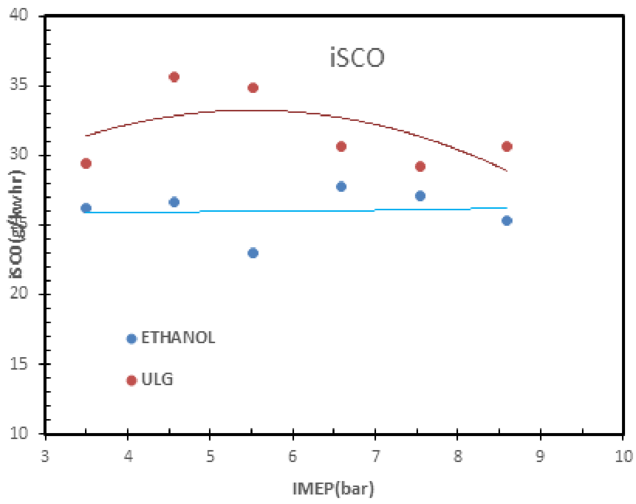


Fig.3.8 Instantaneous specific CO emissions for ULG and Ethanol 3.5-8.5

Nitrogen Oxides (NO_x) Emissions

Nitrogen oxides are formed primarily through the thermal Zeldovich mechanism when the burned gas temperature exceeds approximately 1800 K in the presence of sufficient oxygen and adequate residence time. Consequently, NO_x formation depends principally on peak combustion temperature, combustion duration, local oxygen concentration, and combustion phasing [16], [44]. Figure 9 illustrates the variation of NO_x emissions with engine load for gasoline and ethanol. For both fuels, NO_x emissions increased with increasing IMEP owing to the higher combustion temperatures associated with increased engine load and greater heat release. Higher cylinder pressures and elevated flame temperatures promote thermal NO formation, producing the characteristic increase observed for both fuels. Despite ethanol producing higher peak cylinder pressures and improved combustion efficiency, the measured NO_x emissions remained comparable to or slightly lower than those of gasoline throughout much of the investigated operating range.

This behaviour reflects the competing effects of ethanol's combustion properties. On one hand, ethanol's earlier combustion phasing and faster combustion increase heat release intensity, which could potentially elevate combustion temperatures and increase thermal NO_x formation. On the other hand, ethanol possesses a substantially higher latent heat of vaporization than gasoline. During the fuel vaporization process, considerable heat is absorbed from the incoming charge, producing significant charge cooling. This cooling effect lowers the initial in-cylinder temperature before ignition, reducing peak flame temperatures and suppressing thermal NO_x formation. Under the present experimental conditions, the charge-cooling effect appears to have offset much of the temperature increase associated with improved combustion, resulting in NO_x levels comparable to or slightly below those of gasoline.

Another important factor is ethanol's shorter combustion duration. Faster combustion reduces the residence time during which burned gases remain at elevated temperatures, thereby limiting the duration available for thermal NO formation. Consequently, although combustion efficiency improved, the overall NO_x production did not increase proportionally. These observations agree well with previous investigations on ethanol-fuelled GDI engines. Zhang et al. reported that ethanol maintained NO_x emissions comparable to gasoline despite higher combustion efficiency because evaporative cooling moderated peak combustion temperatures [17]. Li et al. similarly concluded that the competing influences of improved combustion and charge cooling determine the final NO_x emissions of ethanol-fuelled spark-ignition engines [20].

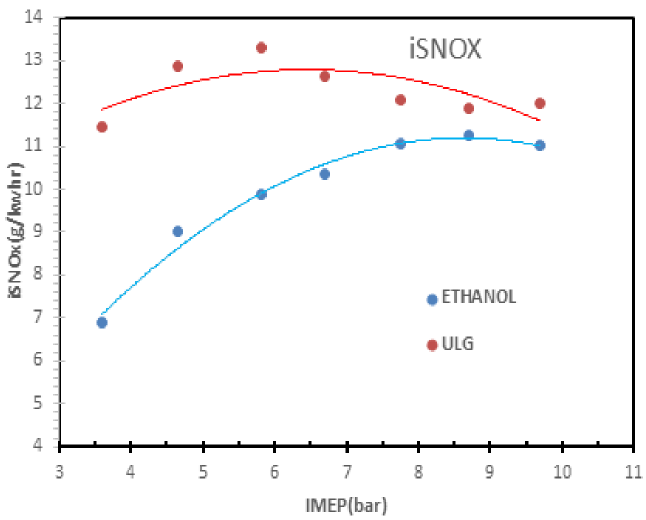


Fig.3.9. Instantaneous Specific NOx Emissions for ULG and Ethanol 3.5- 8.5 bar

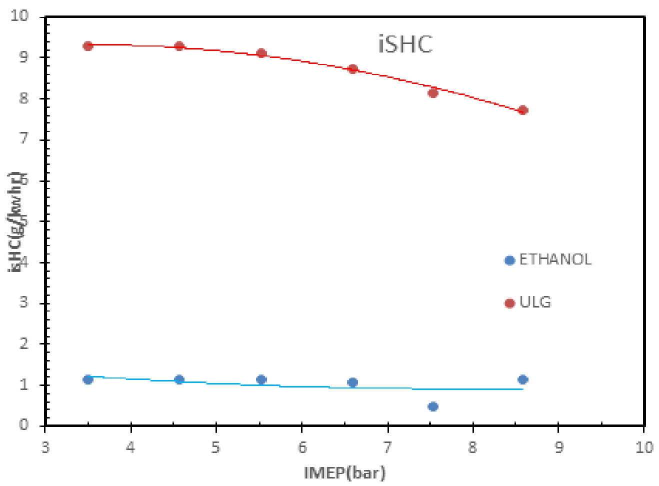
The present results therefore demonstrate that ethanol achieves improved combustion performance without a corresponding increase in NO_x emissions, representing a favourable balance between engine efficiency and environmental performance.

Unburned Hydrocarbon (HC) Emissions

Unburned hydrocarbon emissions originate from fuel that escapes complete oxidation during combustion. Major sources include flame quenching near combustion chamber walls, fuel trapped within piston ring crevices, wall wetting associated with direct fuel injection, incomplete combustion of locally rich mixtures, and cycle-to-cycle combustion variability [19], [43]-[46]. Figure 10 presents the variation of hydrocarbon emissions with engine load for gasoline and ethanol. Hydrocarbon emissions decreased gradually with increasing IMEP for both fuels, indicating progressively more complete combustion at higher engine loads. Increased cylinder temperatures and faster flame propagation at higher loads improve oxidation of unburned fuel, thereby reducing HC emissions. Throughout the investigated operating range, ethanol consistently produced lower HC emissions than gasoline.

The reduction may be attributed primarily to the oxygen contained within the ethanol molecule, which promotes more complete oxidation of hydrocarbons during combustion. In addition, the improved combustion stability, earlier combustion phasing, and shorter combustion duration observed for ethanol reduced the likelihood of partial combustion and flame extinction, thereby decreasing the amount of unburned fuel discharged into the exhaust system. The lower HC emissions also support the combustion efficiency results presented in Section 3.1.4, where ethanol exhibited consistently higher combustion efficiencies than gasoline. Improved combustion completeness naturally results in lower concentrations of partially oxidized combustion products, including unburned hydrocarbons.

These observations are consistent with numerous experimental investigations on oxygenated fuels in GDI engines. Li et al. reported significant reductions in HC emissions with ethanol because of improved oxidation and enhanced flame propagation [20],[40] Zhang et al. similarly observed lower HC emissions for ethanol across a wide engine load range, while Anselmi et al. concluded that renewable oxygenated fuels consistently reduce HC emissions by improving combustion completeness and limiting wall-quenching effects [21], [22]. Overall, the HC emission results further confirm the superior combustion characteristics of ethanol demonstrated throughout the combustion analysis. The reductions in HC emissions, together with the lower CO emissions and comparable NO_x emissions, illustrate the environmental benefits of ethanol while maintaining improved engine performance.



Fig, 10. IsHC emissions for ethanol and ULG 3.5bar imep-8.5bar imep

IV. CONCLUSIONS

This study experimentally compared the combustion, performance, and gaseous emission characteristics of commercial unleaded gasoline (ULG95) and absolute ethanol in a spray-guided gasoline direct-injection (GDI) engine operating at 1500 rpm under stoichiometric conditions over an IMEP range of 3.5–8.5 bar. The major conclusions are as follows:

Ethanol consistently produced higher peak cylinder pressure, earlier combustion phasing (MFB50), shorter combustion duration, and lower COVIMEP values than gasoline, demonstrating superior combustion quality and stability.

Higher combustion efficiency and indicated thermal efficiency were achieved with ethanol because of its oxygenated nature, faster flame propagation, and greater resistance to knock, which allowed combustion to occur closer to the optimum MBT condition. Although ethanol exhibited higher indicated specific fuel consumption than gasoline, this was primarily due to its lower heating value rather than poorer combustion performance.

Ethanol significantly reduced carbon monoxide (CO) and unburned hydrocarbon (HC) emissions while maintaining NO_x emissions comparable to or slightly lower than gasoline because of the combined effects of improved combustion and charge cooling.

Overall, the results demonstrate that absolute ethanol is a promising renewable fuel for spray-guided GDI engines, offering improved combustion performance, higher thermal efficiency, and lower gaseous emissions than conventional gasoline under the investigated operating conditions.

Recommendations

Based on the findings of this study, the following recommendations are proposed:

- Investigate ethanol performance over a wider range of engine speeds and loads.
- Evaluate particulate number (PN) and particulate mass (PM) emissions for ethanol-fuelled GDI engines.
- Study the influence of injection pressure, injection timing, and ethanol-gasoline blends on combustion and emissions.
- Conduct long-term durability and material compatibility studies for sustained ethanol operation.
- Apply advanced combustion diagnostics and CFD modelling to further understand ethanol combustion and emission formation mechanisms.

Nomenclature

Symbol / Abbreviation	Definition
ATDC	After Top Dead Centre
BDC	Bottom Dead Centre
BTE	Brake Thermal Efficiency
CA	Crank Angle
CO	Carbon Monoxide
CO_2	Carbon Dioxide
DISI	Direct-Injection Spark-Ignition
GHG	Greenhouse Gas
HC	Unburned Hydrocarbons
HRR	Heat Release Rate
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure
MF	2-Methylfuran
MTHF	2-Methyltetrahydrofuran
NO_x	Nitrogen Oxides
RON	Research Octane Number
SI	Spark-Ignition
TDC	Top Dead Centre

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