Effects of High-Ethanol Blends on Performance and Emissions of a Spark-Ignition Engine with Single and Dual Injector Configurations

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Effective utilization of high-ethanol fuels in spark-ignition (SI) engines requires injection configurations capable of accommodating significant variations in fuel properties. This study experimentally investigates the performance and emission characteristics of an SI engine fueled with gasoline–ethanol blends (E25, E50, E85, and E100) relative to baseline gasoline (E0), employing single- and dual-port fuel injection configu– rations. Experiments were conducted at a fixed engine speed of 6500 rpm over a range of load conditions. For consistent fuel-to-fuel comparison, gasoline-defined throttle settings were preserved while the injected fuel quantity and load were adjusted to sustain steady speed. Injector mass-flow characteristics were calibrated for each fuel to ensure accurate delivery. Results indicate that ethanol blends generally increased engine output but resulted in higher brake specific fuel consumption (BSFC) due to their lower heating value. At higher throttle openings (> 32%), gasoline exhibited superior performance. Under part-load conditions (< 28%), E25 combined with dual injection achieved brake power comparable to gasoline while improving brake thermal efficiency and reducing BSFC in both injection modes, reflecting enhanced mixture preparation enabled by ethanol-bound oxygen under low-throttle, low-airflow conditions. Increasing ethanol content significantly reduced CO, HC, NO_x, and CO_2 emissions, with dual injection providing additional emission benefits through shorter injection duration and improved mixture homogeneity. These findings demonstrate that dual-injector configurations can effectively enhance mixture preparation and reduce emissions when operating with high-ethanol fuels, offering their potential to improve part-load efficiency in SI engines.

Keywords: *Ethanol – Gasoline Blends, Dual port fuel injection, Spark-ignition engine, Engine performance, Exhaust emissions.*

1. INTRODUCTION

Growing concerns over greenhouse gas emissions and the transition toward low-carbon energy systems have intensified efforts to reduce reliance on fossil-derived fuels in spark-ignition (SI) engines, which are widely employed not only in transportation but also in distributed power generation, agricultural machinery, and small-scale industrial applications. This shift is further reinforced by global climate commitments, particularly the ambition to achieve net-zero carbon emissions by 2050 [1,2]. As a result, biofuels have attracted significant attention as renewable substitutes for conventional petroleum-based fuels. Among various biofuels, bioethanol accounts for the majority of global production and has emerged as one of the most promising fuels for spark-ignition (SI) engines, while also being regarded as environmentally beneficial because of its near-closed carbon cycle [3,4]. Ethanol offers several favorable properties, including a high octane number, inherent oxygen content, and compatibility with existing SI engine platforms, enabling its use in blended form without major structural modifications [5]. Consequently, low-level ethanol–gasoline blends have already been implemented in many countries [6,7].

Previous studies have shown that the influence of ethanol–gasoline blends on SI engine performance strongly depends on blending ratio and operating conditions. Experimental investigations with low-to-moderate ethanol contents (E10–E40) reported that small additions of ethanol can improve combustion and emissions under certain load and speed conditions, whereas higher blends tend to reduce brake power because of their lower heating value and higher latent heat of vaporization compared with conventional gasoline [8,9]. Similar trends were observed by Badrawada and Susastriawan [10], who reported more complete combustion and reduced HC and CO emissions for low ethanol fractions (5–10%) over a wide engine-speed range (4000–8000 rpm). Studies involving higher

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ethanol contents, including applications up to E85, indicated that while intermediate blends such as E20 can enhance power, torque, and emissions without major system modifications, higher ethanol contents (E55, E85) generally increase fuel consumption and constrain engine output [11]. In addition to these effects, ethanol's high octane number improves knock resistance and can enhance thermal efficiency and operational stability across varying conditions [12,13].

Despite these advantages, increasing ethanol content introduces several technical challenges for SI engine operation. The lower heating value of ethanol requires higher fuel delivery to maintain the desired load, while its high latent heat of vaporization and strong charge-cooling effects can shift mixture preparation toward incomplete vaporization and increased cycle-to-cycle variability, particularly under part-load conditions where intake airflow is limited [14,15]. Singh et al. [16] demonstrated that, under boosted intake pressures (1.0–1.5 bar) and ethanol blends ranging from 30% to 100%, the injection strategy and the resulting mixture forma–tion govern whether efficiency benefits can be realized: optimizing a single-injection mode (with varied timing and duration) improved efficiency, whereas a dual-injection split did not consistently outperform single injection under their tested intake conditions, indicating that split injection is not inherently beneficial without sufficient air motion and evaporation time, thereby highlighting the importance of injection strategy, am–bient conditions, and mixture formation in effectively utilizing ethanol fuel blends. Consistent with this mec–hanism-driven view, rapid ethanol evaporation can also promote localized vapor accumulation near the intake port, degrading mixture homogeneity when mixing is air-limited [17]. To address fuel-air interaction issues, intake-flow enhancements such as swirling vanes have been investigated and shown to improve mixture uniformity and combustion efficiency for low-to-moderate blends (up to E25). However, their effective–ness deteriorates at higher ethanol fractions because changes in spray atomization and evaporation behavior reduce the practical contribution of swirl to fuel–air mixing [18]. From a fuel delivery perspective, Bambang Sulistyo et al. [19] reported that introducing a dedicated ethanol injector in the intake port, together with increased ethanol injection pressure, improved mixing and reduced fuel consumption by enlarging the droplet–air interfacial area; this finding also emphasizes the role of fuel delivery strategy and mixture formation within the intake port in influencing the combustion efficiency of fuel blends. Nevertheless, the optimal injection pressure and targeting remain coupled to intake flow (throttle control) as well as injector placement, and therefore require further comprehensive investigation. Similarly, Purwanto et al. [20] showed that coordinated adjustment of injection timing (350°aTDC, 355°aTDC) and ignition timing (3°bTDC, 7°bTDC) reduced emis–sions for E10–E30 by mitigating knock and improving combustion completeness. The study also indicated that emissions from gasoline–ethanol blends are jointly influenced by engine control parameters, such as injec–tion and ignition timing, and the intrinsic physico–chemical properties of ethanol; however, broader timing ranges and higher ethanol fractions were not explored. In contrast to port-based strategies, Zhuang et al. [21] investigated a dual-fuel injection system combining high-pressure direct ethanol injection with port gasoline injection, thereby relocating mixture preparation from the intake manifold to the in-cylinder chamber. Their results emphasized that early ethanol injection promoted vaporization and charge uniformity, whereas delayed injection increased wall wetting and mixture hetero–geneity. Increasing injection pressure further intensified ethanol–piston interaction and produced locally rich regions near the exhaust valve, adversely affecting combustion quality. Similar observations were reported by Decheng Li et al. [22], highlighting that direct-injection-assisted ethanol delivery introduces stratifi–cation and wall-impingement phenomena that differ fundamentally from those encountered in conventional port fuel injection systems.

Collectively, these studies predominantly evaluate ethanol utilization under engine conditions that are recalibrated or re-optimized for each specific fuel, inc–luding adjustments to airflow, injection parameters, or ignition timing. While such approaches are effective for identifying achievable optimal performance, they make it difficult to distinguish the intrinsic effects of fuel properties from those introduced by concurrent modifi–cations of the air-handling system. In many practical steady-state SI engine applications—such as small-scale power generation—engines typically operate at fixed speed with predefined throttle settings, meaning that fuel flexibility must primarily be achieved through adaptation of fuel delivery rather than continuous re-optimization of the intake-air system. Under these constraints, increasing ethanol content requires a larger injected fuel mass to sustain the same operating condition, which alters mixture preparation even when the airflow command remains unchanged. However, the coupled behavior between increased ethanol fuel demand and a fixed intake-air configuration has not been systematically examined in previous experimental studies.

Accordingly, the present study experimentally eva–luates the effects of ethanol blending ratios (E25, E50, E85, and E100) on SI engine performance and emis–sions using single- and dual-port fuel injection confi–gurations under steady-state operation representa-tive of small-scale power generation. To directly assess the influence of fuel properties and injector configuration, identical intake-air conditions were maintained for all tested fuels. The throttle positions established with baseline gasoline operation were preserved for all tested fuels, while the injected fuel quantity and load were regulated to maintain a constant engine speed. This baseline-referenced experimental framework enables direct fuel-to-fuel comparison under non-reoptimized air-handling conditions, thereby isolating the effects of fuel–injection configuration on charge preparation and engine performance. Fuel-specific injector calibration was implemented to ensure accurate mass delivery for each blend, and engine performance parameters including brake power, brake specific fuel consumption (BSFC), brake thermal efficiency (BTE), and gaseous emissions (CO, CO_2, NO_x, and HC), were analyzed. This approach provides experimental evidence of how injector configuration governs mixture preparation under fixed intake-air flow conditions and consequently influences engine performance during high-ethanol operation, offering a practical basis for fuel-flexible SI engine operation under realistic steady-state conditions.

2. MATERIALS AND METHODS

2.1 Baseline-throttle-referenced fuel-adaptive fueling strategy

In this study, fuel delivery for gasoline–ethanol blends was regulated using a baseline-throttle-referenced fuel-adaptive fueling strategy implemented through an Adaptive Fuel Injection Control (AFIC) algorithm. The objective of this approach was to maintain identical air-handling conditions defined by gasoline operation while adapting the injected fuel mass to accommodate the physicochemical differences of ethanol-containing fuels. Baseline experiments were first conducted using gasoline (E0) at a constant engine speed of 6500 rpm. For each load condition, the relationship between engine load and throttle opening was established, and the corresponding throttle positions were recorded as reference values. These gasoline-defined throttle positions were subsequently fixed and applied unchanged to all ethanol–gasoline blends (E25–E100), thereby enforcing a gasoline-referenced airflow throughout the experiments. An intermediate control unit was introduced between the original engine electronic control unit (ECU) and the injectors to acquire the gasoline injector pulse width signal under the baseline conditions. Using the injector flow characteristics previously calibrated for gasoline, the controller calculated the actual injected gasoline mass (m_{fE0act}). As illustrated in the control schematic (Figure 1, where E50 is used as the representative sample), this measured fuel quantity served as the reference for determining the required fuel mass of the ethanol blend.

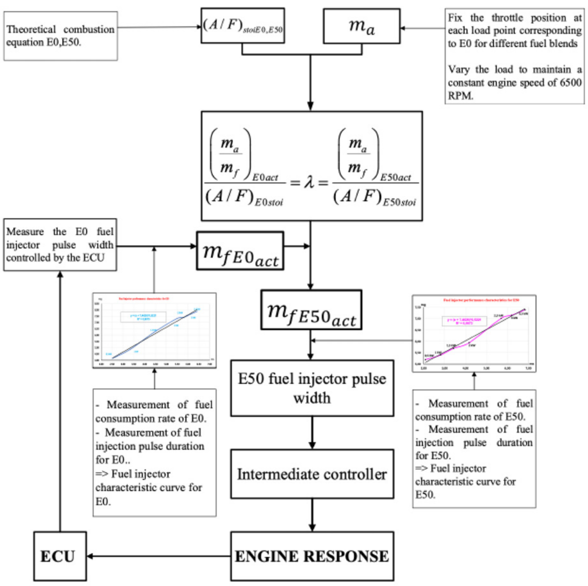

Figure 1. Control flowchart of the baseline-throttle-referenced fuel-adaptive injection strategy

For each gasoline-ethanol blend, the target injected mass ($m_{fE50act}$) was determined by conserving the excess-air ratio (λ) relative to the gasoline baseline, scaling the reference fuel mass according to the stoichiometric air–fuel ratio of the corresponding blend while maintaining identical throttle position and engine speed. Based on the experimentally determined injector flow characteristics of each fuel, the controller computed the corresponding injector pulse width and issued a corrected injection command to the actuators. This fuel-adjusted injection signal was continuously updated in a closed-loop manner, together with dynamometer load regulation, to stabilize engine operation at 6500 rpm for all test conditions. By maintaining gasoline-referenced throttle settings while adaptively modifying fuel delivery, the method ensures that observed variations in engine performance arise primarily from fuel-property effects and injector configuration rather than from re-optimization of the air-handling system, enabling controlled comparison of single- and dual-injector configurations under non-reoptimized air–flow conditions.

2.2 Injector flow characterization method for fuel-adaptive calibration

Figure 2 presents the procedure used to characterize injector flow behavior for gasoline and ethanol–gasoline blends under the same steady-state operating framework described previously. The engine speed was maintained at 6500 rpm while load was varied, and the gasoline-referenced throttle positions were retained for all tested fuels (E0, E25, E50, E85, and E100).

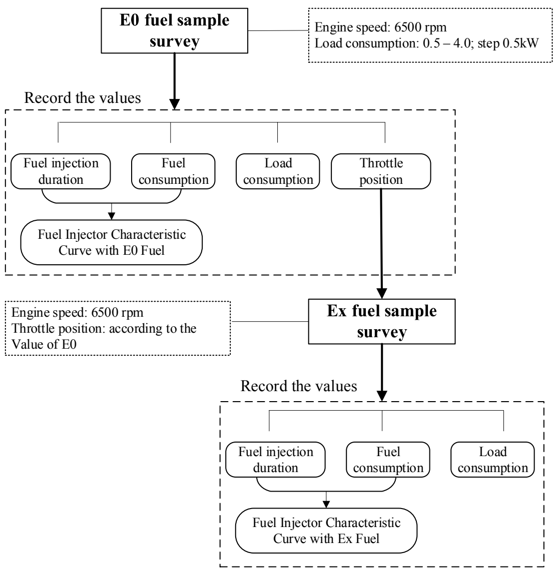

Figure 2. Experimental procedure for fuel-specific injector flow characterization

This ensured that injector characterization was performed under identical air-handling conditions to those used during the comparative engine experiments. For each fuel, the relationship between the injected fuel mass per cycle and the commanded injection duration was experimentally determined and expressed using a first-order linear model as in equation (1):

$$y = (x - a) \times b \tag{1}$$

where y is the actual injected fuel mass per engine cycle, and x is the nominal injector energizing duration commanded by the controller. The injector opening delay, characterized by the parameter *a*, reflects the time required for the injector to initiate opening and establish a stable fuel flow, while the parameter *b* corresponds to the characteristic fuel injection rate of the injector. This formulation defines an effective injection duration (x−a), allowing the injected mass to be related directly to measurable injector flow characteristics rather than relying on ECU-commanded duration alone. Consequently, the model provides a consistent physical basis for quantifying how fuel properties alter injector delivery behavior.

Figure 3 shows the resulting injector characteristic curves for all tested fuels. These fuel-specific mappings were implemented in the AFIC algorithm to convert the required injected mass into the corresponding injection duration for each blend, thereby enabling accurate fuel-adaptive control without modifying the baseline air-handling settings.

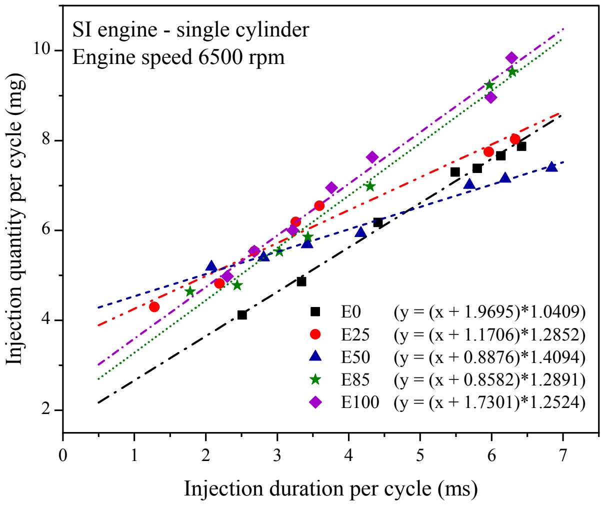

Figure 3. Fuel-dependent injector mass-flow characteristics

2.3 Modified intake manifold for dual injector configuration

In the present work, the fuel injection mass of the gasoline-ethanol blends is higher than that of gasoline under the same operating conditions due to the inverse relationship between fuel mass and the calorific value of the blend. Furthermore, the rapid evaporation of ethanol-containing mixtures poses issues for proper mixing with the intake airflow within the intake port. Therefore, a modified intake manifold was developed to enable the installation of a dual-port fuel injection configuration, as shown in Figure 4. The injectors are arranged symmetrically at an inclination of 12° relative to the vertical axis, with their sprays directed toward the intake-valve region to maintain targeting comparable to the baseline configuration. This arrangement ensures stable fuel delivery and repeatable spray introduction into the intake airstream while preserving the original engine configuration and the same air-handling control used during gasoline operation. The manifold modification was not intended to optimize in-cylinder flow or mixture formation, but rather to provide a practical structural arrangement that allows comparison between injector configurations. All other intake-system characteristics were retained so that any observed differences in engine behavior could be attributed primarily to fueling strategy and fuel properties rather than to geometric changes.

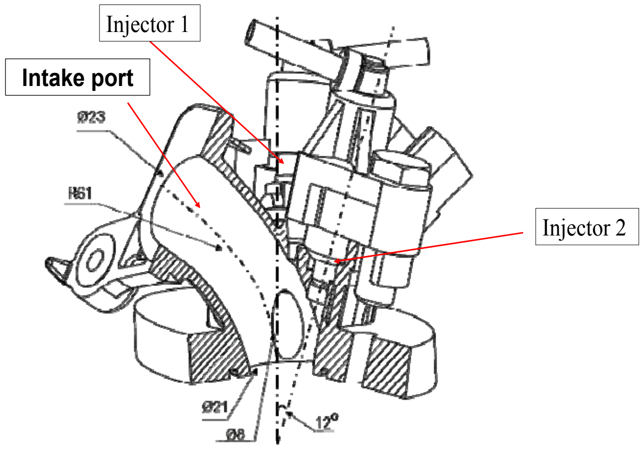

Figure 4. Modified intake manifold enabling dual port-fuel-injector configuration.

2.4 Experimental set-up

The experimental platform consisted of a single-cylinder spark-ignition engine with a swept volume of 109.1 cm³ and a compression ratio of 9.3:1. An electronic fuel injection (EFI) system, along with an electronic ignition module, was integrated into the engine. Engine loading was implemented through a generator-based electrical load system. The engine shaft drove a 12 kW three-phase AC generator via a flexible coupling, as illustrated in Figure 5.

The brake load was imposed by regulating the generator excitation current, which controlled the electromagnetic resisting torque applied to the crankshaft. The generated electrical power was dissipated through an external resistive load bank, enabling stable and continuously adjustable steady-state operation representative of practical small-scale power-generation conditions. Engine torque was measured using an INA114-based strain-gauge load cell integrated into the generator assembly. Fuel consumption was determined gravimetrically using a high-precision Digi DS502 electronic balance with a resolution of 0.01g. Engine speed was continuously monitored using an encoder sensor to ensure operation at the prescribed constant speed of 6500 rpm throughout all test conditions.

The injector drive signal issued by the original engine control unit (ECU) was intercepted and recorded in real time by a microcontroller-based interface. This signal was subsequently processed using the Adaptive Fuel Injection Control (AFIC) algorithm described in Section 2.1 to calculate the corrected injection duration required for each tested fuel. The control system operated in a closed loop in conjunction with load regulation to maintain the gasoline-referenced throttle setting and constant engine speed. A Qrotech Qro-402 gas analyzer was used to continuously measure exhaust gases, including NO_x, CO, CO_2, and HC, for the purpose of assessing emission characteristics.

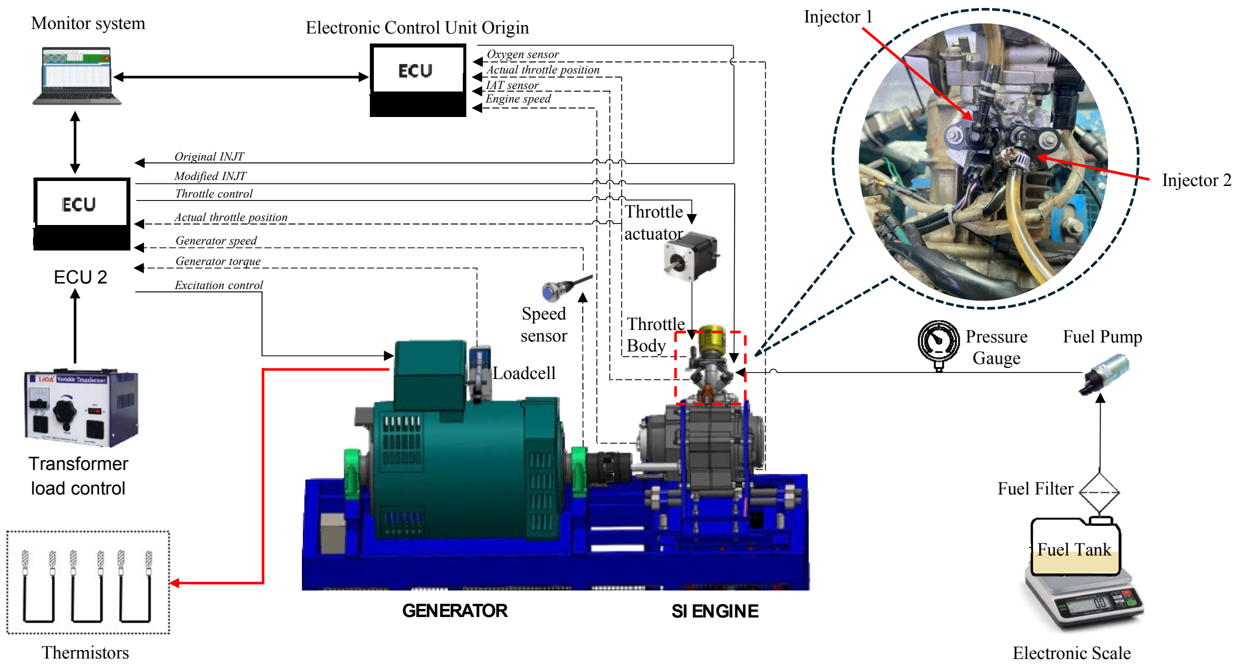

Figure 5. Schematic of the experimental set-up with generator-based electrical loading

2.5 Fuel properties and test conditions

Five fuels were tested: gasoline (E0) and ethanol–gaso–line blends containing 25% (E25), 50% (E50), 85% (E85), and 100% (E100) ethanol by volume. These fuels were selected to examine the influence of increasing ethanol content on engine performance and emission characteristics under the baseline-throttle-referenced operating framework. The main physicochemical pro–perties are listed in Table 1.

Table 1 Properties of test fuels

Fuel properties	Values				
	E0	E25	E50	E85	E100
Octance number	95.4	100	101	103	109
Stoichiometric air-fuel ratio	15.14	13.58	12.04	9.9	9
Lower heating value [MJ/kg]	42	38.2	29.2	29.31	27
Density (g/cm^3)	0.75	0.76	0.77	0.79	0.81
Initial boiling point [ºC]	47	43	43.5	60	80
Final boiling point [ºC]	186	190	181	82	81

With increasing ethanol fraction, the lower heating value (LHV) decreases progressively because of ethanol's intrinsic oxygen content, resulting in a reduced energy density compared with gasoline [23]. For E100, the LHV is approximately 36% lower than that of E0. Consequently, a greater injected fuel mass is required to achieve the same load under identical intake-air conditions. Ethanol also exhibits a lower stoichiometric air–fuel ratio and a higher latent heat of vaporization, which affect mixture preparation through charge-cooling effects and modified evaporation behavior. In addition, changes in fuel density influence injector flow characteristics, necessitating fuel-specific calibration as implemented through the AFIC strategy. Conversely, the higher octane number of ethanol enhances knock resistance and supports stable combustion under high load [24,25,26]. Changes in the distillation behaviors caused by ethanol blending further modify the phase-transition process during intake. Relative to gasoline, ethanol-containing fuels vary the temperature range over which fuel evaporation occurs, promoting earlier vapor formation in the intake port and altering local vapor distribution. Under the baseline-throttle-referenced conditions of the present study, this primarily affects mixture preparation rather than the global air–fuel ratio. The modified evaporation pathway can enhance diffusion and fuel–air interaction, contributing to lower combustion temperatures and reduced NO_x formation; however, it may also increase susceptibility to premature vapor formation within the fuel delivery system, potentially leading to vapor lock [27]. These thermophysical effects highlight the importance of appropriately managing fuel delivery characteristics to maintain stable mixture preparation when operating with high-ethanol blends under non-reoptimized airflow conditions.

Table 2: Experimental conditions

Fuels	E0, E25, E50, E85, E100
Engine speed (rpm)	6500
Engine load / step (kW)	0.5 – 4.0; 0.5
Throttle position (%)	12, 16, 21, 24, 28, 33, 37, 44
Fuel injection strategy for ethanol blends	Case 1: using a single injector during the intake cycle Case 2: using two injectors with the injection time split equally between each injector during the intake cycle

The experimental conditions used in this study are detailed in Table 2. Engine speed was maintained at a constant 6500 rpm by coordinated regulation of the generator excitation current and fuel injection quantity. Engine load was varied from 0.5 to 4.0 kW in increments of 0.5 kW. For ethanol-blended fuels, the throttle position at each load point was kept identical to that established during the corresponding gasoline test. Two injection configurations were examined. In Case 1, fuel was delivered through a single injector, with injection duration adjusted to supply the required mass for each blend. In Case 2, two injectors operated simultaneously during the intake stroke, sharing the total commanded fuel mass within each engine cycle to improve fuel distribution under the increased fueling demand associated with high-ethanol blends. In both cases, injection duration was determined using the Adaptive Fuel Injection Control (AFIC) algorithm. All measurements of fuel consumption, engine performance, and emissions were repeated under identical conditions to ensure repeatability, and the averaged values were used for analysis. The fuel supply system was flushed prior to each fuel change to prevent cross-contamination between test fuels.

3. RESULTS AND DISCUSSION

3.1 Brake power characteristics

Figure 6 presents the variation in power output as a function of load consumption for gasoline and ethanol-gasoline blends (E25, E50, E85, and E100) under single- and dual-injector configurations across different throttle-opening levels. At low throttle openings (12%–21%), ethanol-gasoline blends produced higher power output than gasoline, with average improvements ranging from 8.24% to 45.38%, particularly for higher ethanol fractions (E50–E100).

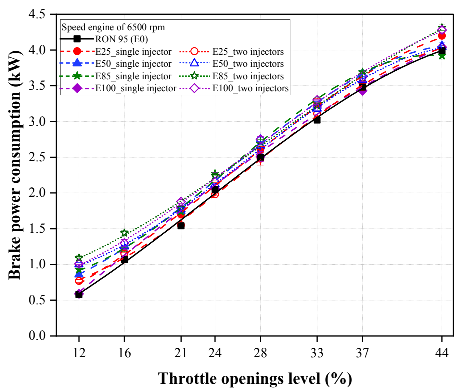

Figure 6. Effect of ethanol blending ratio and injector configuration on engine power under baseline-throttle-referenced operation

Under the baseline-throttle-referenced experimental framework employed for ethanol blends and owing to their lower stoichiometric air–fuel ratio compared to gasoline, a larger injected fuel mass was required to compensate for the lower heating value. The reduced stoichiometric air demand allowed this increased fuel quantity to remain compatible with the limited intake-air supply at part-load conditions. As a result, the combined effect of higher injected mass and lower stoichiometric air requirement enhanced the total chemical energy released per cycle without inducing severe air deficiency, thereby improving brake power under low throttle openings. This behavior reflects the intrinsic coupling between elevated fuel demand and fixed air-handling conditions imposed in the present experimental strategy. As throttle opening increases, the power advantage of ethanol blends gradually diminishes, with the maximum difference reduced to less than 7.35% at higher loads. The airflow becomes less restrictive in these conditions, leading to a decrease in the relative influence of compensatory fuel enrichment. Consequently, the reduced energy density of ethanol becomes the dominant factor [28].

A similar trend is observed for the dual-injector configuration. Notably, at low throttle openings (12%–21%), the dual-injector mode with high-ethanol blends yields greater power enhancement than the single-injector mode. Under fixed throttle constraints, splitting the required fuel mass between two injectors shortens injection duration per injector and promotes more uniform spray introduction into the intake airstream. This improves mixture preparation and combustion stability at part-load conditions where air motion is limited, thereby amplifying the power benefit of ethanol operation [29]. Beyond 21% throttle opening, however, this difference between single- and dual-injector configurations becomes less pronounced, suggesting that mixture preparation is less sensitive to injector configuration as airflow increases. Among all blends, E25 consistently exhibited the smallest deviation from gasoline across load levels, with differences of 8.35% and 4.94% under single- and dual-injector configurations, respectively. This indicates that moderate ethanol content can balance additional oxygen availability and fuel mass increase without substantially altering overall performance characteristics. Comparable observations were also reported by R. Contiu et al. [30].

3.2 Brake specific fuel consumption

Nevertheless, within the low throttle range of 12% – 24%, the BSFC of the E25 blend decreased by 19.24% and 7.71% relative to gasoline under the single- and dual-injector configurations, respectively. At part-load operation, where combustion completeness becomes more sensitive to mixture quality, the moderate oxygen content of E25 promotes more complete oxidation and improved energy conversion efficiency despite the limited intake-air supply. Under these conditions, the improvement in combustion effectiveness compensates for the lower energy density of the blend, resulting in reduced fuel consumption per unit brake power.

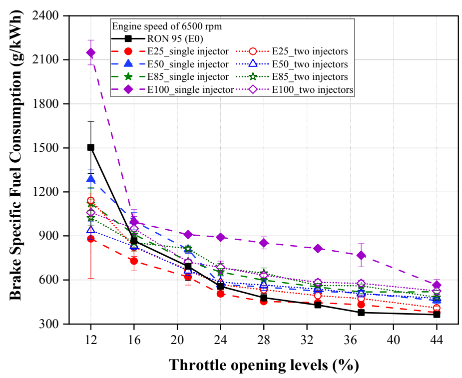

Figure 7. Effect of ethanol blending ratio and injector configuration on brake specific fuel consumption under baseline-throttle-referenced operation

For higher ethanol fractions (E50–E100), BSFC increased under single-injector operation, reflecting the dominant influence of lower fuel energy density. However, the dual-injector configuration consistently reduced BSFC compared to the single-injector mode, particularly for E100, which demonstrated an average reduction of 23.13% across the tested load conditions. By distributing the required fuel mass between two injectors, the injection duration per injector was shortened, improving fuel dispersion and charge preparation. This enhanced combustion stability and reduced incomplete combustion losses, thereby mitigating the BSFC penalty typically associated with high-ethanol fuels [31].

3.3 Brake thermal efficiency

Brake thermal efficiency (BTE) reflects the effectiveness with which the chemical energy of the injected fuel is converted into useful mechanical output, thereby integrating the combined effects of fuel consumption and brake power. Figure 8 presents the variation in BTE of gasoline and ethanol–gasoline blends for both injector configurations across different throttle-opening levels. At low throttle openings (12%–24%), ethanol blends displayed markedly higher BTE than gasoline, especially with maximum improvements of 65.86% and 78.46% recorded for E85 and E100 in the dual-injector configuration. This trend can be explained by part-load operation, where restricted airflow increases the sensitivity of fuel energy utilization to mixture preparation quality. In such conditions, the dual-injector mode combined with oxygenated ethanol blends facilitates more uniform fuel distribution [32].

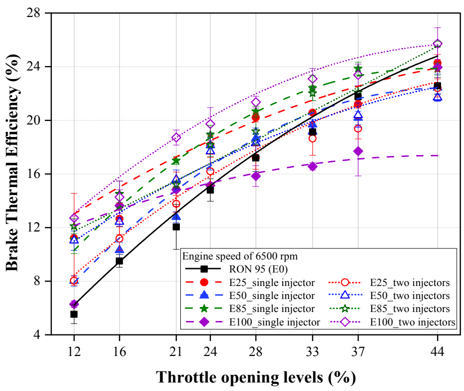

Figure 8. Variation of brake thermal efficiency with throttle opening as influenced by ethanol blending ratio and injector configuration

As throttle opening increases, the difference in BTE between gasoline and ethanol blends progressively narrows. Greater airflow availability mitigates the constraint imposed by the fixed air-handling condition, consequently decreasing the sensitivity of combustion to charge-preparation effects. As a result, efficiency becomes increasingly governed by the intrinsic energy content of the fuel rather than by injector configuration. Consequently, the higher heating value of gasoline becomes a more influential factor in determining overall thermal efficiency at higher loads. Interestingly, for E100 in the single-injector configuration at throttle openings above 21%, further increases in BTE become limited. The increased fuel requirement at higher ethanol fractions places additional demand on the fueling system, which can constrain the effectiveness of energy conversion when only one injector is utilized. For instance, extended injection duration may deteriorate spray dispersion quality. By contrast, the alternative configuration maintains more stable thermal efficiency across the same operating range. These results indicate that, within the fixed air-handling framework applied in this study, injector configuration influences the ability to sustain thermal efficiency during high-ethanol operation.

3.4 Exhaust gas emission characteristics

Figure 9 illustrates the carbon monoxide (CO) emis–sions of gasoline and ethanol–gasoline blends across throttle openings for both injector configurations. Over–all, E25 and E85 in the single-injector configuration, as well as E100 in both configurations, exhibited lower CO levels compared with gasoline. The reduced carbon intensity of ethanol-containing fuels and their inherent oxygen content contribute to improved carbon oxida–tion, parti–cularly at low-to-moderate throttle openings. However, within the throttle range of 16%–37%, noti–ceable fluc–tuations in CO emissions were observed for E85 under the dual-injector mode and for E50 under both injection configurations, with E25 exhibiting minor variations. These irregularities can be attributed to the increased fuel mass demand, which extends injection duration during the intake stroke. Under fixed intake airflow conditions, prolonged fuel injection may disturb the balance between fuel evaporation behaviors and air–fuel mixing, thereby leading to localized oxidation inefficiencies, particularly when additional fuel mass is supplied to compensate for the lower heating value of ethanol. Similar behavior associated with mixture inho–mogeneity and incomplete carbon oxidation has been reported by P. Sakthivel et al. [33]. When the throttle opening exceeds 37%, CO emission declines markedly, likely due to enhanced intake airflow that improves charge homogenization and supports more complete combustion processes.

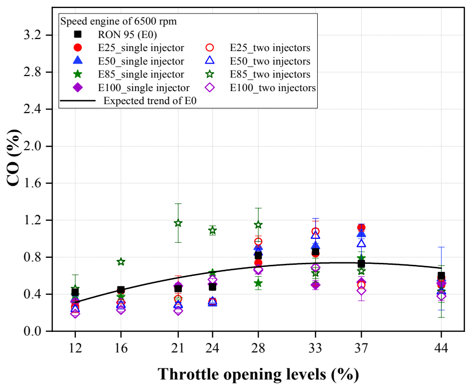

Figure 9. CO emissions of ethanol–gasoline blends under single- and dual-injector configurations across throttle-opening levels

Figure 10 presents the carbon dioxide (CO_2) emis–sions of gasoline and ethanol–gasoline blends across throttle-opening levels for both injector configurations. CO_2 concentration increases with throttle opening for all fuels, reflecting higher overall fuel consumption and greater carbon oxidation at elevated load conditions. Across the entire operating range, ethanol-containing fuels generally exhibit lower CO_2 emissions compared with gasoline. Under the single-injector configuration, average CO_2 reductions relative to E0 were 5.58%, 3.80%, 2.85%, and 15.65% for E25, E50, E85, and E100, respectively. This reduction is primarily attri–buted to the lower carbon content per unit mass of ethanol compared with gasoline, which decreases the total carbon available for oxidation despite the higher fuel mass required at elevated ethanol fractions [34]. In the dual-injector configuration, CO_2 emissions exhibit slight variations relative to the single-injector case, with average differences of 0.71%, 3.32%, 10.18%, and 0.29% for E25, E50, E85, and E100, respectively. These differences reflect changes in overall fuel utilization effectiveness rather than fundamental alterations in carbon chemistry. Since CO_2 formation is directly linked to the total carbon input and brake output level, injector configuration influences CO_2 primarily through its effect on fuel consumption efficiency, as demonstrated in Figure 7, rather than through mixture homogeneity alone [31].

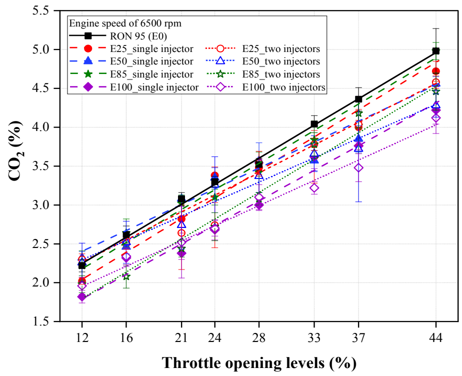

Figure 10. CO_2 emissions of ethanol–gasoline blends under single- and dual-injector configurations across throttle-opening levels

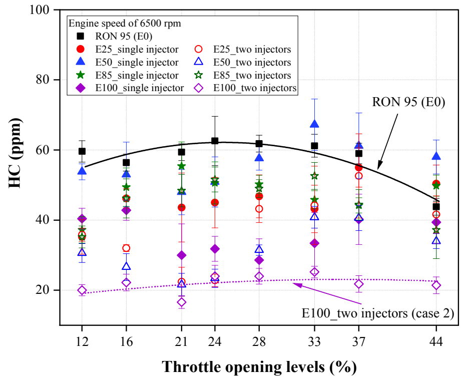

Figure 11. HC emissions of ethanol–gasoline blends under single- and dual-injector configurations across throttle-opening levels.

Hydrocarbon (HC) emissions provide an indicator of the unburned fuel fraction across operating conditions. Figure 11 compares HC emissions of gasoline and ethanol–gasoline blends for both injector configurations over the investigated throttle range. Across most operating points, ethanol-containing fuels exhibit lower HC emissions than gasoline. Under the single-injector configuration, average HC reductions of 19.89%, 1.81%, 16.40%, and 37.01% were observed for E25, E50, E85, and E100, respectively, relative to E0. This reduction is largely attributed to ethanol's inherent oxygen content and higher flame propagation speed, which facilitate improved oxidation of fuel-rich zones. This observation aligns with the findings reported by W. Purwanto et al. [20].

The dual-injector configuration yields consistently lower HC levels compared with the single-injector mode, with average reductions of 19.27%, 45.21%, 4.36%, and 37.49% for E25, E50, E85, and E100, res–pectively. By distributing the required fuel mass bet–ween two injectors, the system maintains more stable charge development and reduces localized fuel accumu–lation, thereby limiting hydrocarbon slip. The prono–unced HC reduction for high-ethanol blends further corroborates the improvements in energy conversion efficiency discussed in the BSFC and BTE analyses. At throttle openings exceeding 37%, HC emissions decrease substantially, reflecting enhanced oxidation capacity associated with increased airflow availability.

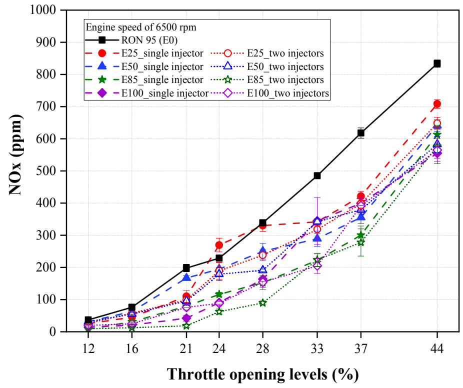

Figure 12. NO_x emissions of ethanol–gasoline blends under single- and dual-injector configurations across throttle-opening levels

Nitrogen oxide (NO_x) formation is strongly go–verned by combustion temperature and in-cylinder ther–mal history. Figure 12 presents NO_x emissions of gasoline and ethanol–gasoline blends across throttle-opening levels for both injector configurations. For all fuels, NO_x emissions increase with throttle opening, reflecting higher combustion temperatures and cylinder pressures at elevated load conditions. Compared with gasoline, ethanol-containing fuels exhibit significantly lower NO_x levels. The charge-cooling effect associated with ethanol's high latent heat of vaporization, combined with its lower heating value, reduces peak combustion temperature, thereby suppressing thermal NO_x formation. Similar trends have been reported in previous investigations [28,35].

Under the single-injector configuration, average NO_x reductions relative to E0 were 22.18%, 24.40%, 54.23%, and 52.70% for E25, E50, E85, and E100, respectively. These reductions become more prono–unced at higher ethanol fractions, consistent with the increasing contribution of evaporative cooling and lower flame temperature. In the dual-injector configu–ration, additional NO_x reductions are observed as compared with the single-injector case, with average decreases of 7.28%, 8.96%, 26.96%, and 6.93% for E25, E50, E85, and E100, respectively. Rather than fundamentally altering the chemical pathway of NOx formation, the dual-injector setup contributes to more uniform charge development and moderated local tem–perature gradients, which can further limit peak thermal conditions responsible for NOx generation.

4. CONCLUSIONS

This study investigated the performance and emission characteristics of high-ethanol blends in a spark-ignition engine operated under a baseline-throttle-referenced framework using single- and dual-injector configu–rations. Unlike conventional recalibrated approaches, throttle positions defined under gasoline operation were preserved, and fuel injection was adaptively adjusted to maintain constant engine speed. This methodology enabled the isolation of the interaction between incre–ased fuel mass demand and fixed air-handling condi–tions. The key conclusions are summarized as follows:

- In part-load operation (12–24% throttle opening), ethanol blends enhance brake power and thermal efficiency despite their lower heating value. The combination of increased injected fuel mass and reduced stoichiometric air requirement remains compatible with the limited intake airflow, thereby improving energy release per cycle. E25 demon–strated the most favorable balance between fuel consumption and efficiency. As throttle opening increases, the influence of fuel mass compensation diminishes, and engine performance becomes incre–asingly governed by intrinsic fuel energy content.
- Brake-specific fuel consumption increases with ethanol fraction; however, the dual-injector confi–guration mitigates this penalty for high-ethanol blends by stabilizing charge development and imp–roving overall energy conversion effectiveness under airflow-constrained conditions compared to the single-injector configuration.
- Emission results further demonstrate that ethanol fuels significantly reduce CO, HC, and NO_x emis–sions relative to gasoline, primarily due to enhanced oxidation characteristics and charge-cooling effects, while CO_2 reductions are associated with lower carbon intensity of the fuel.
- The dual-injector configuration provides additional benefits under airflow-constrained conditions, where elevated fuel mass demand challenges combustion stability. By improving charge consistency without modifying the air-handling system, the dual-injector strategy enhances performance and emission control for high-ethanol operation.

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NOMENCLATURE

y	Injected fuel mass per engine cycle
x	Nominal injection duration per cycle
a	Injector opening delay
b	Injector characteristic injection rate
m_{fE0act}	Actual injected fuel mass for gasoline (E0)
$m_{fE50act}$	Actual injected fuel mass for E50 blend
λ	Excess air ratio (lambda coefficient)
LHV	Lower Heating Value

Abbreviations

AC	Alternating Current
AFIC	Adaptive Fuel Injection Control
BSFC	Brake Specific Fuel Consumption
BTE	Brake Thermal Efficiency
CA	Crank Angle
CO	Carbon Monoxide
CO_2	Carbon Dioxide
ECU	Electronic Control Unit
EFI	Electronic Fuel Injection
E0	Gasoline with 0% ethanol by volume
E10	Ethanol–gasoline blend with 10% ethanol by volume
E20	Ethanol–gasoline blend with 20% ethanol by volume
E25	Ethanol–gasoline blend with 25% ethanol by volume
E30	Ethanol–gasoline blend with 30% ethanol by volume
E40	Ethanol–gasoline blend with 40% ethanol by volume
E50	Ethanol–gasoline blend with 50% ethanol by volume
E55	Ethanol–gasoline blend with 55% ethanol by volume
E85	Ethanol–gasoline blend with 85% ethanol by volume
E100	Neat ethanol
GPI	Gasoline Port Injection
HC	Hydrocarbons
IAT	Intake Air Temperature
INA114	Instrumentation Amplifier 114
INF	Injection Signal
NO_x	Nitrogen Oxides
RON	Research Octane Number
SI	Spark-Ignition

УТИЦАЈ МЕШАВИНА СА ВИСОКИМ САДРЖАЈЕМ ЕТАНОЛА НА ПЕРФОРМАНСЕ И ЕМИСИЈЕ МОТОРА СА ПАЉЕЊЕМ СВЕЋИЦОМ СА КОНФИГУРАЦИЈАМА ЈЕДНОГ И ДВА ИЊЕКТОРА

В.Т. Чау, В. Дај

Ефикасно коришћење горива са високим садржајем етанола у моторима са паљењем свећицом (СИ) захтева конфигурације убризгавања које могу да прихвате значајне варијације у својствима горива. Ова студија експериментално истражује карак–теристике перформанси и емисије СИ мотора који користи мешавине бензина и етанола (Е25, Е50, Е85 и Е100) у односу на основни бензин (Е0), користећи конфигурације убризгавања горива са једним и два отвора. Експерименти су спроведени при фиксној брзини мотора од 6500 о/мин у различитим усло–вима оптерећења. Ради доследног поређења горива, подешавања гаса дефинисана бензином су сачувана, док су количина убризганог горива и оптерећење подешени да би се одржала стална брзина. Карак–теристике масеног протока инјектора су калиб–рисане за свако гориво како би се осигурала тачна испорука. Резултати показују да мешавине етанола генерално повећавају снагу мотора, али резултирају већом потрошњом горива специфичном за кочење (БСПГ) због њихове ниже топлотне вредности. При већим отворима гаса (> 32%), бензин је показао супериорне перформансе. Под условима делимичног оптерећења (< 28%), Е25 у комбинацији са двост–руким убризгавањем постигао је кочиону снагу упо–редиву са бензином, уз побољшање термичке ефи–касности кочница и смањење кочења у оба режима убризгавања, што одражава побољшану припрему смеше коју омогућава кисеоник везан за етанол под условима ниског гаса и ниског протока ваздуха. Повећање садржаја етанола значајно је смањило емисије CO, HC, NO_x и CO_2, при чему двоструко убризгавање пружа додатне предности у погледу емисије кроз краће трајање убризгавања и побољ–шану хомогеност смеше. Ови налази показују да конфигурације са двоструким убризгавачима могу ефикасно побољшати припрему смеше и смањити емисије при раду са горивима са високим садржајем етанола, нудећи свој потенцијал за побољшање ефикасности делимичног оптерећења код мотора са силовим напајањем.