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Investigating the Impact of Ignition Timing Variations on Single-Cylinder Otto Engine Performance with E50 Fuel Blend

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Abstract: The exponential growth in motorized vehicle usage presents myriad challenges, encompassing environmental pollution and sustained energy shortages. To address these challenges, the exploration of sustainable energy alternatives is imperative, with ethanol-based fuels emerging as a viable option. This investigation delves into the performance of a single-cylinder Otto engine, with a focus on the effects of ignition timing variations using a 50% ethanol and 50% pertalite blend, denoted as E50. The ignition timing was systematically varied to standard, +2°, +4°, and -2°. The results demonstrated that the +4° ignition timing, in conjunction with E50, delivered superior performance, culminating in a maximum torque of 8.02 Nm at 4000 rpm and a peak power output of 4.15 kW at 8000 rpm. Concurrently, optimal engine efficiency was achieved, with the Brake Specific Fuel Consumption (BSFC) reaching its lowest value of 0.307 Kg/kW.h at 5000 rpm and Brake Thermal Efficiency (BTE) peaking at 36.10% at the same rotational speed. When contrasted with alternative fuels, the E50 blend resulted in an average torque reduction of 13.27% and a 14.46% decrease in power output. Despite this, significant enhancements in engine efficiency were observed. A 25.05% improvement in BSFC was noted, albeit with a reduction in fuel efficiency, while BTE experienced a 5.02% increase, indicative of augmented engine efficiency, particularly at the +4° ignition timing. This study underscores the potential of E50 and altered ignition timing in reducing reliance on fossil fuels, thus contributing to the transition towards sustainable energy solutions in the automotive sector.

Keywords: Single-cylinder Otto engine; Ignition timing; Performance evaluation; E50 fuel blend; Sustainable energy alternatives

1 Introduction

The pervasive utilization of motorized vehicles contemporaneously exerts a twofold impact on the environment and human existence. Despite playing a crucial role in enhancing human mobility and facilitating the transit of goods and services, these vehicles concurrently contribute to escalating energy shortages, air pollution, traffic congestion, and the occurrence of accidents [1, 2]. The combustion process within motor vehicle cylinders is identified as a major source of harmful exhaust emissions, comprising compounds such as carbon dioxide (CO_2), carbon monoxide (CO), nitrogen oxides (NOx), and hydrocarbons (HC) [3–5]. In spark ignition (SI) powered vehicles, the propulsion process is marred by the inadvertent release of hazardous exhaust gases, rich in chemicals such as CO_2, CO, NOx, and HC, thereby inflicting harm upon the fragile ecosystem [6–8]. Notably, motorized vehicles are responsible for emitting 8 kilograms of CO_2 per gallon of fuel combusted, accounting for 19% of global CO_2 emissions, while a formidable 70% of CO emissions emanate from vehicle exhausts worldwide [9].

A plethora of research endeavors has been undertaken with the aim of alleviating the petroleum crisis and mitigating air pollution originating from SI engines. A prominent approach entails the utilization of renewable fuels derived from biomass. In this context, the glucose content in biomass is subjected to a series of intricate processes, culminating in its transformation into ethanol fuel. The conversion process commences with fermentation, facilitated by microorganisms, followed by stringent purification procedures to attain the requisite ethanol concentration [10–12]. The adoption of biomass-derived ethanol not only serves as a sustainable alternative to conventional fossil fuels but also marks a significant step towards achieving energy sustainability. Through the exploration of biochemical processes and the application of advanced purification techniques, the foundation is being laid for a future wherein renewable, biomass-derived fuels significantly contribute to mitigating the environmental impact of motorized vehicles and addressing the challenges posed by the petroleum crisis [13, 14].

The optimization of thermal efficiency and performance in SI engines, when utilizing ethanol-gasoline blends, necessitates precision in the adjustment of compression ratios in accordance with the Research Octane Number (RON). Through such calibration, the preservation of engine components is ensured, and the full potential of mixed fuels is harnessed, resulting in maximized torque, power, and fuel efficiency while concurrently mitigating the risk of detrimental knocking phenomena due to misaligned compression settings. This underscores the imperative of meticulous engineering for the realization of efficient and robust SI engines [15–17]. The pivotal role of ignition timing adjustment in SI engines, particularly when fueled by ethanol-gasoline blends, is underscored by its impact on achieving optimal combustion. Through the precise manipulation of ignition timing, the fuel mixture is afforded efficient combustion, culminating in enhanced engine performance and a reduction in emissions. This delicate balance underscores the quest for efficiency and environmental stewardship in the domain of mixed-fuel SI engines [13, 18].

Harijono et al. [19] have conducted research on the utilization of bioethanol as an alternative fuel in Otto engines, employing a compression ratio of 90:1 and testing various ethanol-gasoline (pertalite) blends, comprising 10%, 15%, and 20% ethanol. The findings indicate that the use of a 10% ethanol blend (E10) results in the lowest observed maximum values for power, torque, and Specific Fuel Consumption (SFC), being 6.6 hp, 6.58 Nm, and 0.0120 kg/hp, respectively. In a related study by Sakthivel et al. [20], compression ratios of 9.4:1, 10.9:1, and 11.5:1 were examined in conjunction with a 30% ethanol-gasoline mixture (E30), achieving a RON of 100. Further variations involved advancing the ignition degree by 4° CA bTDC and 8° CA bTDC. Notably, a peak power increase of 8% was recorded at a compression ratio of 11.5:1 and an ignition advance of 4° CA bTDC at 4500 rpm (60 kmph). Concurrently, reductions in CO and HC emissions were observed at 52% and 43%, respectively, albeit with a 32% increase in NOx emissions under a compression ratio of 11.5:1 and an advanced ignition degree of 4° CA TDC. A decrease of 8.9% in SFC was noted when comparing compression ratios of 11.5:1 and 9.4:1. The utilization of E30 was found to contribute significantly to fuel savings and performance enhancement, particularly when paired with increased compression ratios and optimal ignition timing settings.

The commercialization of ethanol as a fuel mixture or additive with gasoline has been widely adopted in various countries, notably in the United States and Brazil, where flex-fuel vehicles have been developed to augment the utilization of ethanol-gasoline fuels [21–23]. In similar vein, countries such as Indonesia, Thailand, Japan, China, and India have commenced the integration of alternative fuels, specifically ethanol, as a 10% additive [24]. In Indonesia, a range of SI engine fuels are available, namely Premium, Pertalite, Betamax, and Betamax Turbo, each characterized by octane ratings of 88, 90, 92, and 98 respectively (www.pertamina.com). For the attainment of optimal engine performance, the quality of fuel plays a pivotal role. Fuel serves not only as the combustion source for SI engines but also as a critical component in supporting engine performance, facilitating optimal combustion, and enhancing the performance of SI engines [25].

In light of the aforementioned context, the objective of this research has been delineated as a thorough investigation, primarily through experimental methodologies, to elucidate the impact of ignition timing alterations on the performance parameters of a single-cylinder, four-stroke Otto engine fueled by a 50% ethanol blend (E50). By conducting an exhaustive analysis of the efficiency rendered by this particular fuel blend, the research endeavors to furnish valuable insights and pioneer innovative solutions, ultimately contributing to a substantial reduction in dependence on fossil fuels.

2 Methodology

The methodology employed in this study necessitated the meticulous calibration of testing equipment, with preparations encompassing the computation of standard timing angles at predetermined engine revolutions. A protractor, timing light, and tachometer comprised the instruments utilized for this purpose. A series of tests were conducted to measure torque, power, and fuel consumption, with the subsequent comparative analysis entailing the juxtaposition of experimental results obtained in this study with findings from previous research. This comparison was undertaken with due consideration of extant theories, with the aim of ascertaining congruence with real-world conditions. The data garnered in this study encompassed variables such as torque, power, fuel consumption, and thermal efficiency. These were subject to analysis, facilitated through graphical and tabular representations, to yield insights into the engine's performance. The analytical approach adopted was descriptive in nature, complemented by a comparative study. This entailed a narration of existing data and an exploration of cause-and-effect phenomena, with a focus on the interplay between various factors. The specific relationships under investigation were RPM (Revolutions Per Minute) in relation to torque, RPM in relation to power, and RPM in relation to SFC. The experimental apparatus featured an engine with a capacity of 100 cm^3, employing a carburetor system and utilizing a fuel mixture of Pertalite-ethanol (E50) and Pertalite. Details pertaining to the engine specifications are delineated in Table 1.

Table 1. Engine specifications

Engine Type	4 Langkah, SOHC 2 Valve
Cylinder	Single
Bore × Stroke	50×49.5 mm
Compression	$9 : 1$
Transmission	4-speed $(\mathrm{N} - 1 - 2 - 3 - 4)$
Type of Ignition	CDI-AC

Table 2 displays the ignition timing variations used in this study, which include the standard ignition timing, advance 2° bTDC, advance 4° bTDC, and retard 2° bTDC.

Table 2. Variations of ignition timing

Engine Speed (rpm)	**Ignition Timing Before TDC (°)**			
	Standard	**+2**	**+4**	**−2**
2250	21	22	24	19
2500	25	27	29	23
2750	28	30	32	26
3000	30.5	32.5	34.5	28.5
3250	33	35	37.5	31.5
3500 − 9250	35	37	39	33
9500 − 15000	34	36	38	32

The ignition timing was systematically mapped using the AFR Meter BRT software, which was interfaced with a laptop. The Dynojet 250i dynamometer was employed for the assessment of engine performance, yielding data pertaining to torque, power, and fuel flow rate. The initiation of the testing procedure was predicated upon the meticulous preparation of both the requisite tools and fuel. This preparation entailed a comprehensive examination of the engine oil, an assessment of the pressure in the rear tire of the motorcycle, ensuring the front tire of the motorcycle was securely fastened to the lock, aligning the rear wheel with the roller of the Dynojet 250i, positioning the blower appropriately, and affixing the engine rotation indicator cable to the coil cable. Concurrently, a calibration of the burret measuring instrument and stopwatch was conducted. The preparation of materials culminated with the procurement of Pertalite and ethanol fuels, from which an E50 fuel mixture was subsequently formulated. A schematic of the test arrangement is provided in Figure 1, offering a visual representation of the experimental setup.

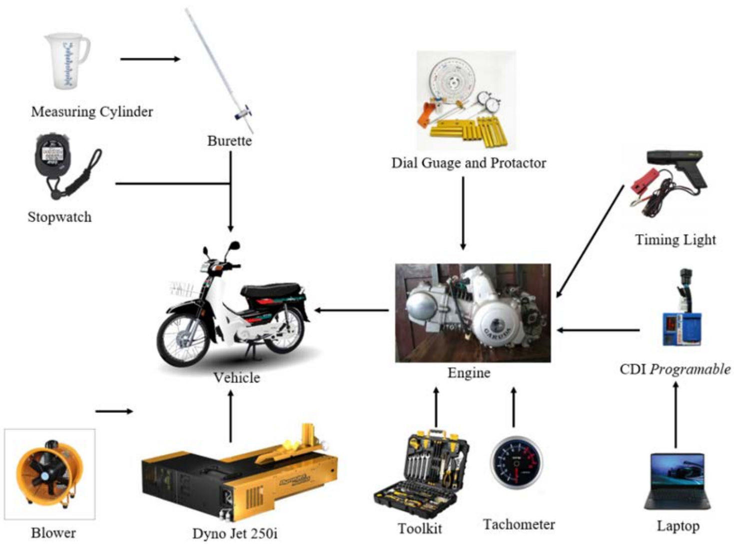

Figure 1. The test scheme

The Dynojet 250i dynamometer was utilized as a crucial instrument in this study, meticulously gathering data on engine performance. This device operates based on the eddy current principle, providing a sophisticated means of load control. The motorcycle's wheel was positioned on the Dynojet 250i roller, which is intricately linked to a rotating disc within a tightly regulated magnetic field. This arrangement facilitates a braking effect on the disc through an advanced eddy current brake system, meticulously calibrated to control the load and accurately measure engine torque. The data obtained, presented in terms of brake horsepower (bhp) for power and foot-pounds (ft. lbs) for torque, offers a precise quantification of the engine's performance characteristics. The Dynojet 250i stands as a testament to the integration of advanced scientific principles, ensuring the acquisition of highly accurate and precise engine testing data.

For the experimental procedure, the motorcycle was securely fastened to the Dynojet 250i dynamometer, with an engine rpm indicator duly installed. The engine was allowed to idle for a duration of five minutes, subsequent to which the engine speed was incrementally increased to 3000 rpm, facilitating the measurement of torque and power. Ignition angle tests were conducted over a range of engine speeds, spanning from 3000 to 9000 rpm, utilizing a programmable CDI. Concurrently, fuel flow rate tests were executed, involving the filling of a burette and timing the consumption of 10 ml of fuel across the aforementioned range of engine speeds. Each test was replicated across standard ignition angles, encompassing -2, +2, and +4 degrees.

Upon completion of the testing and data acquisition phases, performance indices were calculated. The BSFC, a key performance index, was determined by calculating the ratio of fuel consumption to the power generated by the engine.

• The formula of BSFC:

$$\dot{\mathrm{m}}_f = \left(\frac{V}{t}\right) \times \rho$$

$$bsfc = \frac{\dot{\mathrm{m}}_f}{Bp}$$

• The formula of BTE:

$$\eta_{tb} = \frac{Bp}{\dot{\mathrm{m}}_f Q_{lhv} \eta_c}$$

This formula yields valuable insights into the engine's efficiency with regard to fuel utilization. Concurrently, the BTE is derived through a comparison between the power produced by the engine and the theoretical maximum power achievable from complete fuel combustion. These computations provide substantial understanding of the engine's operational efficacy and contribute significantly to the enhancement of fuel efficiency.

3 Results

In the conducted experiment, the third gear ratio was utilized, facilitating full throttle opening across a range of 3000 rpm to 9000 rpm. For the acquisition of fuel flow rate data, the fuel tank hose was replaced with a measuring tube (burette), enabling the determination of fuel consumption per 10 ml with the aid of a stopwatch. This measurement was conducted at engine speeds varying from 4000 rpm to 8000 rpm, with increments of 1000 rpm. The investigation included trials utilizing both gasoline and E50 fuels, with all assessments conducted employing the variable speed engine methodology.

Table 3. Fuel specifications

Num	Properties	Unit	Value of Fuel Properties	
			E50	Pertalite
1.	Density	Gr/ml	0.76565	0.7425
2.	Lower Heating Value	Cal/g	7916.4	11089.8

Table 3 elucidates the variations in calorific value and density between E50 and pertalite fuels. It is of paramount importance that assessments are conducted to ascertain the calorific value of the fuel, given its crucial role in the calculation of the engine's BTE. In parallel, the fuel density must be meticulously evaluated, as it is integral to the determination of the engine's brake-specific fuel consumption. A salient observation that can be drawn from the data is the inverse correlation between the ethanol content in the fuel blend and its calorific value; an increase in ethanol content correlates with a reduction in the fuel's calorific value.

3.1 Torque and Power

The torque and power generated across various ignition timings and fuel types are depicted in Figures 2 and 3. Torque, the rotational force necessary for axle movement, and power, the capacity to perform work rapidly, were both evaluated. A programmable Capacitor Discharge Ignition (CDI) was utilized to alter the ignition timings, demonstrating that an advance in timing could enhance both torque and power, whereas a retardation could diminish these values relative to the standard ignition timing.

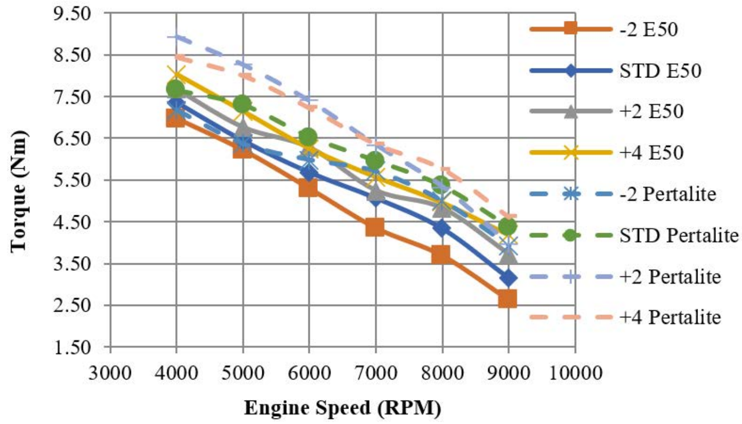

Figure 2. Torque of all ignition timing variations used E50 and gasoline fuel

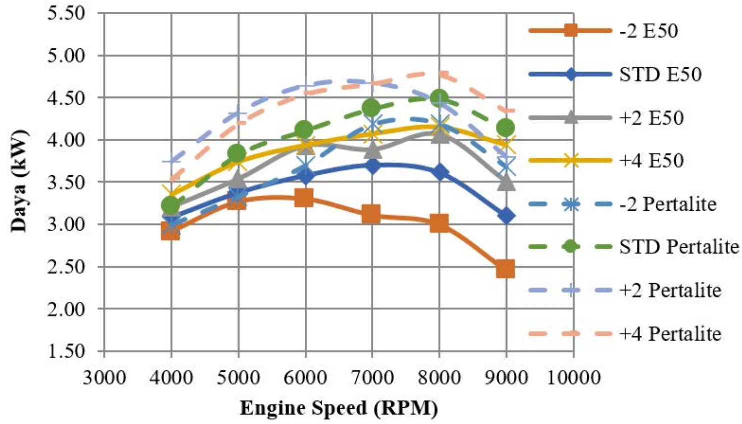

Figure 3. Power of all ignition timing variations used E50 and gasoline fuel

In Otto engines, the ignition position is quantified in degrees of crankshaft rotation, specifically before the Top Dead Center (TDC) on the compression stroke. An overly retarded ignition timing, leading to maximum cylinder gas pressure combustion post TDC, culminates in suboptimal engine performance, elevated exhaust emissions, power loss, and unburned fuel residuals [26–28]. Conversely, excessively advanced ignition timing results in rapid gas pressure combustion, inducing knocking – a condition where gas pressure forces the piston towards TMB during compression, disrupting the combustion process. Optimal combustion necessitates precise ignition timing, ensuring that the spark plug's high-voltage spark concludes the combustion process near TDC [29].

Additionally, the gas valve's opening degree inversely influences torque and power, acting as a barrier to air flow into the intake manifold and resulting in reduced manifold pressure relative to the ambient air pressure. A wider valve opening diminishes this resistance, lowering the manifold's vacuum level. However, this also escalates engine speed, abbreviating fuel combustion duration and preventing complete burning of the fuel-air mixture, thereby diminishing the pressure from the combustion process and, consequently, the engine's torque and power [30].

The calorific value disparity between pertalite and E50 fuels also plays a pivotal role in their respective torque and power outputs. Pertalite possesses a superior calorific value compared to ethanol, and an increase in ethanol content results in a reduced calorific value for the fuel. This, in turn, leads to an inadequate airflow mass in the E50 blend for complete combustion, potentially decreasing the energy released during the expansion phase and, consequently, the torque and power [30].

With a +4° ignition timing, E50 fuel exhibits superior torque and power compared to standard ignition timings of +2° and -2°. Peak values of 8.02 Nm and 4.15 kW are achieved at engine speeds of 3000 rpm and 8000 rpm, respectively. For both E50 and pertalite fuels, optimal performance is observed with pertalite at a +4° ignition timing, yielding a maximum power of 4.77 kW at an 8000-rpm engine speed, and a peak torque of 8.93 Nm at a 3000-rpm engine speed with a +2° ignition timing.

3.2 BSFC and BTE

Presented in Figure 4 is the graphical representation of BSFC results derived from experiments conducted on E50 and pertalite fuels, subjected to predetermined variations in ignition timing. The graph elucidates the quantity of fuel necessitated by the engine to generate a power output of 1 kW over a duration of one hour. For the E50 fuel at a +4° ignition timing variation, the BSFC was observed to be minimal, recording a value of 0.307 kg/kW.h at an engine speed of 4000 rpm. In the case of gasoline, the most optimal BSFC, quantified at 0.22 kg/kW.h, was achieved with a +2° ignition timing variation at the same engine speed. A discernible trend is depicted in the graph, indicating a decrease in BSFC at lower engine speeds, followed by an increment as engine speeds escalate. This phenomenon is attributed to the increased number of engine-driven components at lower speeds, contrasting with a decrease in engine power output at higher speeds, culminating in an elevated BSFC [8]. When a comparative analysis is conducted between ethanol and gasoline, the former is found to exhibit higher BSFC values. This observation is corroborated by ethanol's higher density and lower calorific value [31, 32]. The elevated fuel density culminates in increased BSFC values for ethanol, while its diminished calorific value exerts an augmentative effect on BTE [29, 31]. The maximum BSFC recorded for ethanol stands at 0.727 kg/kW.h, achieved at an engine speed of 9000 rpm.

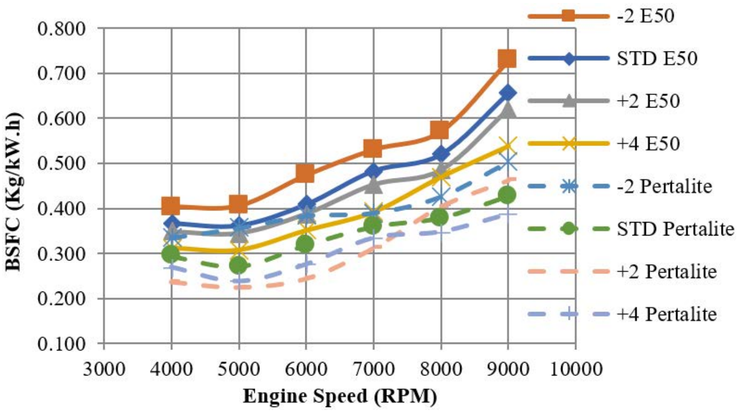

Figure 4. BSFC of all ignition timing variations using E50 and gasoline fuel

In Figure 5, the BTE results for E50 fuel and pertalite are presented, accounting for predetermined variations in ignition angles. E50 fuel, at an ignition angle of +4°, exhibits the pinnacle of thermal efficiency, registering at 36.10% at an engine speed of 4000 rpm. Pertalite fuel, when subjected to an ignition angle of +2°, demonstrates a BTE of 35.38% at the identical engine speed. The superior thermal efficiency of ethanol in comparison to pertalite is attributable to its substantial oxygen content, which stands at 34.7 wt%. This elevated oxygen level plays a crucial role in augmenting thermal efficiency, a phenomenon that has been substantiated by the findings of previous studies, notably the research conducted by Iodice et al. [31], which yielded analogous results.

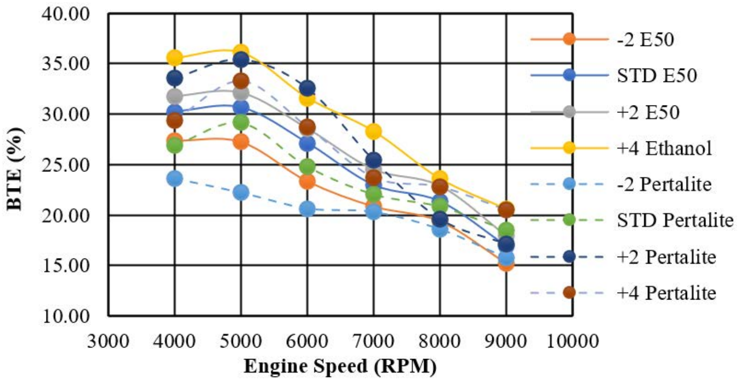

Figure 5. BTE of all ignition timing variations using E50 and gasoline fuel

The investigation elucidates the profound ramifications of manipulating ignition timing through a programmable CDI, manifesting in enhancements to engine torque, power, and thermal efficiency, as well as a concomitant abatement in SFC. In the context of E50 fuel, the torque reaches its zenith at 4000 rpm, exhibiting fluctuations across various ignition settings: 7.36 Nm at the standard ignition angle, 7.89 Nm at +2°, 8.02 Nm at +4°, and 6.96 Nm at -2°. Concurrently, the apogees of power output are discerned at 7000 rpm with standard timing, 8000 rpm at both +2° and +4° angles, and 6000 rpm at -2° angle, culminating in 3.70 kW, 4.06 kW, 4.15 kW, and 3.29 kW, respectively. A nadir in fuel consumption is consistently observed at 5000 rpm across the spectrum of ignition angles, with values oscillating between 0.36 Kg/kW.h and 0.40 Kg/kW.h. Furthermore, thermal efficiency attains its maximum at 36.10% under a +4° ignition angle at 5000 rpm. This elucidates the intricate interplay between variations in ignition timing and the resultant engine performance metrics, specifically in the context of E50 fuel.

4 Conclusions

The evaluations of engine performance conducted in this study elucidate the effects of fuel type and ignition timing on a 100cc Honda Grand motorcycle. It has been observed that the employment of E25 fuel with standard ignition timing precipitates a discernible diminution in engine performance metrics, encompassing torque, power output, and BTE. This decrement in engine performance is concomitant with an augmentation in SFC, in comparison to the application of pertalite fuel. Conversely, the utilization of E50 fuel in conjunction with a +4 ignition timing has been found to substantially enhance engine performance, as evinced by the recorded values of 8.02 Nm for torque, 8.93 kW for power output, and 36% for brake thermal efficiency. This improvement is attributed to the inherently slower combustion kinetics of E50 fuel, necessitating an advanced ignition timing to attain optimal combustion pressures, observed to occur 5°-10° subsequent to TDC. Furthermore, E50 fuel has been demonstrated to offset approximately 86.73% and 85.54% of the mean power and mean torque values obtained with pertalite fuel, respectively.

In terms of BSFC and BTE, increments of 25.05% and 5.02% have been recorded, respectively. These fluctuations are ascribed to the intrinsic properties of ethanol, notably its lower calorific value and higher density in comparison to pertalite. The elevation in BSFC and BTE values is posited to exert a significant influence on the overall energy utilization efficiency in vehicular applications, fostering a reduction in dependence on fossil fuels and ameliorating environmental impacts. In light of these findings, it is imperative to underscore the criticality of a meticulous understanding and management of BSFC and BTE for the optimization of vehicle efficiency and the mitigation of environmental and operational costs. The insights garnered from this investigation underscore the potential of ethanol-blended fuel formulations in the amelioration of fossil fuel dependency and advocate for extended research, particularly within the realm of larger engine capacities, to further refine the optimization of fuel blends.

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Data Availability

The data used to support the findings of this study are available from the corresponding author upon request.

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Conflicts of Interest

The authors declare that they have no conflicts of interest.

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Nomenclature

Bp	Power (kW)
V	Fuel volume $\left(\mathrm{m}^3\right)$
t	Time (s)
$bsfc$	Specific fuel consumption $(\mathrm{kg/kW.h})$
ρ	Fuel density $\left(\mathrm{kg/m}^3\right)$
$\dot{\mathrm{m}}_f$	Mass fuel flow rate $(\mathrm{kg/h})$
η_{tb}	Thermal efficiency $(\%)$
η_c	Combustion Efficiency $(\%)$
Q_{lhv}	Caloric value $(\mathrm{kJ/kg})$