

SPARK-TIMING CALIBRATION OF A GASOLINE-AMMONIA MOTORCYCLE ENGINE USING CFD AND EXPERIMENTS

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(Received: July 24, 2026; Revised: September 07, 2026; Accepted: September 11, 2026)

DOI: 10.31130/ud-jst.2026.24(9).621E

Abstract - This study investigates ignition-map requirements for a Honda Lead 110cc motorcycle engine fuelled with gasoline, a 30% gasoline 70% NH₃, and pure NH₃ at 7500 rpm, wide-open throttle, and $\phi=1$. Three-dimensional CFD simulations using the Partially Premixed/Zimont-Lipatnikov model were combined with original ignition-map measurements using SmartFi 2 and an oscilloscope. The simulated gasoline MBT timing, 33°CA BTDC, agreed with the measured value of 33.1° CA BTDC. Increasing ammonia content shifted MBT to 42°CA BTDC for the blend and 55°CA BTDC for pure ammonia. At MBT, power decreased from 6.40 kW for gasoline to 5.56 and 4.63 kW, respectively. Predicted NO_x decreased from 5500 ppm to 1900 and 1700 ppm. Simulation-experiment NO_x deviations averaged 11.8%. The results provide a practical basis for remapping ignition in gasoline-ammonia motorcycle engines.

Key words - Ammonia-gasoline dual fuel; Motorcycle spark-ignition engine; Spark advance; Maximum brake torque; CFD simulation

1. Introduction

The decarbonization of road transport is essential to achieving the climate objectives of the Paris Agreement and Vietnam's commitment to net-zero emissions by 2050. Although vehicle electrification is the long-term pathway, its large-scale deployment requires substantial investment in charging infrastructure, battery supply chains, and electricity generation. This challenge is particularly relevant to Vietnam, where motorcycles remain the dominant mode of personal transport. Therefore, low-carbon fuels that can be introduced into existing internal-combustion-engine platforms may provide a practical transitional option.

Ammonia (NH₃) has attracted increasing attention as a carbon-free energy carrier and fuel because it contains no carbon, can be synthesized from low-carbon hydrogen, and has an established global infrastructure for production, storage, and transport [1-4]. Unlike hydrogen, ammonia can be liquefied under relatively moderate pressure at ambient temperature, resulting in a high volumetric hydrogen density and facilitating onboard storage [1-2]. These advantages have stimulated research on ammonia utilization in gas turbines, marine engines, stationary engines, and, more recently, internal combustion engines [4-7]. Vietnam also has favorable conditions for producing renewable electricity and e-fuels from solar, wind, and biomass resources [8-10]. Consequently, ammonia could potentially complement direct electrification in

applications where the continued use of liquid- or gaseous-fuelled engines is necessary during the energy transition.

However, direct use of ammonia in spark-ignition engines is constrained by unfavorable combustion characteristics. Compared with conventional hydrocarbon fuels, ammonia has a low laminar burning velocity, a high auto-ignition temperature, a relatively high minimum ignition energy, and a narrow flammability range [3-5], [11-13]. Its laminar burning velocity is commonly reported to be only a fraction of that of gasoline under comparable conditions [3], [5], [11-13]. These properties make flame-kernel development and subsequent flame propagation more difficult, particularly in small high-speed engines, where the available crank-angle duration for combustion is short. Moreover, ammonia-compatible fuel systems require suitable materials because liquid ammonia can adversely affect copper-, zinc-, and certain elastomer-containing components [3].

A practical method for overcoming the slow combustion of ammonia is dual-fuel operation with gasoline. Gasoline supplies highly reactive hydrocarbon species and promotes early flame development, whereas ammonia partially replaces the carbon-containing energy input. Recent studies have demonstrated the feasibility of ammonia-gasoline combustion in spark-ignition engines under various operating conditions [4]. Nevertheless, the appropriate ignition timing remains a key unresolved calibration variable. If ignition is too late, the slow ammonia-containing flame releases heat predominantly after top dead center (TDC), thereby reducing expansion work and engine power. Conversely, excessive spark advance increases negative work during compression and may increase peak pressure, temperature, and nitrogen-oxide emissions. The optimum spark advance, conventionally associated with maximum brake torque (MBT), must therefore be identified for each fuel composition and operating condition.

The sensitivity of MBT timing to ammonia content is expected to be particularly strong in motorcycle engines. These engines have small combustion chambers, high rotational speeds, and short residence times. Turbulent combustion theory indicates that the flame-propagation process depends on the interaction between turbulence intensity and laminar burning velocity, commonly represented by the ratio u'/S_L and related nondimensional parameters such as the Damköhler number [14-16]. As

engine speed increases, turbulence intensity generally rises while the time available for combustion decreases. For fuels with low S_L , such as ammonia, this interaction can require a substantially earlier spark than for gasoline. Therefore, the ignition map cannot be transferred directly from the original gasoline calibration to ammonia-containing fuels without modification.

Previous ammonia-engine studies have mainly addressed pure ammonia, ammonia-hydrogen mixtures, or large-displacement automotive and stationary engines [7], [11-12], [17]. CFD simulation is especially valuable for screening spark timings and fuel compositions because it enables controlled comparisons over a wide operating range and can be extended to conditions that are difficult to investigate experimentally, including pure-ammonia operation [4-5]. However, studies that combine in-cylinder CFD analysis, direct measurement of the original ignition timing, and practical recalibration of the original motorcycle ECU remain limited. In particular, quantitative information on the MBT shift from gasoline to gasoline-ammonia blends and pure ammonia in a high-speed motorcycle engine is scarce.

This study addresses this gap by investigating the optimum spark advance of a Honda Lead 110 cc motorcycle engine fuelled with gasoline, a 30% gasoline-70% ammonia blend, and pure ammonia. First, three-dimensional CFD simulations are performed to determine the effects of spark advance on in-cylinder pressure, power output, and NO_x emissions at 7500 rpm and wide-open throttle. The resulting MBT angles and power penalties are compared among the three fuels. Second, the original gasoline ignition characteristic is measured independently using a SmartFi 2 diagnostic device and an oscilloscope synchronized with the crankshaft-position signal. This dual measurement establishes the reference ignition timing used to assess the consistency of the gasoline simulation. Finally, the simulated MBT shifts are translated into remapped ignition characteristics for gasoline-ammonia operation, and simulated NO_x emissions are compared with measurements for gasoline and the 30% gasoline-70% ammonia blend. The results provide a quantitative basis for ignition-map calibration when retrofitting high-speed motorcycle engines for ammonia-gasoline dual-fuel operation.

2. Materials and Methods

2.1. Test Engine and Operating Conditions

The study was conducted on a single-cylinder Honda Lead 110 cc four-stroke engine equipped with the original PGM-FI electronic fuel-injection system. The engine has a cylinder bore of 50mm, a piston stroke of 55mm, and a compression ratio of 11:1. When fuelled with gasoline, its rated power is 6.4 kW at 7500 rpm.

The numerical and experimental analyses focused on the rated-speed condition, $n = 7500$ rpm, at wide-open throttle (WOT) and an overall equivalence ratio of $\phi=1$. Three fuel cases were considered in the CFD analysis: gasoline, 30% gasoline-70% NH_3 , and pure NH_3 . The gasoline case provides the baseline corresponding to the original engine calibration. The 30% gasoline-70% NH_3

case represents the highest ammonia proportion that was experimentally investigated at the rated speed, whereas pure NH_3 was examined numerically because stable operation with this fuel could not yet be tested on the available experimental platform.

2.2. CFD Simulation Method

The in-cylinder processes were simulated using Ansys Fluent 2021 R1. The Partially Premixed Combustion model was coupled with the Zimont-Lipatnikov turbulent flame-speed model and the standard $k-\epsilon$ turbulence model. This modelling framework was selected because the premixed fuel-air charge is ignited by a spark and subsequently propagates as a turbulent flame through the combustion chamber.

Each calculation covered one complete engine cycle of 720°CA, from the beginning of the intake process to the end of the expansion process. The simulation therefore accounts for the evolution of the in-cylinder flow, turbulence, compression, ignition, combustion, and expansion processes. For each fuel, several spark-advance angles were evaluated. The optimum spark advance was identified as the timing giving the maximum effective power, i.e., the maximum-brake-torque condition.

The CFD results were assessed against the reference gasoline condition in two ways. First, the calculated gasoline MBT angle at 7500rpm was compared with the original ignition advance independently measured on the engine. Second, the predicted NO_x concentrations for gasoline and the 30% gasoline-70% NH_3 blend were compared with exhaust measurements. After this assessment, the model was used to estimate the combustion characteristics, MBT timing, power output, and NO_x emissions for pure NH_3 , for which direct experimental operation was not available.

2.3. Dual-Fuel Experimental System and Test Procedure

To minimize modification of the original motorcycle, an auxiliary NH_3 supply system was installed in parallel with the original gasoline PGM-FI system rather than replacing the factory fuel-delivery architecture. As illustrated in Figure 1(a), gasoline is supplied through the original fuel tank, pump, injector, and ECU. The added branch comprises the NH_3 storage tank, pressure-reducing valve, pressure gauge, heater, flow meter, and an auxiliary NH_3 injector. This arrangement permits independent control of gasoline and ammonia delivery while retaining the original gasoline system as a fallback mode.

Figure 1(b) shows the assembled experimental hardware, including the fuel-storage cylinders, control unit, and motorcycle. The NH_3 fuel was stored as a liquid in an 8-L SS316L stainless-steel tank with a capacity of approximately 4kg. Before injection, its pressure was reduced to 4-5bar and the fuel was vaporized. The gaseous NH_3 was then injected upstream of the throttle valve into the intake system. All components exposed to ammonia, including pipes, valves, and seals, were made of or selected from ammonia-compatible materials, namely SS316 stainless steel and PTFE/EPDM, to limit corrosion and material degradation.

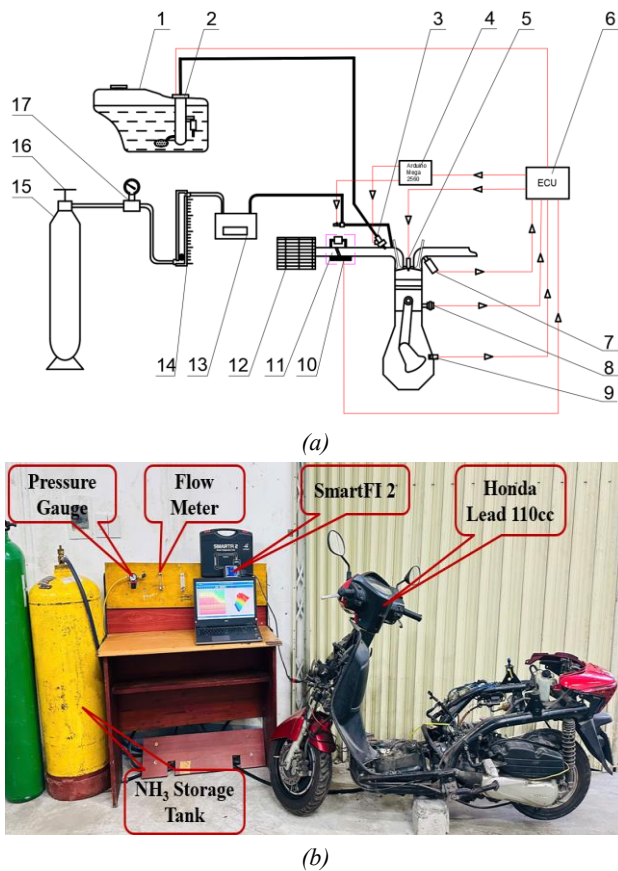


Figure 1. Schematic of the experimental setup (a) and photograph of the experimental apparatus and materials (b)

1. Gasoline tank; 2. Fuel pump; 3. Gasoline injector; 4. Control circuit; 5. Spark plug; 6. ECU; 7. Oxygen sensor; 8. Lubricant temperature sensor; 9. Crankshaft position sensor; 10. MAP, IAT, and TP sensor cluster; 11. Throttle valve; 12. Air filter; 13. Heater; 14. Flow meter; 15. NH_3 storage tank; 16. Pressure-reducing valve; 17. Pressure gauge.

An Arduino Mega 2560-based auxiliary controller was used to operate the added fuel branch. Signals related to manifold pressure, throttle position, engine speed, and oxygen-sensor response were monitored to determine the operating condition. The controller generated a PWM command through a MOSFET driver to actuate the NH_3 injector and coordinate its operation with the gasoline injector. The engine was started on gasoline; ammonia injection was introduced only after stable operation was achieved. In the event of a sensor or control-system fault, the auxiliary system was configured to return the engine to gasoline-only operation.

Ammonia consumption was measured by a flow meter, whereas gasoline consumption was determined gravimetrically. The blend ratio was established from the measured fuel flow rates. Exhaust-gas measurements were conducted using a Sauermann gas analyser and sampling probe. The tested cases included gasoline, and 30% gasoline-70% NH_3 . Each condition was repeated three times at WOT and 7500 rpm. Because ammonia is toxic and irritates the eyes, skin, and respiratory tract, all tests were performed in a ventilated area with appropriate personal protective equipment, as represented by the practical setup in Figure 1(b).

2.4. Measurement of Original Spark Advance

Accurate identification of the original gasoline ignition timing is essential because it serves as the reference for evaluating the gasoline CFD result and for constructing the ignition correction for ammonia-containing fuels. Therefore, the original spark advance was determined using two independent methods: direct reading of the ECU calibration through SmartFi 2 and time-resolved measurement using an oscilloscope.

2.4.1. SmartFi 2 Measurement

The SmartFi 2 diagnostic device was connected to the original ECU through the OBD-II data-link connector, as shown in Figure 2. This configuration enabled the original ignition map to be read without removing or disassembling the ECU. The measured map covered the engine-speed range from 1600 to 7500 rpm and was used as the gasoline reference characteristic for subsequent remapping.

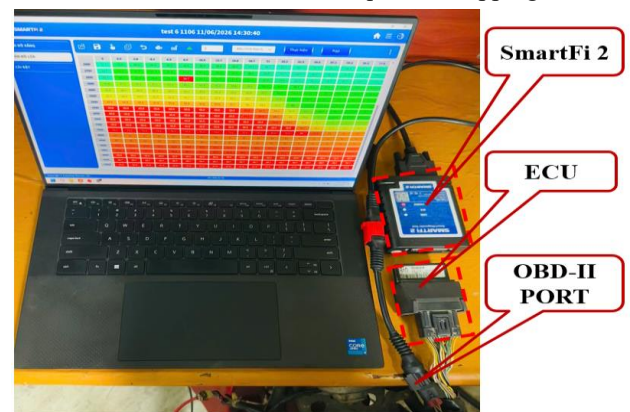


Figure 2. Measurement of spark advance using the Smart Fi 2 diagnostic device connected through the OBD-II port.

2.4.2. Oscilloscope Cross-Validation

An independent oscilloscope measurement was performed to verify the ignition timing obtained from SmartFi 2. Figure 3 presents the Hantek DSO5072P digital storage oscilloscope and the Tektronix P6015A high-voltage probe used to acquire the ignition signal. The oscilloscope has two channels, a bandwidth of 70 MHz, and a sampling rate of 1GSa/s. The P6015A probe is a passive 1000:1 high-voltage probe, rated to 20 kV DC and 40 kV peak, allowing measurement at the spark-plug terminal.

The secondary ignition-voltage signal and the crankshaft-position (CKP) sensor signal were recorded simultaneously. The flywheel contains nine equally spaced trigger teeth; tooth No. 9 was used as the TDC reference. The time difference between the ignition event and the TDC-reference pulse, $\Delta t_{\text{ignition}}$, was converted into crank angle. Measurements were performed at 1700 rpm, 5000 rpm, and 7500 rpm, representing idle, intermediate-speed, and rated-speed conditions, respectively.

The duration of one crankshaft revolution is:

$$T = \frac{60}{n} \times 10^3 [\text{ms}] \quad (1)$$

where n is the engine speed in rpm. The original spark advance, ϕ_s , expressed in degree crank angle before TDC, was then calculated as:

$$\varphi_o = \Delta t_{\text{ignition}} \times n \times \frac{360}{60 \times 10^3} [^\circ \text{CABTDC}] \quad (2)$$

The dual measurement approach reduces reliance on a single instrument: SmartFi 2 provides the stored ECU map, whereas the oscilloscope verifies the actual timing event relative to the crankshaft reference under engine operating conditions.



(a)



(b)

Figure 3. Hantek DSO5072P digital oscilloscope (a) and Tektronix P6015A high-voltage probe used for ignition-pulse measurements (b)

3. Results and Discussion

3.1. Simulation of the Optimal Spark Advance

3.1.1. Gasoline Operation

Figure 4 presents the calculated temperature field at four representative crank angles for gasoline operation. Following ignition at 33°CA BTDC, a high-temperature flame kernel is established near the spark-plug region at 327°CA. The flame then expands rapidly through the central part of the chamber at 340°CA and reaches the peripheral regions by 355°CA. At 361°CA, the high-temperature burned-gas region occupies nearly the whole chamber, indicating rapid and relatively symmetric flame development. This combustion behavior is consistent with the high reactivity of gasoline and provides the reference condition for evaluating ammonia-containing fuels.

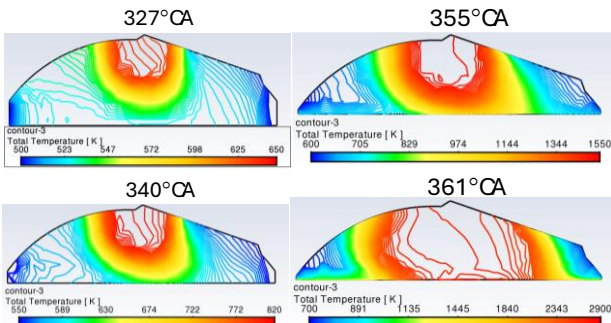
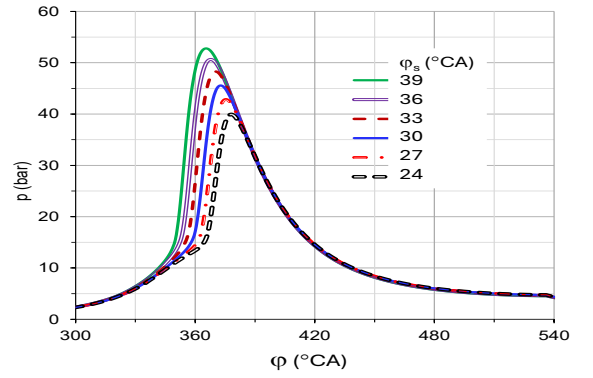
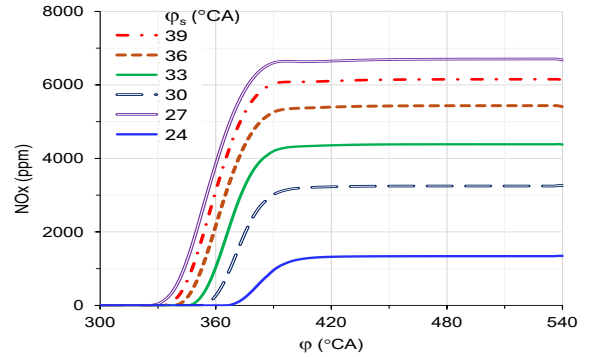


Figure 4. In-cylinder temperature contours illustrating flame development for gasoline operation at $n=7500 \text{ rpm}$ and $\phi=1$



(a)



(b)

Figure 5. Effects of spark advance on in-cylinder pressure traces (a) and predicted NO_x emissions (b) during gasoline operation $n = 7500 \text{ rpm}$, $\phi=1$

As shown in Figure 5(a), advancing the spark from 24°CA to 39°CA BTDC increases the peak in-cylinder pressure and shifts its occurrence closer to TDC. Earlier ignition allows a larger fraction of the chemical energy to be released during the early expansion process. However, excessively early combustion also increases the pressure before TDC and hence the negative compression work. Therefore, the maximum pressure does not directly correspond to the maximum effective power.

The predicted power at this point is identical to the rated gasoline power of the engine. Figure 5(b) further shows that predicted NO_x emissions rise strongly with spark advance, from 1200 ppm at 24°CA BTDC to 6870 ppm at 39°CA BTDC. This trend follows the increase in peak combustion temperature and pressure, as well as the longer residence time of burned gases under high-temperature conditions. At the gasoline MBT point, the predicted NO_x concentration is 5500 ppm.

Table 1. Predicted engine power and NO_x emissions at different spark-advance angles during gasoline operation ($n = 7500 \text{ rpm}$, $\phi=1$)

φ_s (°CA BTDC)	24	27	30	33	36	39
Pe (kW)	5.95	6.11	6.24	6.40	6.28	6.18
NO_x (ppm)	1200	3100	4200	5500	6050	6870

This trade-off is quantified in Table 1. The effective power increases from 5.95 kW at 24°CA BTDC to a maximum of 6.4 kW at 33°CA BTDC, then decreases to 6.18 kW at 39°CA BTDC. Accordingly, the gasoline MBT timing is identified as $\varphi_s = 33^\circ \text{CABTDC}$.

3.1.2. 30% Gasoline-70% NH₃ Operation

The pressure histories in Figure 6(a) are intermediate between those obtained with gasoline and pure NH₃. Relative to gasoline, the pressure peak is lower and occurs later at comparable spark timings, indicating slower heat release caused by the high ammonia fraction. Therefore, the blend requires a larger spark advance to position the main heat-release phase within the useful part of the expansion stroke.

Figure 6(b) shows that NO_x remains very low at spark advances below 33°CA BTDC, then rises rapidly as the spark is advanced beyond 37°CA BTDC. The predicted NO_x concentration at the blend MBT point is 1900 ppm, approximately 65% lower than that for gasoline at its MBT timing. This reduction is consistent with the lower pressure and, consequently, lower peak-temperature level shown in Figure 6(a).

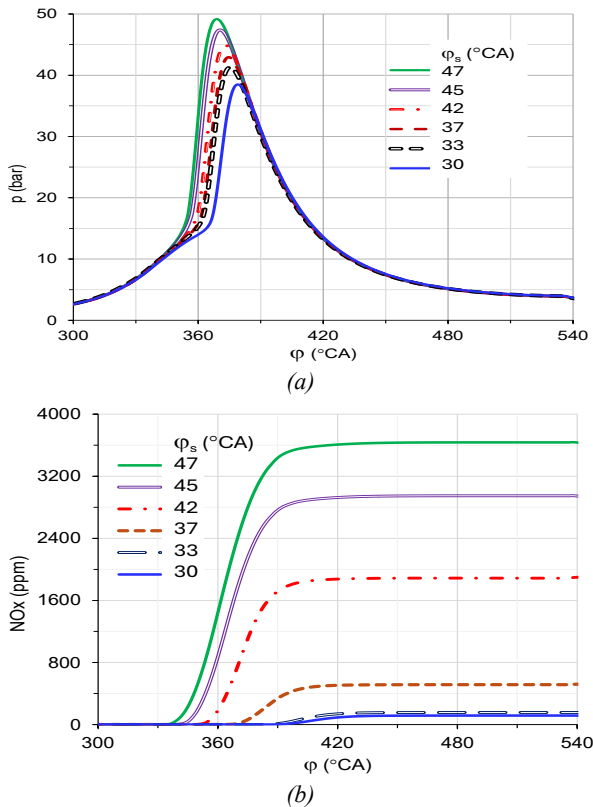


Figure 6. Effects of spark advance on in-cylinder pressure traces (a) and predicted NO_x emissions (b) for the 30% gasoline-70% NH₃ blend at $n = 7500$ rpm, $\phi = 1$, and WOT

Table 2. Predicted engine power and NO_x emissions at different spark-advance angles for the 30% gasoline-70% NH₃ blend ($n = 7500$ rpm, $\phi = 1$)

ϕ_s (°CA BTDC)	30	33	37	42	45	47
P_e (kW)	4.75	5.11	5.47	5.56	5.45	5.32
NO _x (ppm)	100	110	450	1900	2900	3650

As indicated in Table 2, the power rises from 4.75 kW at 30°CA BTDC to 5.56 kW at 42°CA BTDC, then declines as ignition is further advanced. The MBT timing of the 30% gasoline-70% NH₃ blend is consequently $\phi_s = 42^\circ$ CA BTDC.

This value is 9°CA earlier than that of gasoline. At MBT,

the engine produces 5.56 kW, corresponding to a 13% power reduction relative to gasoline. Nevertheless, the gasoline fraction substantially improves combustion compared with pure ammonia by supporting initial flame development and increasing the overall reactivity of the charge.

3.1.3. Pure NH₃ Operation

Pure ammonia exhibits the slowest combustion characteristics. As shown in Figure 7(a), the pressure rise is substantially lower and broader than in the gasoline and blended-fuel cases. At equal spark advance, the pressure peak is shifted further into the expansion stroke, indicating that a considerable part of the heat release occurs too late to be converted effectively into useful work. Even at $\phi_s = 33^\circ$ CA BTDC, the peak pressure is only approximately 38 bar.

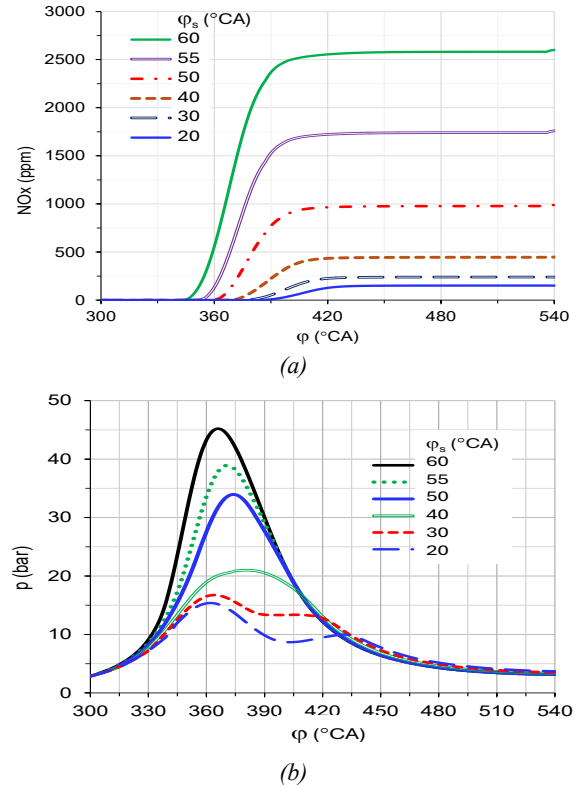


Figure 7. Effects of spark advance on in-cylinder pressure traces (a) and predicted NO_x emissions (b) during pure NH₃ operation $n = 7500$ rpm, $\phi = 1$, and WOT

As illustrated in Figure 7(b), predicted NO_x increases progressively with spark advance, reaching 2550 ppm at 60°CA BTDC. At MBT, the predicted NO_x concentration is 1700 ppm. Although ammonia introduces fuel-bound nitrogen, the present simulations predict lower total NO_x than gasoline under the corresponding MBT condition, owing to the substantially lower pressure and inferred peak combustion temperature. This prediction should be interpreted cautiously because pure-ammonia operation was not experimentally validated in the present study.

Table 3. Predicted engine power and NO_x emissions at different spark-advance angles during pure NH₃ operation ($n = 7500$ rpm, $\phi = 1$)

ϕ_s (°CA BTDC)	20	30	40	50	55	60
P_e (kW)	2.87	3.50	4.10	4.56	4.63	4.52
NO _x (ppm)	200	250	490	1000	1700	2550

Table 3 confirms that a large spark advance is required to compensate for the low reactivity of ammonia. The effective power increases from 2.87 kW at 20°C BTDC to 4.63 kW at 55°C BTDC, before decreasing to 4.52 kW at 60°C BTDC. The pure-ammonia MBT timing is therefore $\phi_s = 55^\circ \text{CA BTDC}$.

This timing is 22°C earlier than the gasoline MBT timing. Despite this large advance, the maximum predicted power remains 27.7% below the gasoline value. The result highlights the intrinsic limitation of neat ammonia in a high-speed spark-ignition engine without a combustion promoter or an advanced ignition-assistance strategy, as also identified in previous ammonia-engine studies [5], [6].

3.1.4. Comparison of MBT Characteristics

Table 4 summarizes the effect of ammonia substitution on the simulated MBT characteristics. Increasing the ammonia fraction shifts MBT systematically toward earlier crank angles and reduces the attainable power. The 30% gasoline-70% NH_3 blend provides a compromise between gasoline and pure ammonia: its MBT timing is 33°C BTDC and its power penalty is limited to 13.0%, whereas pure ammonia requires 55°C BTDC and causes a 27.7% power penalty.

Table 4. Comparison of predicted MBT timing, power output, power reduction, and NO_x emissions for the three fuels at $n = 7500 \text{ rpm}$, $\phi = 1$

Fuel	ϕ_s (° CA)	P_e (kW)	ΔP_e (%)	NO_x (ppm)
Gasoline	33	6.40	0.0	5500
30% Gasoline - 70% NH_3	42	5.56	13.0	1900
Pure NH_3	55	4.63	27.7	1700

For the three investigated fuel compositions, the MBT shift from gasoline to pure ammonia is:

$$\Delta\phi_{s,MBT} = 55 - 33 = 22^\circ \text{CA}$$

Over the range from 0% to 100% ammonia, this corresponds to an average MBT increase of approximately:

$$\Delta\phi_{s,MBT} \approx 2.2^\circ \text{CA per 10% ammonia}$$

This value should be regarded as a first-order calibration guide for the present engine at 7500 rpm, WOT, and $\phi = 1$, rather than as a universal linear correlation. The results demonstrate that ignition-map correction is indispensable when ammonia is introduced into a gasoline motorcycle engine. They also show that a gasoline pilot fraction can significantly reduce the ignition-advance requirement and power penalty associated with pure-ammonia operation.

3.2. Experimental Measurement of the Original Spark Advance

3.2.1. SmartFi 2 Measurement

Figure 8 shows that the original gasoline spark advance increases nonlinearly with engine speed. It remains nearly constant at approximately 13.5-14.5°C BTDC in the low-speed range of 1600-2400 rpm, increases rapidly in the intermediate-speed range, and then approaches a plateau of approximately 33°C BTDC above 5500 rpm. At the rated

speed of 7500 rpm, the original ECU map provides a spark advance of approximately 33°C BTDC. This value agrees with the simulated gasoline MBT timing identified in Section 3.1 and was therefore adopted as the reference condition for ignition-map correction with ammonia-containing fuels.

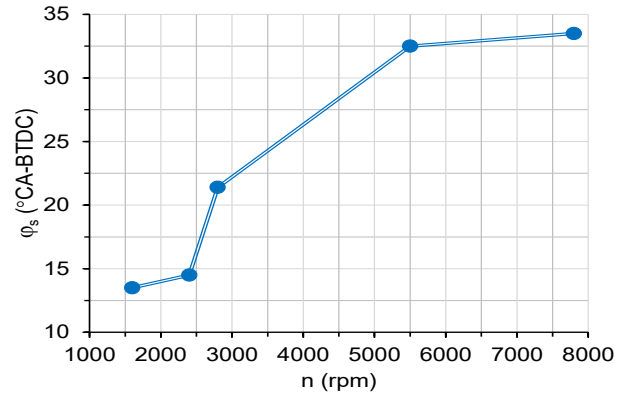


Figure 8. Original gasoline spark-advance characteristic as a function of engine speed, measured using the SmartFi 2 diagnostic device

3.2.2. Oscilloscope Cross-Validation

The SmartFi 2 results were cross-checked by simultaneously recording the CKP signal and secondary ignition-voltage signal with the oscilloscope. As shown in Figure 9, the time interval $\Delta t_{\text{ignition}}$ between the ignition event and the tooth-No.-9 TDC reference was measured directly and converted to crank angle using Equation (2).

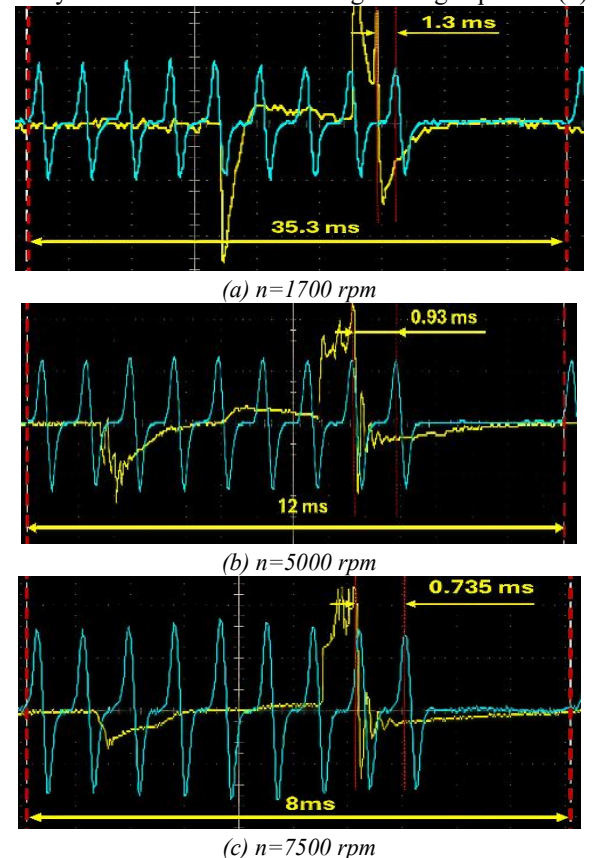


Figure 9. Oscilloscope traces of the CKP and secondary ignition-voltage signals at (a) 1700 rpm, (b) 5000 rpm, and (c) 7500 rpm

At 1700 rpm, $\Delta t_{\text{ignition}} = 1.30$ ms gives $\phi_s = 13.5^\circ\text{CA}$ BTDC. This result agrees with the $13.5\text{--}14.5^\circ\text{CA}$ BTDC range obtained from SmartFi 2. At 5000 rpm, the measured interval of 0.93 ms corresponds to $\phi_s = 28.0^\circ\text{CA}$ BTDC. At the rated speed of 7500 rpm, $\Delta t_{\text{ignition}} = 0.735$ ms yields $\phi_s = 33.1^\circ\text{CA}$ BTDC.

The measured value at 7500 rpm differs by less than 1°CA from both the SmartFi 2 result and the simulated gasoline MBT value of 33°CA BTDC. The agreement at the rated operating condition supports the use of the CFD-derived MBT shift to construct ignition maps for gasoline-ammonia blends. In particular, the progressively earlier MBT timings predicted for the 30% gasoline-70% NH_3 blend and pure NH_3 can be referenced to the experimentally confirmed original gasoline map.

3.3. Remapped Ignition Map for Gasoline-Ammonia Operation

The original gasoline calibration cannot be applied directly after ammonia is introduced because the lower reactivity of the gasoline- NH_3 charge delays the main heat-release process. Therefore, the original ECU map was adjusted using the SmartFi 2 device via Bluetooth. The remapping procedure used the original gasoline spark-advance characteristic as the reference and applied a fuel-dependent advance correction derived from the simulated MBT timings. The fuel-injection settings were adjusted separately to achieve the prescribed fuel-blend ratio; the present analysis focuses on the ignition-map modification.

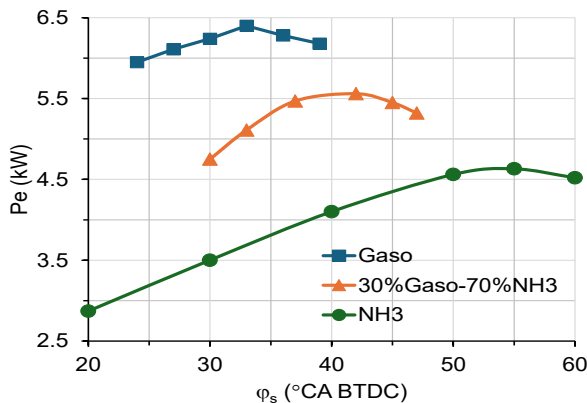


Figure 10. Engine power as a function of spark advance for gasoline, the 30% gasoline-70% NH_3 blend, and pure NH_3 at $n = 7500$ rpm, $\phi = 1$, and WOT

As shown in Figure 10, the maximum-power points shift systematically toward earlier ignition as the ammonia fraction increases. For gasoline, the simulated optimum timing is 33°CA BTDC, which agrees with the 33.1°CA BTDC value measured independently at 7500 rpm using the oscilloscope. This agreement provides the reference point for correcting the original ignition map. The MBT timing increases to 42°CA BTDC for the 30% gasoline-70% NH_3 blend and to 55°CA BTDC for pure NH_3 . Accordingly, the required additional advance at the rated operating condition is:

$$\Delta\phi_{s,30\text{Gasoline-}70\text{NH}_3} = 42 - 33 = 9^\circ\text{CA}$$

$$\Delta\phi_{s,\text{NH}_3} = 55 - 33 = 22^\circ\text{CA}$$

These corrections compensate for the slower combustion of the ammonia-containing mixtures and move the principal heat-release period closer to the optimum crank-angle interval for expansion work.

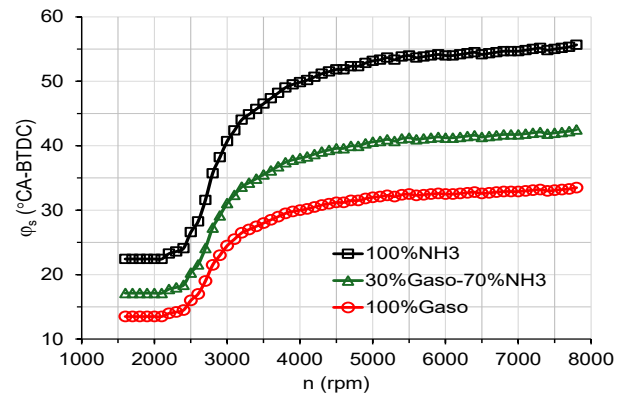


Figure 11. Remapped spark-advance characteristics for gasoline, the 30% gasoline-70% NH_3 blend, and pure NH_3 over the investigated engine-speed range

Figure 11 presents the resulting ignition characteristics over the engine-speed range. The remapped curves retain the nonlinear shape of the original gasoline map: the advance is relatively low at low speed, increases rapidly in the transition region, and gradually approaches a plateau at high speed. However, the entire characteristic shifts upward as the ammonia fraction increases. At 7500 rpm, the remapped curves reach approximately 33°CA BTDC for gasoline, 42°CA BTDC for the 30% gasoline-70% NH_3 blend, and 55°CA BTDC for pure NH_3 , consistent with the MBT results in Figure 10.

The remapped ignition characteristics were subsequently uploaded to the original ECU for engine tests. This procedure provides a practical route for adapting an existing gasoline motorcycle engine to gasoline-ammonia dual-fuel operation without replacing the original ECU or its basic control architecture.

3.4. Comparison Between Simulated and Measured NO_x Emissions

Emission tests were conducted at the rated condition of 7500 rpm and WOT for gasoline and the 30% gasoline-70% NH_3 blend. These two fuels were selected because the engine could operate stably at the rated speed. The measurements were used to evaluate the predictive capability of the CFD model before applying it to pure-ammonia operation, which could not be tested with the present facility.

As shown in Figure 12, NO_x emissions increase monotonically as spark advance increases for all fuel cases. Earlier ignition advances combustion toward TDC, raising the peak pressure and temperature and extending the residence time of burned gases at elevated temperature. These conditions promote NO formation. The effect is strongest for gasoline, for which NO_x rises from approximately 1200 ppm at 24°CA BTDC to approximately 6870 ppm at 39°CA BTDC

The corresponding NO_x concentrations at MBT are 5500 ppm for gasoline, 1900 ppm for the 30% gasoline-

70% NH₃ blend, and 1700ppm for pure NH₃. Although ammonia contains fuel-bound nitrogen, the lower predicted NO_x values for the ammonia-containing fuels are associated with their slower combustion and lower pressure and temperature levels relative to gasoline. The proximity of the predicted MBT NO_x values for the blend and pure ammonia should not be interpreted as evidence that their NO_x-formation routes are identical; rather, it reflects the combined effect of fuel nitrogen chemistry and the lower thermal-NO formation tendency under the simulated conditions.

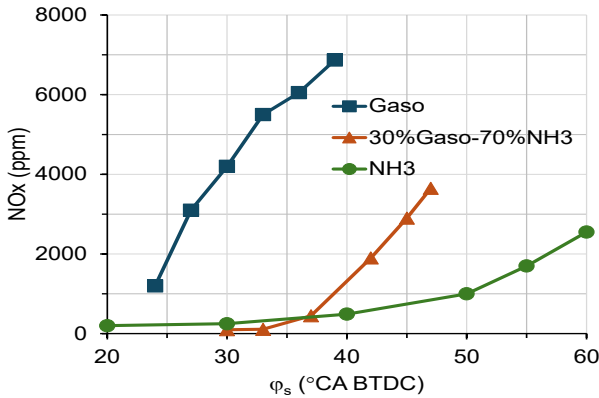


Figure 12. Predicted NO_x emissions as a function of spark advance for gasoline, the 30% gasoline-70% NH₃ blend, and pure NH₃ $n = 7500$ rpm, $\phi = 1$, and WOT

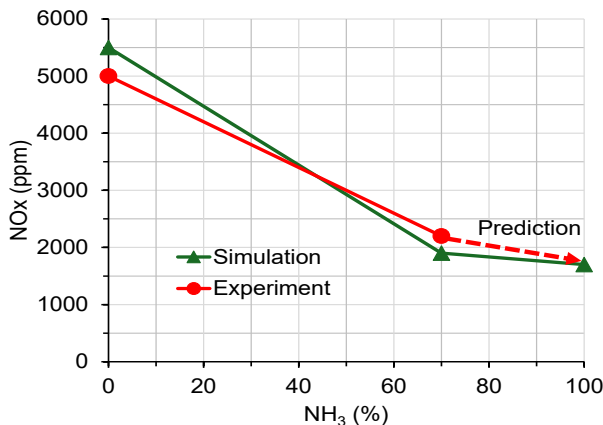


Figure 13. Comparison between simulated and measured NO_x concentrations at MBT for different ammonia fractions at $n = 7500$ rpm, $\phi = 1$, and WOT

Figure 13 compares predicted and measured NO_x concentrations at the corresponding MBT conditions. For gasoline, the simulation predicts 5500 ppm, compared with the measured value of 5000 ppm. For the 30% gasoline-70% NH₃ blend, the corresponding simulated and measured values are 1900 and 2200 ppm, respectively. The absolute deviations are 10% for gasoline and approximately 13.6% for the ammonia-containing blend; the mean absolute deviation is approximately:

$$\bar{\delta}_{NOx} = \frac{10 + 13.6}{2} = 11.8\%$$

The agreement is acceptable for the present CFD-based comparison, considering the sensitivity of NO_x formation to local temperature, equivalence-ratio distribution,

turbulence-chemistry interaction, and uncertainties in exhaust sampling. Together with the agreement between the simulated gasoline MBT timing and the independently measured original spark timing, this result supports the use of the model to estimate trends under experimentally inaccessible conditions.

Direct experimental verification of pure-NH₃ operation and brake-power measurement was not possible with the available equipment. Therefore, the predicted pure-ammonia NO_x concentration of approximately 1700 ppm at MBT should be considered a model-based estimate, not an experimentally validated value. Future work should include controlled pure-ammonia engine tests and direct torque measurement to establish a more comprehensive validation of the model and the practical emissions performance of ammonia operation.

4. Conclusions

This study combined in-cylinder CFD simulation, ignition-timing measurement, and exhaust-emission testing to assess ignition-map requirements for a Honda Lead 110cc motorcycle engine fuelled with gasoline, a 30% gasoline-70% NH₃ blend, and pure NH₃ at 7500 rpm, WOT, and $\phi = 1$. The main conclusions are as follows:

1. The simulated gasoline MBT timing was 33°CA BTDC, at which the predicted power reached 6.40 kW, equal to the rated engine power. This value agreed closely with the original ECU timing measured independently by SmartFi 2 and an oscilloscope, the latter giving 33.1°CA BTDC at 7500 rpm.

2. Increasing the ammonia fraction required systematically earlier ignition because of the slower combustion process. The predicted MBT timing increased from 33°CA BTDC for gasoline to 42°CA BTDC for the 30% gasoline-70% NH₃ blend and 55°CA BTDC for pure NH₃. Relative to gasoline, the required ignition advance was therefore 9°CA for the blend and 22°CA for pure ammonia.

3. At MBT, the 30% gasoline-70% NH₃ blend produced 5.56 kW, corresponding to a 13.0% power reduction relative to gasoline. Pure NH₃ produced 4.63 kW, with a 27.7% power reduction. Thus, a gasoline pilot fraction substantially improves the combustion performance of ammonia in the high-speed motorcycle engine.

4. The predicted NO_x concentration at MBT decreased from 5500 ppm for gasoline to 1900 ppm for the 30% gasoline-70% NH₃ blend and 1700 ppm for pure NH₃. For the two experimentally accessible fuels, the mean deviation between simulated and measured NO_x was approximately 11.8%, supporting the use of the model to estimate pure-ammonia operation.

5. The CFD-derived MBT corrections were translated into remapped ignition characteristics while retaining the original gasoline ECU architecture. This approach provides a practical basis for adapting gasoline motorcycle engines to gasoline-ammonia dual-fuel operation. Further work should experimentally verify pure-NH₃ operation and brake power.

Acknowledgments: This work was supported by the Ministry of Education and Training of Vietnam through the research project “Investigation and Development of an Extended-Range Electric Motorcycle (EREM) Powered by a Hydrogen Carrier Fuel Engine,” Code: B2026-DNA-09.

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