NET ZERO MOTORCYCLE FUELED WITH ETHANOL-AMMONIA FLEXIBLE FUEL

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Abstract - This paper presents the results of simulation and experimental research on converting an electronic fuel-injected Honda RSX motorcycle into an Ethanol-NH_3 dual-fuel motorcycle. This study investigates the influence of air-fuel equivalence ratio (ϕ), ignition timing (φ_s), engine speed (n), and ethanol blending ratio (E0A to E100A) on the performance and emissions of a spark-ignition engine. Through simulation or experimental analysis: It was found that $\phi \approx 1.0$ provides the optimal balance for pressure, temperature, and indicated efficiency (Wi), but also results in peak NOx emissions. Higher engine speeds reduce peak pressure, temperature, HRR, and NOx emissions due to shortened combustion time. Increasing ethanol content in the fuel improves combustion quality, increases peak pressure and temperature, reduces CO emissions, but raises NOx levels due to higher combustion temperatures. The results provide a basis for optimizing ignition timing and fuel blends for improved engine efficiency and lower emissions under various operating conditions.

Key words - Combustion simulation; NH_3; Ethanol; Honda RSX motorcycle; Exhaust gas emissions

1. Introduction

Amidst the rapid and growing exploitation and consumption of fossil fuels - resources that are inherently finite - the world is facing an impending energy crisis. At the same time, global climate change is largely driven by the increasing concentration of greenhouse gases in the atmosphere. Among these, carbon dioxide (CO_2) contributes the most (81%), followed by methane (CH_4) at 10%, nitrous oxide (N_2O) at 5%, and fluorinated gases at 3%. In [1], projections suggest that atmospheric CO_2 levels could double in the first half of this century, significantly accelerating global warming and its associated consequences.

To address the dual challenges of energy demand and environmental protection, numerous studies have explored the use of alternative fuels - particularly Ethanol-NH_3 and its blends with gasoline - in spark-ignition engines as a means to reduce harmful emissions. However, most of these studies have focused on automotive engines, while motorcycles - which typically feature small displacement, high-speed, air-cooled engines - have received comparatively little attention, despite their dominance in many developing countries such as Vietnam [2] - [5].

The rapid advancement of renewable energy technologies and the increasing adoption of water electrolysis as a viable energy storage solution have enabled the economic production of green ammonia, a carbon-free fuel. Upon combustion, ammonia primarily emits water vapor and nitrogen, making it an environmentally favorable alternative to conventional fossil fuels.

In [5], furthermore, as shown in Table 1, ammonia can be stored as a liquid under moderate pressure at ambient temperature, providing an energy density comparable to that of other fuels, particularly regarding its competitive Lower Heating Value (LHV). A life cycle assessment conducted by Bicer and Dincer [6] demonstrated that greenhouse gas emissions could be reduced by nearly a factor of three when using ammonia-powered vehicles instead of gasoline-fueled ones. Although there is currently no commercial transport fleet operating on ammonia, the concept of utilizing it as a substitute for fossil fuels in transportation has been under consideration for several decades.

Combining ethanol with other fuels such as ammonia (NH_3) offers the potential to harness the advantages of each fuel type. Ethanol promotes good combustion, while ammonia presents a carbon-free energy carrier. Recent research has demonstrated that ethanol–ammonia fuel blends can significantly improve engine efficiency and combustion stability. However, such mixtures may also increase emissions of NO_x and CO_2, highlighting the need for further optimization of blending ratios and engine control strategies, [7]. In [8], [9], blending ammonia with methanol or ethanol, combined with a multi-point ignition strategy, is a viable solution to improve the combustion process in SI engines.

In Vietnam, motorcycles are the primary mode of personal transportation, averaging two people per vehicle, and are a major contributor to urban air pollution. Although some efforts have been made to apply alternative fuels such as ethanol-gasoline blends to motorcycles, published studies on ethanol–ammonia blends for these vehicles remain scarce.

In this paper, the development of a hybrid ethanol-ammonia fuel system for motorcycles, applied to the Honda RSX, is proposed as a practical and feasible solution to reduce dependence on fossil fuels and minimize environmental pollution. The main contributions of this paper are as follows:

Modification of the engine fuel supply system, including: microcontroller programming connected to Electronic Control Unit (ECU), arrangement of NH_3 and Ethanol injectors, establishing principles for flexible adjustment of Ethanol-NH_3 fuel composition.

Experimental measurement of engine pollution emissions running on Ethanol-NH_3 and comparison with simulation results at specific operating conditions.

***Table 1.** Ammonia properties and comparison with other fuels at 300 K and 0.1 MPa, unless stated otherwise*

Species	Ammonia	Methanol	Hydrogen	Methane	Gasoline
Formula	NH_3	CH_3OH	H_2	CH_4	-
Storage	**Liquid**	**Liquid**	**Compressed**	**Compressed**	**Liquid**
Storage temperature (K)	300	300	300	300	300
Storage pressure (MPa)	1.1	0.1	70	25	0.1
density@ storage conditions (kg.m^3)	600	784.6	39.1	187	~740
FL in air (vol.%)	15-28	6.7-36	4.7-75	5-15	0.6-8
LBV @ stoichiometry (m.s^{-1})	0.07	0.36	3.51	0.38	0.58
Auto-ignition T (K)	130	119	>100	120	90-98
LHV (MJ/kg)	18.8	19.9	120	50	44.5

2. Material and method

The combustion chamber of the J52C engine is simulated using Ansys Fluent software to determine the distribution of Ethanol and NH_3 mixture, combustion and emissions. The fuel is defined by the percentage ratio of % E and % NH_3 (E15A indicates 15% Ethanol and 85% NH_3).

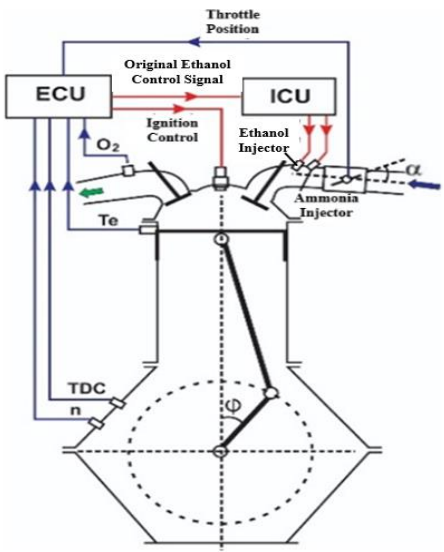

***Figure 1.** Schematic diagram of J52C engine with modified fuel system*

Figure 1 describes the J52C engine installed on the Honda RSX motorcycle, which has been modified with separate NH_3 and Ethanol injection systems, where the gasoline injector is used to inject Ethanol. The gasoline ECU and sensors remain unchanged. An Injection Control Unit (ICU) and an injector are added to control NH_3 injection. The NH_3 injector does not necessarily need to be mounted directly into the intake manifold but can be installed at any convenient location near the injection pipe placement. Figure 2 describes the intake manifold after modification and NH_3 injection pipe installation.

The Ethanol-NH_3 injection system for Honda RSX motorcycle engines, as described in Figure 3, is designed based on the original electronic fuel injection system of the motorcycle. The output signal from the ECU, instead of controlling the gasoline injector as originally designed, becomes the input signal for the ICU such as Arduino microcontroller. The output signal from the ICU is divided into two channels at a ratio, which is controlled either manually or automatically by the microcontroller program, determining the injection timing. One channel controls the Ethanol injector, while the other controls the NH_3 injector. This method has the advantage of utilizing the entire structure of the existing gasoline injection system, simplifying the modification of the engine's fuel system.

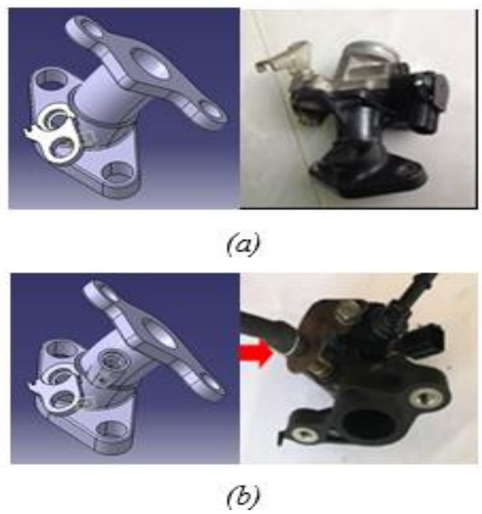

***Figure 2.** a - Original intake manifold, b - modified intake manifold for NH_3 injector installation*

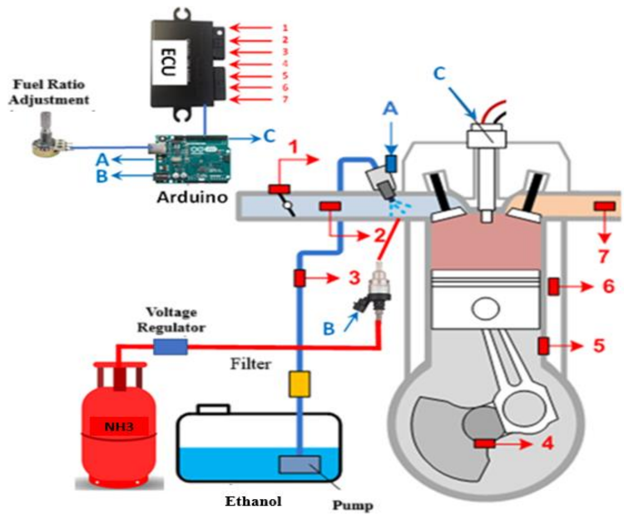

***Figure 3.** Multi-fuel Supply System Diagram for Gasoline-Ethanol-NH_3 Engine in Motorcycle*

1. Throttle Position Sensor; 2. Intake Pressure Sensor; 3. Fuel pressure Sensor; 4. Speed Sensor; 5. Knock Sensor; 6. Cylinder Wall Temperature Sensor; 7. Oxygen Sensor; A. Ethanol Injector Control Signal; B. NH_3 Injector Control Signal; C. Ignition Control Signa.

3. Result and discussion

3.1. Modeling Ethanol-NH_3 Mixture Formation

Figure 4 illustrates that, before ignition, the area around the spark plug exhibits an optimal relative richness ratio of the mixture (ϕ approximately 1.0). The NH_3-rich region is predominantly concentrated near the spark plug, while the Ethanol-rich region is located farther away. This distribution of fuel concentration is highly advantageous for the combustion process. Given Ethanol's higher octane rating, its distribution farther from the spark plug helps minimize the risk of knocking as the pressure and temperature of the mixture rise. The octane stratification of the fuel mixture presents a notable benefit for forced ignition engines utilizing Ethanol.

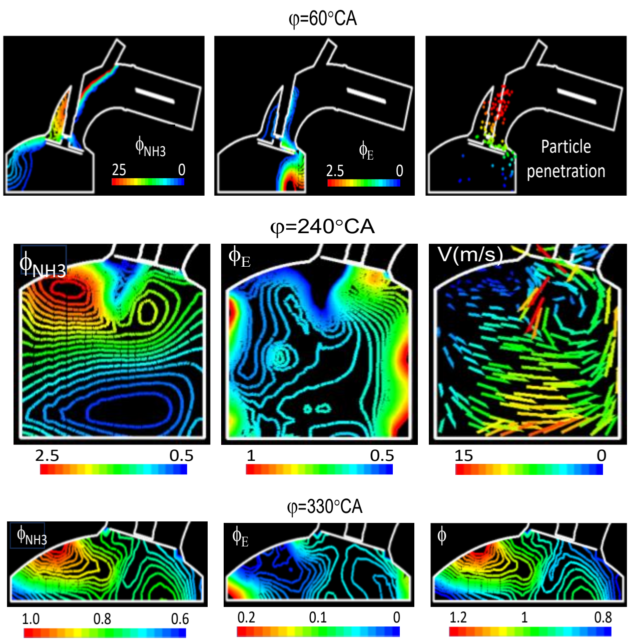

Figure 4. *Evolution of particle density and distribution of NH_3, Ethanol in the cylinder*

After injection begins, approximately 15° crankshaft angle later, the Ethanol particles are drawn into the cylinder by the intake air flow, causing the particle density to increase rapidly. When in contact with the hot air in the cylinder, the first Ethanol particles evaporate quickly, causing their density to decrease rapidly. As the amount of particles drawn into the cylinder continues to increase, the gas temperature in the cylinder decreases, causing the particle density to increase again. At around 90° crankshaft angle, due to increased air flow velocity in the cylinder enhancing heat transfer, the evaporation rate increases again, thus reducing particle density.

3.2. Combustion Process Simulation

3.2.1. Influence of Relative Richness Ratio of the Mixture

Figure 5, 6, 7, 8 show the effect of air-fuel equivalence ratio on engine performance and emissions. In which, Figure 5 illustrates the relationship between combustion pressure p (bar) and crank angle φ (°CA), where the pressure reaches a peak of 39.96 bar at the optimal equivalence ratio ($\phi = 1.0$). The combustion temperature T (K) also reaches a maximum of 2137K at $\phi = 1.0$, consistent with the optimal combustion conditions shown in Figure 6. CO emissions increase sharply when $\phi > 1.0$, which can be explained by an insufficient supply of oxygen to fully oxidize CO to CO_2, as depicted in Figure 7. In Figure 8, the NO_x concentration reaches a maximum of 923 ppm at $\phi = 1.0$. This is due to the fact that the formation mechanism of NO is highly sensitive to temperature, and the peak temperature at $\phi = 1.0$ provides the optimal condition for NO_x formation.

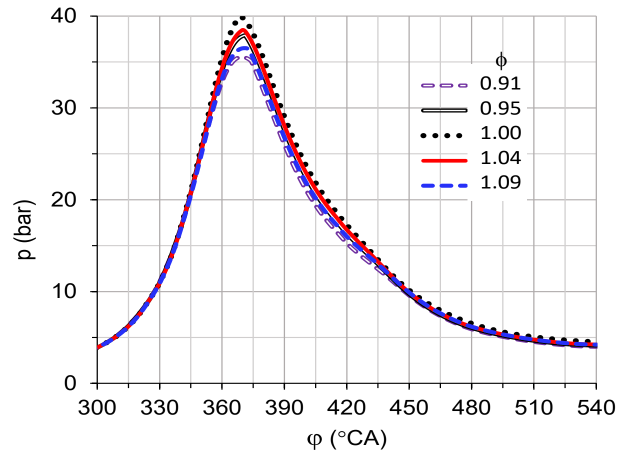

Figure 5. *Effect of air-fuel equivalence ratio on cylinder pressure variation according to crankshaft rotation angle*

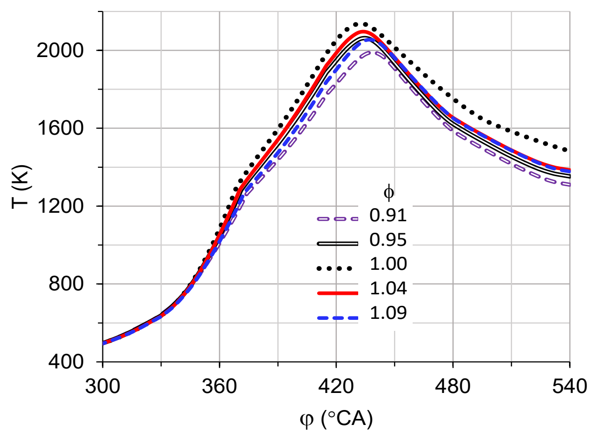

Figure 6. *Effect of air-fuel equivalence ratio on combustion temperature T (K)*

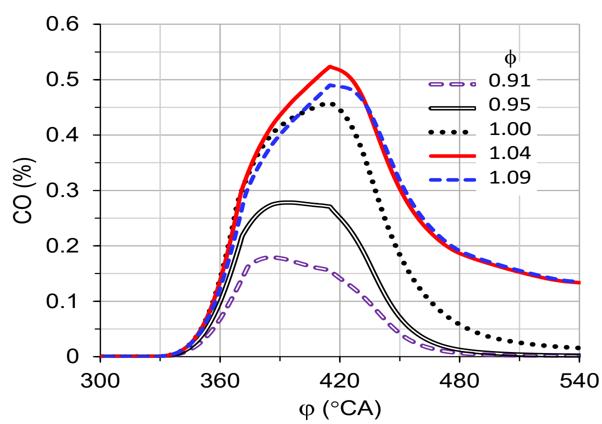

Figure 7. *CO emissions corresponding to air-fuel equivalence ratio values*

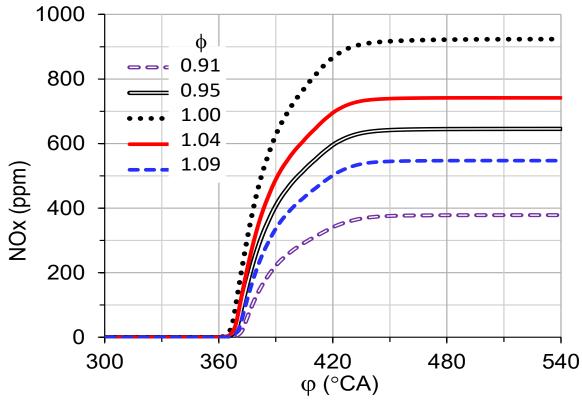

Figure 8. *NO_x emissions corresponding to air-fuel equivalence ratio values*

Table 2. *The relationship between the equivalence ratio (ϕ) and key combustion parameters*

ϕ	0.95	0.91	1	1.04	1.09
Pmax (bar)	37.9	35.55	39.96	38.49	36.5
Tmax (K)	2063	1989	2137	2096	2055
Texhaust (K)	1352	1310	1483	1385	1378
Wi (J/cyc)	100.95	94.01	108.99	104.17	99.84
Pe (kW)	5.68	5.29	6.13	5.86	5.62
COmax (%V)	0.28	0.18	0.46	0.52	0.49
COexhaust (%V)	0.002	0.001	0.016	0.133	0.135
HC (%V)	0.002	0.001	0.003	0.42	1.113
NOx (ppm)	646	378	923	742	547

3.2.2. Influence of Ignition Advance Angle

Increasing the ignition advance angle increases the maximum pressure within the combustion process. As the ignition advance angle continues to increase, the engine's indicated work per cycle also rises. For a particular engine operating mode, the area of the work diagram reaches its maximum at the optimal ignition advance angle as in Figure 10.

Figure 11 illustrates NOx emissions as a function of crank angle for various ignition advance angles (φ_s). Increasing the ignition advance angle (φ_s) results in higher NOx emissions due to earlier combustion and elevated combustion temperatures. The NOx concentration reaches its peak value of 1039 ppm at an ignition advance angle of φ_s = 35° crank angle.

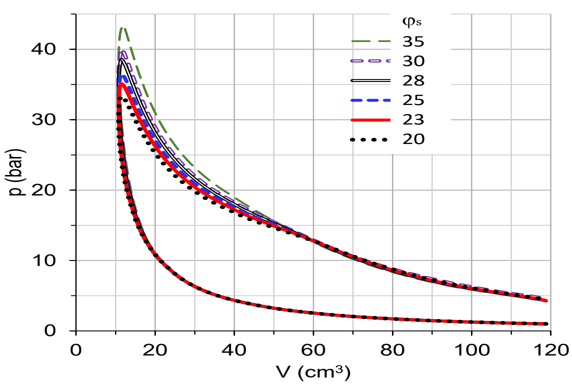

Figure 10. *Effect of ignition advance angle on cylinder pressure variation according to crankshaft rotation angle*

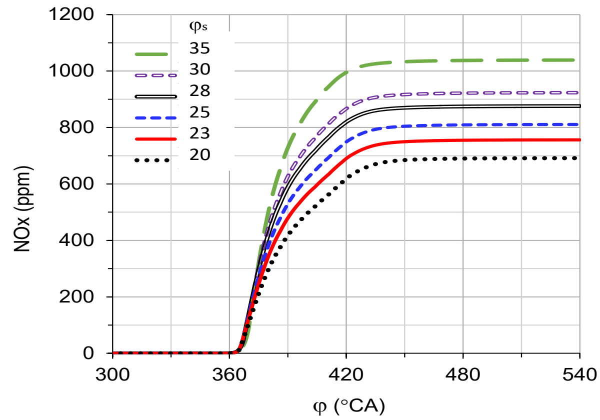

Figure 11. *NO_x emissions corresponding to ignition advance angle values*

Table 3. *Effect of ignition timing (φ_s) on thermodynamic parameters and emissions of the engine fueled with ethanol–ammonia blend*

φs (°CA)	20	23	25	28	30	35
Pmax (bar)	33.32	35.1	36.51	38.55	39.96	43.41
Tmax (K)	2098	2103	2108	2115	2137	2137
Texhaust (K)	1426	1416	1408	1397	1483	1374
Wi (J/cyc)	98.41	100.9	102.86	105.34	108.99	111.08
COmax (%V)	0.41	0.42	0.43	0.45	0.46	0.49
COexhaust (%V)	0.016	0.014	0.013	0.011	0.016	0.009
HC (%V)	0.002	0.002	0.003	0.003	0.003	0.003
NOx (ppm)	691	756	811	876	923	1039

3.2.3. Effects of Engine Speed

Figure 12 shows the relationship between pressure P (bar) and crank angle φ (°CA) at different engine speeds. As the engine speed increases, the peak pressure decreases due to the shorter combustion duration. The maximum pressure reaches 49.98 bar at an engine speed of 2000 rpm.

The heat release rate (HRR) as a function of crank angle decreases significantly with increasing engine speed, as shown in Figure 13. This is because the combustion reaction does not have sufficient time to complete.

Figure 14 presents the relationship between temperature T(K) and crank angle φ (°CA) for various engine speeds. As engine speed increases, the temperature decreases due to shorter combustion time and greater heat losses.

Table 4. *Variation of parameters with engine speed*

n (rpm)	2000	3000	4000	5000	6000	7500
Pmax (bar)	15	15	15	15	15	15
Tmax (K)	48.98	41.77	38.66	35.1	32.89	30.37
Texhaust (K)	2341	1250	1338	1416	1482	1593
Wi (J/cyc)	121.43	113.04	107.3	100.9	96.56	88.64
Pe (kW)	1.82	2.54	3.22	3.78	4.35	4.99
COmax (%V)	0.84	0.6	0.5	0.42	0.38	0.34
COexhaust (%V)	0.002	0.005	0.008	0.014	0.022	0.04
HC (%V)	0.002	0.002	0.003	0.002	0.002	0.002
NOx (ppm)	4197	2547	1482	756	417	171

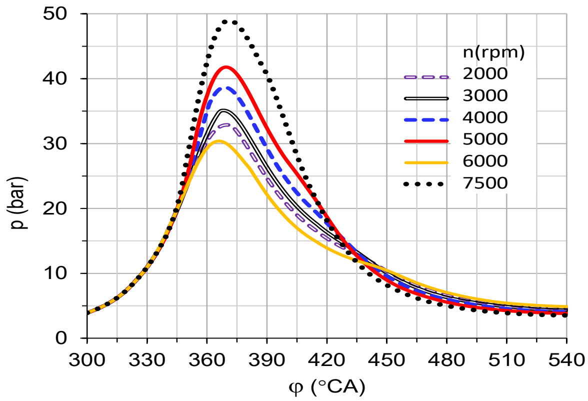

Figure 12. *Effect of engine speed on cylinder pressure variation according to crankshaft rotation angle*

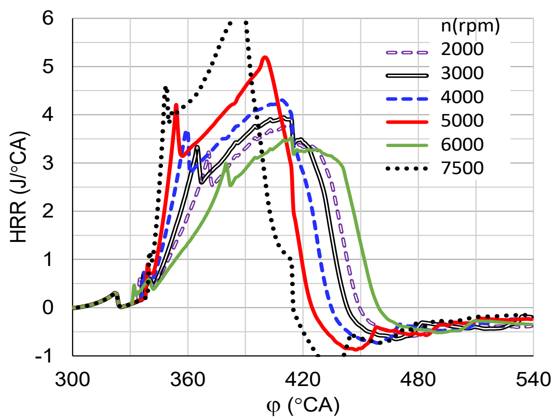

Figure 13. *Effect of engine speed on heat release rate*

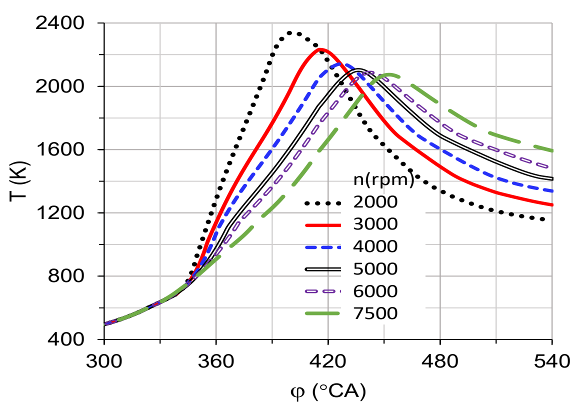

Figure 14. *Effect of engine speed on combustion temperature T (K)*

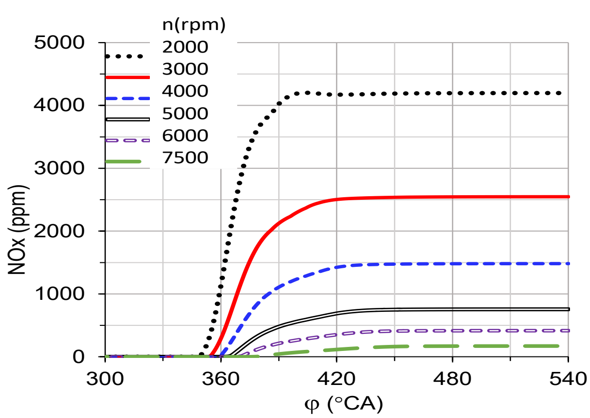

Figure 15. *NO_x emissions corresponding to engine speed*

NO_x emissions decrease with increasing engine speed because lower temperatures reduce the formation of NO_x. The NO_x concentration reaches a maximum value of 4197 ppm at an engine speed of 2000 rpm in Figure 15.

3.2.4. Effects of Ethanol Content

Figure 16, shows the combustion pressure *p* as a function of crank angle *φ* for fuel mixtures of ethanol and ammonia. When the engine runs on pure ammonia fuel (E0A), the peak combustion pressure reaches 34.13 bar. As the ethanol content increases to E15A and E50A, the pressure rises to 35.1 bar and 36.59 bar respectively, and reaches a maximum of 37.38 bar with pure ethanol (E100A). This can be explained by ethanol having a higher octane number and better combustion characteristics.

Since ethanol contains oxygen within its molecular structure, it enhances combustion quality and supports the oxidation of CO to CO_2, thereby reducing CO emissions as the ethanol content increases, resulting in cleaner combustion as shown in Figure 17.

Figure 18 indicates that the peak combustion temperature of ammonia in the engine is 2007K, while ethanol reaches a higher peak combustion temperature of 2236K. Consequently, the NOx emissions of the engine using ammonia fuel are only 463 ppm, compared to 1465 ppm when using ethanol, as illustrated in Figure 19.

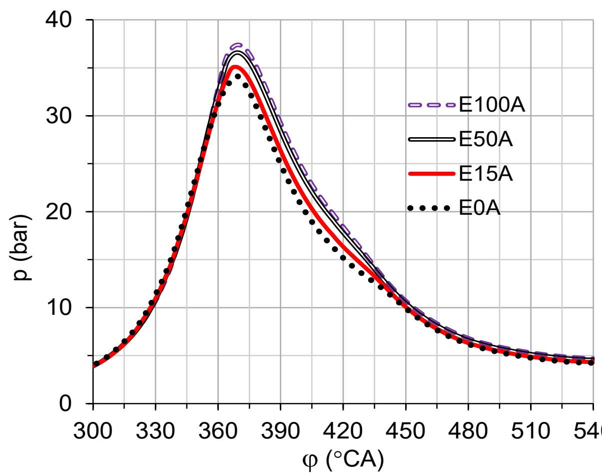

Figure 16. *Effect of ethanol content on cylinder pressure variation according to crankshaft rotation angle*

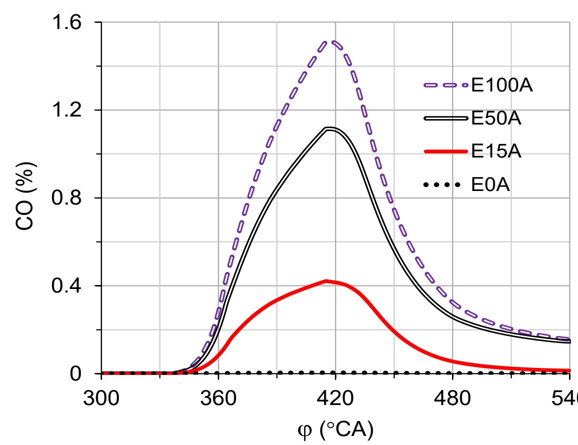

Figure 17. *CO emissions corresponding to ethanol content*

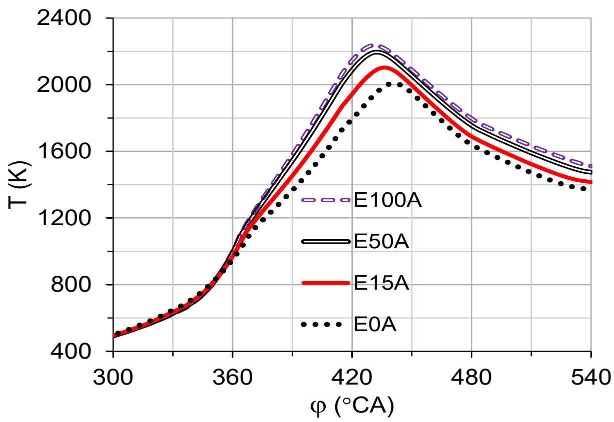

Figure 18. *Effect of ethanol content on combustion temperature T (K)*

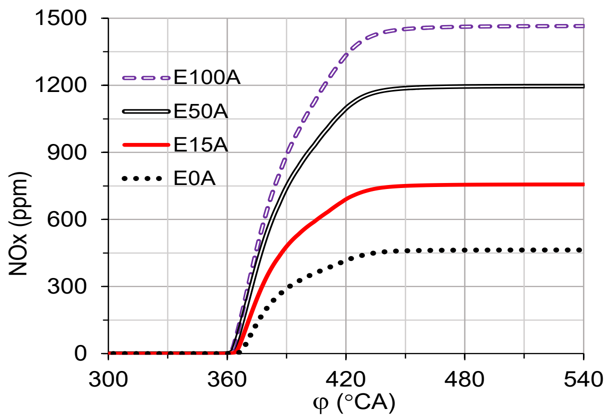

Figure 19. *NO_x emissions corresponding to ethanol content*

Table 5. *Comparison of Efficiency and Emissions of Different Ethanol Fuel Blends*

Fuel	**E0A**	**E15A**	**E50A**	**E100A**
ϕ	1	1	1	1
n (rpm)	5000	5000	5000	5000
φs (ºCA)	23	23	23	23
Pmax (bar)	34.13	35.1	36.59	37.38
Tmax (K)	2007	2103	2195	2236
Texhaust (K)	1369	1416	1475	1511
Wi (J/cyc)	93.05	100.9	110.77	115.24
Pe (kW)	3.49	3.78	4.15	4.32
COmax (%V)	0	0.42	1.11	1.51
COexhaust (%V)	0	0.014	0.147	0.156
HC (%V)	0.003	0.002	0.011	0.001
NOx (ppm)	463	756	1196	1465

4. Conclusion

Based on the research results presented, the paper draws the following conclusions:

Ethanol has low volatility at low temperatures, which affects engine performance during startup or under low-load conditions. Ammonia effectively supports the engine using ethanol under these operating conditions.

Ammonia has a lower calorific value than traditional fuels, which results in a decrease in engine power as the NH_3 content in ethanol increases. At an engine speed of 5000rpm, the power output when running entirely on ethanol and completely on ammonia are 4.32 kW and 3.94 kW, respectively.

The peak combustion temperature of ammonia in the engine is 2007K, whereas the peak combustion temperature of ethanol is 2236K, which results in NOx emissions from an ammonia-powered engine being only 463 ppm compared to 1465ppm when running on ethanol.

As engine speed decreases, indicated cycle work increases, NO_x emissions rise, but CO emissions decrease when using an ethanol-ammonia fuel blend.

The combination of ammonia and ethanol is an effective solution for applying renewable fuels in internal combustion engines. Ammonia helps reduce NO_x and CO emissions while improving the ethanol engine's performance in low ambient temperature conditions.

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