Article citation info:
Nguyen Thin Q, Nguyen Van D. An analytical study on the performance of spark-ignition engines operating with ammonia fuel. Combustion Engines. 0000;XXX(X):xx-xx. https://doi.org/10.19206/CE-216441

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An analytical study on the performance of spark-ignition engines operating with ammonia fuel

ARTICLE INFO

Received: 10 July 2025
Revised: 17 December 2025
Accepted: 6 January 2026
Available online: 24 January 2026

Ammonia is a promising fuel due to its low cost, carbon-free energy carrier, renewable hydrogen carrier, scalability and flexibility, global availability, and long-term energy storage. It can be synthesized from renewable energy sources and used as a carbon-free fuel in internal combustion engines. When ammonia is completely burned, due to the relatively low combustion temperature of the ammonia-air mixture and the absence of carbon in the molecule, the only toxic emissions are nitrogen oxides. These nitrogen oxide emissions can be effectively reduced by the selective catalytic reduction technology, in which ammonia acts as a reductant in the selective catalytic reduction system. This study presents the characteristics of a spark-ignition internal combustion engine 4CH 7.6/7 powered by ammonia fuel. The results show that the maximum brake torque on the wide-open throttle operating characteristics increases by 6.1%, while the maximum brake power decreases by 30%, and the brake-specific ammonia fuel consumption is almost 2 times higher than that of gasoline at the same engine power. Moreover, with a research octane number of ammonia up to 110, it will help improve anti-knock properties. Along with that, increasing the compression ratio and using turbochargers can be considered to improve engine performance.

Key words: *SI engine, gasoline fuel, ammonia fuel, knock probability determination.*

1. Introduction

Climate change resulting from greenhouse gas emissions has become an increasingly pressing global issue. A substantial portion of these emissions – including greenhouse gases and other harmful pollutants – originates from human activities, particularly from internal combustion engine vehicles. As most automobiles still operate on fossil fuels, the transportation sector is a major contributor to global greenhouse gas emissions and environmental pollution. In response to international and national commitments to achieving net-zero emissions, researchers and automotive manufacturers are actively seeking solutions to reduce both greenhouse gas emissions and toxic pollutants released into the environment. The solutions being studied and applied are improving the mixing quality of the fuel-air mixture, utilizing high-pressure turbocharging, employing exhaust gas recirculation, and utilizing exhaust gas treatment units [5, 9, 13]. Among the solutions that have been researched and implemented to address emissions are the adoption of alternative fuels and the utilization of renewable energy sources. Environmentally friendly alternative fuels derived from biomass – renewable within the carbon cycle – have garnered significant attention. However, recent research has increasingly focused on fuels that do not emit greenhouse gases during combustion and can serve as substitutes for gasoline in spark-ignition engines. Notably, hydrogen and ammonia have emerged as promising candidates for use in future vehicle applications.

Ammonia is a naturally occurring inorganic chemical compound that is lighter than air and liquefies easily. The boiling point of ammonia is –33.43°C at 1.013 atm and solidifies into colorless crystals at –77.7°C [3]. It is widely used in agriculture and is typically produced from hydrogen derived from fossil fuels, which means it is not considered a "green" product. However, green ammonia can be produced if the hydrogen is generated by alternative methods such as water electrolysis using wind or solar energy. Green ammonia could become a cheaper fuel for the shipping industry compared to hydrogen produced from renewable energy, as it is easier to store and can be combusted in standard internal combustion engines. Ammonia's low viscosity promotes better spray and mixing with air compared to gasoline. Green ammonia is also being researched as a fuel for aircraft, by exposing ammonia to a catalyst to split it into nitrogen and hydrogen, which is then burned in an aircraft engine [14].

The properties of different fuels are compared, as shown in Table 1. It can be observed that the lower heating value of gasoline is twice as high as that of ammonia, and the flame speed of gasoline is up to 39 times greater than that of ammonia. Additionally, the energy density of gasoline is nearly three times higher than that of ammonia [2, 10, 11].

The adoption of alternative fuels such as NH_3 and H_2 plays a pivotal role in efforts to reduce carbon dioxide emissions in the transportation sector, while also driving the transition toward a sustainable energy ecosystem. Some countries have actively incorporated NH_3 into their energy strategies as a low-carbon fuel. Notably, in January 2020, the U.S. House of Representatives introduced a draft bill recognizing NH_3 as a “low-carbon fuel”. At the same time, the Royal Society in the United Kingdom released a policy report titled “Ammonia: zero-carbon fertiliser, fuel and energy store”, providing scientific foundations and policy analysis to support the integration of NH_3 into clean energy technologies [4]. However, research has shown that compared to other fuels, ammonia fuel has certain drawbacks, including a high auto-ignition temperature and a slow flame speed, which are among the most significant challenges faced by ammonia-fueled engines [2, 3, 8, 14].

Table 1. Compare the properties of gasoline with hydrogen and ammonia [2, 3, 10–12].

Property	Gasoline	Gaseous H_2	Liquid H_2	Ammonia
Molecular	C_8H_8	H_2	H_2	NH_3
Fuel density [kg/m^3]	698.3	17.5	71.1	602.8
Lower heating value [MJ/kg]	44.5	120.1	120.1	18.8
Energy density [MJ/m^3]	31,074	2101	8539	11,333
Volumetric energy density at 0.1 MPa and 293 K [MJ/L]	34	0.010–0.011	–	11.5
Laminar burn velocity [m/s]	0.58	3.51	3.51	0.15
Auto-ignition temperature [K]	573	844	844	930
Research octane number (RON)	90–98	> 130	> 130	110
Mass carbon content [%]	85.5	–	–	–
Flammability limits [vol%]	1.4–7.6	4–75	4–75	16–25
Absolute minimum ignition energy [mJ]	0.14	0.018	–	8.0
Latent heat of vaporazation [kJ/kg]	305	–	445.6	1370
Quench distance [mm]		0.64	–	7

Ammonia (NH_3) can be used as an alternative fuel for internal combustion engines (ICEs). With the explosion of fuel cell research, ammonia has also been studied as a fuel source for proton exchange membrane fuel cells (PEMFCs), alkaline fuel cells (AFCs), and solid oxide fuel cells (SOFCs). SOFCs are the most efficient when compared to ammonia fuel cells and ICEs. However, their disadvantages are power density, load adaptability, and relatively high battery manufacturing costs. When operating at near maximum power, the efficiency of internal combustion engines has been shown to be higher than both ICEs and AFCs. In addition, internal combustion engines have great advantages in terms of manufacturing costs, maintenance costs, global repair teams, long durability, acceptable power density, and operating characteristics [1, 6].

ICEs using NH_3 as an alternative fuel have performance parameters quite comparable to conventional ICEs at both full load and partial load modes. However, conventional ICEs typically produce significantly higher levels of toxic emissions, including hydrocarbons (HC), nitrogen oxides (NO_x) and particulate matter (PM). Most importantly, CO_2 emissions from fossil fuel combustion in ICEs are one of the main contributors to greenhouse gas emissions, thereby aggravating climate change [3, 14]. Therefore, research on carbon-free fuels such as hydrogen and ammonia has received widespread attention both in the automotive industry and in research institutes.

Although H_2 is emerging as a key element in the transition to a carbon-neutral economy, widespread adoption still faces challenges related to safety, vehicle placement to ensure mileage comparable to conventional fuel engines, storage capacity, and distribution infrastructure for hydrogen. Finally, a very important factor to consider is its relatively low volumetric energy density (2.9 MJ/L at 70 MPa), rapid combustion rate, and high exhaust gas temperature [14]. In contrast, NH_3 offers a higher volumetric energy density (7.1 MJ/L) along with an existing transportation infrastructure, providing clear advantages in terms of flexibility and efficiency. These characteristics position NH_3 as a practical choice for future energy systems involving storage, transportation, and power generation. These characteristics position NH_3 as a practical option for future energy systems involving storage, transport. With easier storage than hydrogen, it can take advantage of existing liquefied petroleum gas or compressed natural gas infrastructures for use in converted internal combustion engines [3]. This is a factor that helps reduce fuel costs and makes it more accessible to the market. NH_3 helps create many realizable solutions to address the sustainability challenge in the energy sector.

The use of ammonia as an alternative fuel also faces several obstacles, such as:

– The rate of chemical reaction occurring in the combustion chamber is lower than with traditional fuels due to the high ignition temperature and low flame propagation speed when burning ammonia. This slow chemical reaction rate results in some unburned ammonia being released with the exhaust gases. As a result, ammonia combustion mainly produces NO emissions, with small amounts of NO_2 and nitrogen oxides N_2O, but also unburned ammonia emissions [12]. In this case, a solution for handling unburned NH_3 in the exhaust treatment needs to be considered and studied.

– Ammonia has a high latent heat of vaporization of 1370 kJ/kg. For example, ethanol, liquid hydrogen, and gasoline have latent heats of vaporization of 840, 445.6, and 305 kJ/kg, respectively. This means that if ammonia is injected into the engine combustion chambers, the combustion temperature can drop dramatically, causing incomplete combustion and reducing engine efficiency. The flame extinction distance for a stoichiometric mixture of air and ammonia is 7 mm (Table 1), which is about 10 times more than in the case of hydrogen. This means that heat loss through the walls of the combustion chamber will be lower in the case of ammonia.

– The use of ammonia in SI engines is limited by narrow flammability limits and low flame speed, causing incomplete combustion.

– The effects of ammonia on lubricating oil and engine lifespan have not been fully studied [12].

It can be observed that ammonia is a promising fuel with scalability, flexibility, global availability, and long-term energy storage capability. This study presents the simulation results of a gasoline engine operating on ammonia under the Miller cycle. The findings serve as a foundation for converting conventional engines to operate on green ammonia fuel [7].

2. Research subject and research model

The engine used for this study is a four-cylinder, four-stroke 4CH 7.6/7 engine. Gasoline is injected into the intake manifold, the engine is naturally aspirated and ignited by spark plugs. The main specifications of the 4CH 7.6/7 engine are presented in Table 2.

Table 2. The main specifications of the 4CH 7.6/7 engine

Parameters [unit]	Value
Piston stroke/cylinder diameter [mm/mm]	71/76
Number of cylinders [–]	4
Stroke number [–]	4
Compression ratio [–]	10.5
Engine displacement [cm^3]	1289
Maximum power [kW]	60
Speed at maximum power [min^{-1}]	5000
Maximum torque [Nm]	100
Speed at maximum torque [min^{-1}]	2000
Environmental standards	Euro 4
Inlet valve:	
Diameter [mm]	36
Valve lift [mm]	8.8
Open [deg. CA bTDC]	33
Duration opening [deg. CA]	292
Exhaust valve:	
Diameter [mm]	31
Valve lift [mm]	8.65
Open [deg. CA bTDC]	47
Duration opening [deg. CA]	244

To establish the simulation model of the above-mentioned engine, the following assumptions and considerations are made:

1. The intake gases inside the cylinder are considered ideal and are assumed to remain nearly in thermodynamic equilibrium. During the gas exchange process, the cylinder space is an open thermodynamic system with a changing volume. When this process is completed, the cylinder becomes a closed system.

2. The in-cylinder processes are described using the differential form of the law of conservation of energy, states. The mass of the gases exchanged during the intake process is conserved. In addition, the law of conservation of energy is applied to the flow of the medium through the intake ports. To describe the state of the gas mixture in the combustion chamber, the Clapeyron equation is used.

3. The temperature and composition of the gas mixture determine its internal energy. The heat transfer equations and the quasi-static flow model represent the boundary conditions, heat exchange and mass exchange between the system and the surrounding environment. Heat transfer to the cylinder wall is calculated using Newton's law of cooling.

4. The heat transfer coefficients are determined by the Woschni model. These coefficients are determined separately for the intake, compression and expansion processes. The heat transfer through the wall and the heat exchange of the system are calculated taking into account the temperature distribution along the cylinder and the change in contact area when the volume changes due to the piston movement.

5. The gas dynamic boundary conditions are analyzed in the region adjacent to the cylinder. During gas exchange, the pressure in the intake and exhaust manifolds is assumed to be constant. The gas flow through the valve orifice, which has a time-varying area, is calculated using a one-dimensional, quasi-steady flow model. Additionally, the processes of exhaust gas expulsion and fresh charge intake are also considered.

6. The software supports the simulation of the supercharging system's operation through two main modules: one for simulating the working cycle of a spark-ignition (SI) engine, and another for simulating the integration of the SI engine with a turbocharging system. The schematic diagram of the SI engine model integrated with a turbocharger, following the sequence of the two modules, is shown in Fig. 1.

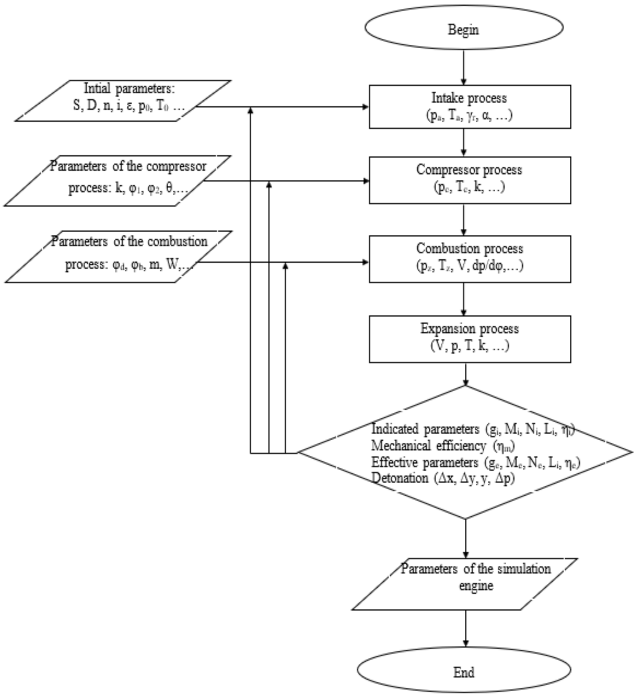

Fig. 1. Mathematical model of 4CH 7.6/7 engine using turbocharger

The boundary condition parameters used for the simulation model are presented in Table 3.

Table 3. Boundary condition parameters

Inlet temperature	298 K
Inlet pressure	0.1 MPa
Exhaust pressure	0.102 MPa
Exhaust temperature	843 K
Piston surface temperature	548 K
Piston tail temperature	573 K
Cylinder wall temperature	518 K

In this study, Wiebe's calculation method is used to calculate the heat release in the combustion chamber of the engine. The Wiebe formula is written as (1).

$$x(\varphi) = 1 - e^{-W\left(\frac{\varphi-\varphi_b}{\varphi_d}\right)^{(m+1)}} \quad (1)$$

where: m, W – coefficient of determination of the heat dissipation curve, φ – current time of the combustion process, φ_b is the crankshaft angle position where the combustion process begins, φ_d – is the combustion duration which is determined by the crankshaft angle.

Depending on the type of fuel used and the operating conditions of the engine, the values of m, W and φ_d need to be determined appropriately to obtain accurate model results.

With the value of the coefficient m, a series of calculations are performed and compared with the experimental values for different model internal combustion engines. The value of m for the model is chosen so that the simulation results obtained (maximum pressure in the combustion chamber and maximum pressure increase rate of the engine) are close to the experimental values.

The thermal efficiency of the SI engine can be improved by achieving an optimal combustion phase and a higher compression ratio. However, this can lead to engine knocking. The phenomenon of engine knocking in an SI engine is one of the main causes of typical damage to the engine's cylinder-piston system. Figure 2 illustrates the knock combustion process of fuel in an SI engine when running on gasoline.

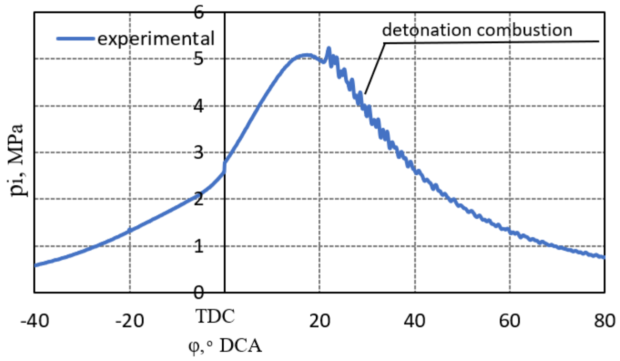

Fig. 2. Knock combustion process in the SI engine 4CH 7.6/7 when using gasoline (n = 4000 rpm; α = 1; IT = 20° bTDC)

For the numerical solution to this problem, a specific index (y) has been chosen, and the calculation is performed using the formula of Daud and Eise [4], which is essentially the Livengood-Wu integral:

$$y = \frac{1}{19.75}\left(\frac{ON}{100}\right)^{-3.4107} \int_0^{\tau} (10.2p)^{1.7} e^{\frac{-3800}{T}} d\tau \qquad (2)$$

where ON is the octane rating of the fuel; τ is the current time since the beginning of compression (seconds); p and T are the corresponding pressure (MPa) and temperature (K) in the engine cylinder; e is the base of the natural logarithm (e = 2.7183).

3. Model accuracy evaluation

To assess the accuracy of the established model, it is necessary to compare the pressure results obtained from the model with the experimental results.

The simulation results are compared with the test results of the 4CH 7.6/7 engine running on gasoline at a crankshaft speed of n = 3000 rpm, with an air-fuel ratio α = 1 and an ignition timing (IT) angle of 15° before top dead center (bTDC). The heat release duration is φ_z = 70°. The results are shown in Fig. 3. It can be seen that the pressure variation in the combustion chamber during simulation closely follows the results measured on the actual engine. The pressure value of the model is broken at the TDC and is larger in the expansion stage.

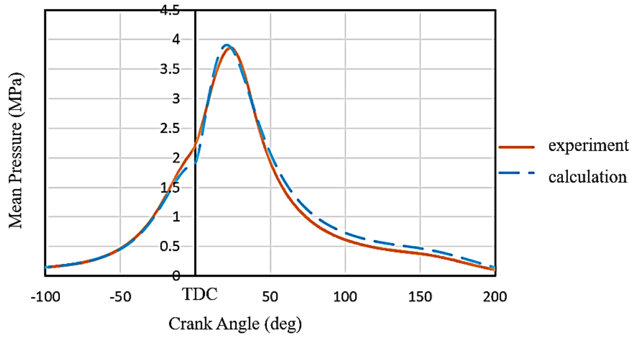

Fig. 3. Simulation and experimental results of SI 4CH 7.6/7 gasoline engine (n = 3000 min^{-1}; α = 1; IT = 15° bTDC; m = 1.7; φ_z = 70°)

Continuing with another engine, the study also evaluated the simulation results performed with an SI 4CH 11.5/7.7 engine using ammonia, with the parameters presented in Table 4. The test engine was placed in the laboratory and converted to operate using ammonia. Figure 4 shows the comparison between the model calculations and experimental data at the engine speed operating mode n = 1000 rpm; excess air coefficient of α = 1, IT = 15° bTDC, φ_z = 52°. In this case, it can be seen that the peak pressure value of the model result and the experimental result coincide at 15 degrees after TDC. However, the pressure value in the combustion chamber of the model is smaller than the experimental value but the deviation is still within the acceptable range.

Table 4. Parameters of the 4CH 11.5/7.7 converter engine using ammonia

Parameter	Meaning
Number of cylinders [–]	4
Stroke number [–]	4
Cylinder diameter [mm]	115
Piston stroke [mm]	77
Connecting rod length [mm]	177
Compression ratio [–]	11.75
Number of valves per cylinder [–]	4
Power [HP]	120
Torque [Nm]	160
Environmental standards	Euro 4
Number of valves [–]	16

It can be observed that, in both test cases, the model yields results that are relatively close to the experimental data. The deviation in value is clearly observed at the TDC, the point where the pressure reaches its maximum, and a part of the expansion phase. However, the error value is within an acceptable range. Therefore, the model can be used for the subsequent stages of research.

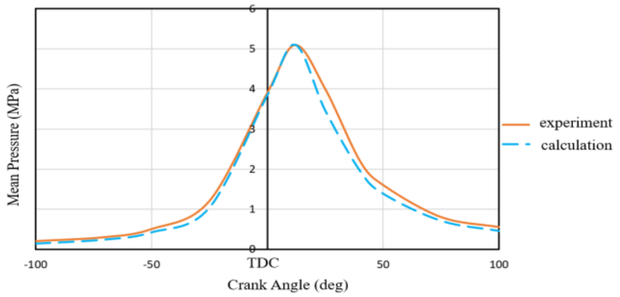

Fig. 4. Comparison of experimental and calculated indicator diagrams of SI engine 4CH 11.5/7.7 when operating on ammonia (n = 1000 min^{-1}; α = 1; IT = 15° bTDC; m = 1.2; φ_z = 52°)

4. Results and discussion

4.1. Engine external characteristics

The external characteristics of the engine are the first focus when converting the engine to run on ammonia. Figures 5a and 5b show the external characteristic results of the engine before and after the conversion. The characteristics are determined when the engine operates at the maximum throttle opening, and the ignition timing is adjusted to achieve the maximum power output.

From the results, it can be seen that when the engine switches from gasoline to ammonia as fuel, the maximum brake torque (M_{emax}) increases by 6.1%, while the maximum brake power (N_{emax}) decreases by 30% when measured under external characteristic conditions. Due to ammonia's lower volumetric energy density and calorific value compared to gasoline, the specific fuel consumption when using ammonia is nearly twice as high as gasoline for the same engine power output. This issue needs to be addressed in maintaining engine power during the transition, as the amount of fuel required per cycle must be recalculated. Additionally, the fuel tank volume may need to be larger to ensure that the vehicle's range remains unchanged.

In addition, another issue that needs to be recalculated is the air/fuel ratio when converting the engine to use ammonia. The theoretical air/fuel ratio when using ammonia is determined through the reaction of ammonia with oxygen in the air through reaction equation 3:

$$4NH_3 + 3O_2 = 2N_2 + 6H_2O + \text{heat} \tag{3}$$

The air/fuel ratio when using NH_3 can be calculated as follows:

$$\alpha_{NH_3} = \frac{m_a}{m_f} \tag{4}$$

where: m_a is mass of air. If air is considered to consist of two components, nitrogen m_{N_2} and oxygen m_{O_2}, then m_a can be calculated as follows:

$$m_a = m_{O_2} + m_{N_2} = 32 \times 3 + 3 \times 79/21 \times 28 = 415.72 \text{ g}$$

It should be noted that nitrogen makes up approximately 79% and oxygen makes up 21% by mass of air.
m_f is the mass of fuel.

$$m_f = 4 \times 17 = 68 \text{ g}$$

$$\alpha_{NH_3} = \frac{m_a}{m_f} = \frac{415.72}{68} = 6.11$$

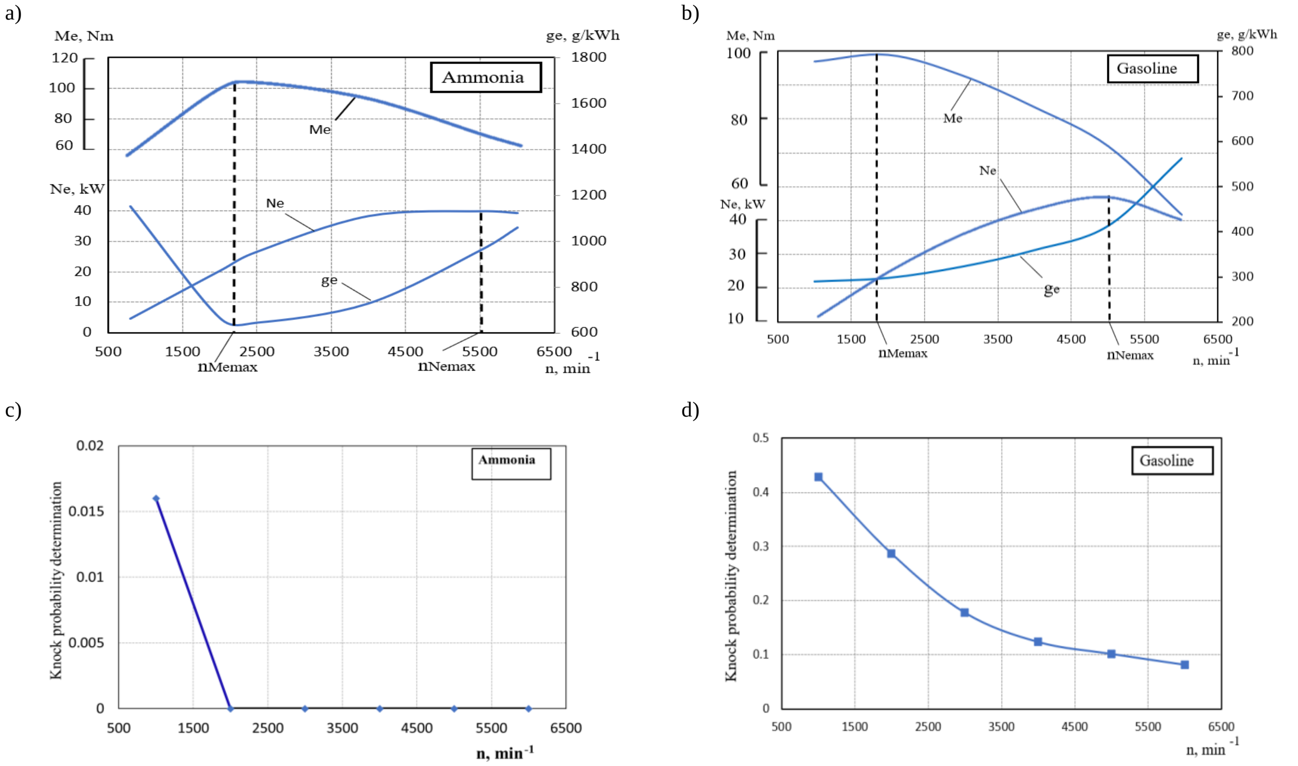

Fig. 5. External speed characteristics and knocking probability of the SI engine 4CH 7.6/7: a, c) engine using ammonia; b, d) engine using gasoline

If we calculate the A/F ratio by volume, we can determine as follows:

$$\alpha^{V}{}_{NH_3} = \frac{V_a}{V_f} = \frac{n_a}{n_f} \quad (5)$$

where V_a, n_a respectively the volume or moles of air, V_f, n_f respectively the volume or moles of fuel.

$$\alpha^{V}{}_{NH_3} = \frac{3 + 3 \times 79/21}{4} = 3.57$$

Percentage of ammonia occupying in the combustion chamber:

$$\%NH_3 = \frac{V_{NH_3}}{V_{total}} \quad (6)$$

where V_{NH_3} is volume of NH_3 in the mixture and V_{total} is mixture volume.

$$V_{total} = V_{NH_3} + V_a$$

Therefore, we get the following result:

$$\%NH_3 = \frac{4}{3 + \frac{3 \times 79}{21} + 4} = 0.22$$

Thus, the A/F ratio of the engine using ammonia has a big change from 14.7 to 6.11. The injection process needs to ensure the amount of air/fuel according to the calculation results.

For SI engines, another critical factor is evaluating the stability of the engine's operation. This factor is the octane rating, used to assess the engine's knocking resistance. Since ammonia has a higher octane rating than gasoline, the probability of knocking in the forced ignition engine when using ammonia decreases. The results are shown in Figures 5c and 5d. It can be observed that for the same engine speed, the probability of knocking occurrence when using ammonia decreases by a factor of 100 to 150 times.

4.2. Effect of compression ratio

The compression ratio is defined as the ratio of the volume of the combustion chamber when the piston is at bottom dead center to the volume when it is at top dead center. The compression ratio significantly affects the efficiency of the engine. The effect of the compression ratio on the engine efficiency of the ideal cycle is shown in the following expression as follows:

$$\eta_t = 1 - \frac{1}{\varepsilon^{k-1}} \quad (7)$$

where: ε is compression ratio; k is the ratio of specific heats and it is defined as follows:

$$k = c_p/c_v \quad (8)$$

where: c_p is the specific heat at constant pressure ($J \cdot kg^{-1} \cdot K^{-1}$), c_v is the specific heat at constant volume ($J \cdot kg^{-1} \cdot K^{-1}$).

Thus, as the compression ratio increases, the efficiency of the cycle also increases.

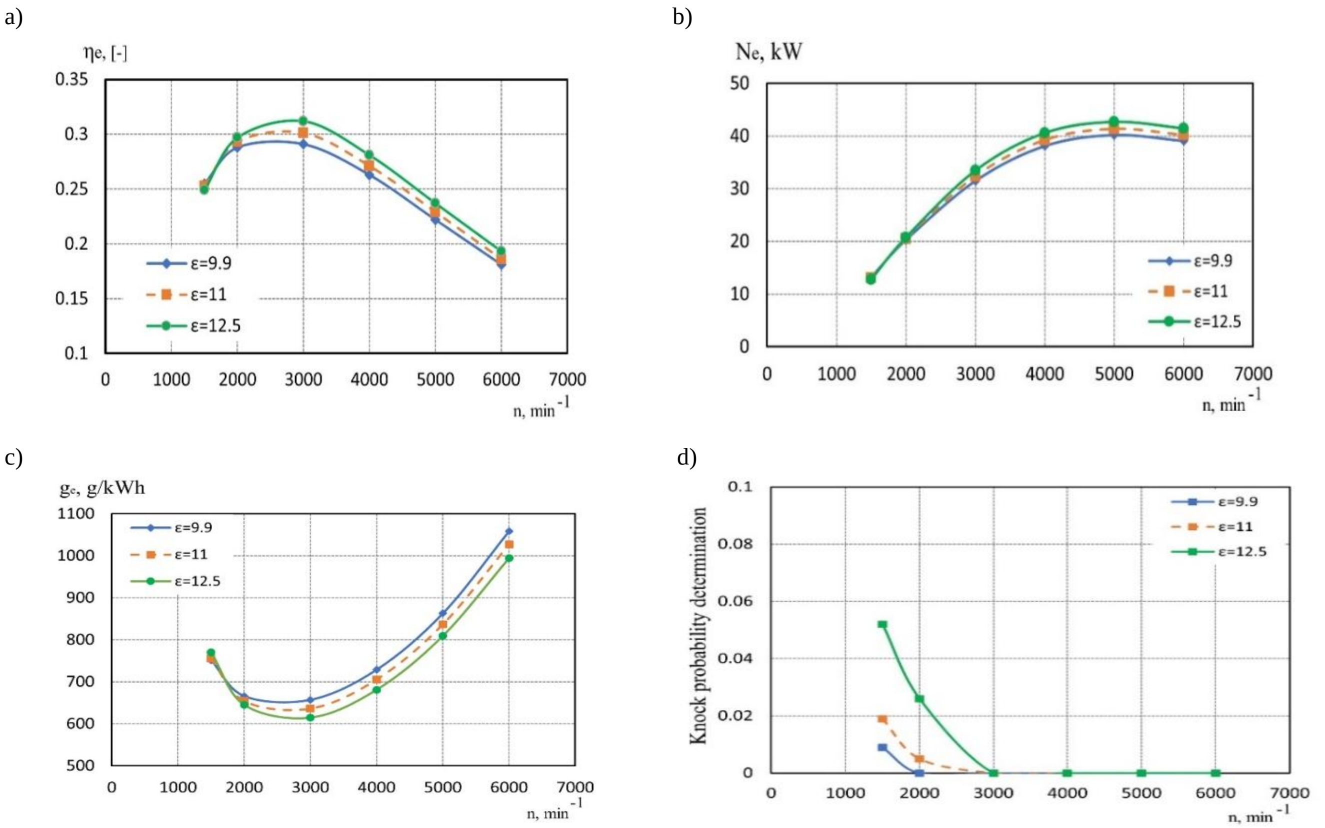

Fig. 6. Variation of engine parameters when using ammonia fuel with different compression ratios: a) brake efficiency; b) brake power; c) brake specific fuel consumption; d) probability of knocking occurrence

Due to the higher octane rating, engines using ammonia can achieve a higher compression ratio. With a high octane number, engines using ammonia can increase the compression ratio. In the calculation model, the engine using ammonia was simulated with compression ratios ranging from 9.9 to 12.5 under full load. The ignition timing was optimized for each speed mode and compression ratio, and the fuel-air mixture was adjusted according to the prescribed ratio for ammonia. The results of the engine are shown in Fig. 6.

It can be seen that, when operating at speeds ranging from 1500 to 6000 rpm, with the throttle fully open and without turbocharging, the probability of knocking combustion when using ammonia fuel decreases as the engine speed increases, and this value increases as the compression ratio of the engine increases (Fig. 6d).

The engine's efficiency and brake power increase as the compression ratio increases. The brake efficiency reaches its maximum at n = 3000 rpm (Fig. 6a), while the brake power reaches its maximum at n = 5000 rpm (Fig. 6b). After reaching the maximum, these values decrease as the engine speed increases. In addition, the minimum BSFC was found in the speed range from 2000 to 3000 rpm (Fig. 6c). The results obtained are a solution to improve the performance of gasoline engines when converting fuel. We can increase the compression ratio for gasoline engines or a further solution is to convert diesel engines to be able to use ammonia.

4.3. Effect of boost pressure

When considering forced ignition engines, an improved cycle of interest is the Miller cycle. The Miller cycle is a variation of the Otto cycle developed to improve thermal efficiency and reduce emissions in internal combustion engines. This cycle was invented by Ralph Miller in 1947 and is often applied to turbocharged gasoline or diesel engines.

The Miller cycle changes the intake valve closing timing to control the compression process, thereby reducing the work consumed during compression and increasing overall efficiency. There are two methods for adjusting the intake valve closing timing: early intake valve closing and late intake valve closing. Gasoline engines using the Miller cycle help reduce the work consumed during compression, increase thermal efficiency, reduce the risk of knocking, and reduce NO_x emissions due to the lower combustion temperature.

In the model, the intake valve closing timing is adjusted to enable the engine to operate according to the Miller cycle. The effect of boost pressure on the forced ignition engine operating under the Miller cycle is considered at an engine speed of 2000 rpm, with the throttle fully open, and the ignition timing optimized for the corresponding boost values. The variation of the cycle parameters is shown in Fig. 7.

The probability of knocking increases as the intake manifold pressure increases and decreases as the intake valve closing angle increases, since it depends on the amount of new fuel-air mixture supplied to the engine. The conversion to ammonia helps the engine have better knock resistance compared to gasoline engines (Fig. 7a, b). The maximum brake torque and brake power are achieved at maximum boost pressure and minimum intake valve closing angle, and these values are higher when the engine runs on ammonia compared to gasoline. As the intake valve closing angle increases, brake specific torque increases (Fig. 7c, d).

The brake efficiency of the engine improves by increasing the intake manifold pressure and intake valve closing angle. In this case, there is no significant difference in brake efficiency when comparing ammonia fuel with gasoline fuel (Fig. 7g, h).

BSFC decreases as the boost pressure increases in both modelling cases (with ammonia and gasoline). This is consistent with the conclusions reached when applying turbochargers to internal combustion engines. Under naturally aspirated engine conditions, the BSFC of an ammonia engine (746 g/kWh) is more than twice that of a gasoline engine (325 g/kWh) at 120 °CA bTDC (Fig. 7e, f). The main reason for the increase in fuel consumption is the large difference in the lower calorific value between the two types of fuel and the same intake valve closing angle.

5. Conclusion

It can be seen that ammonia is an environmentally friendly fuel and also has a renewable nature. Because NH_3 has no carbon in its molecular composition, the combustion products do not contain greenhouse gases. In addition, the advantages of storage, transportation and supply to vehicles make ammonia a fuel with great potential.

When switching from gasoline to ammonia at the engine's external characteristics, the maximum brake torque on the external characteristics increases by 6.1%, while the maximum brake power decreases by 30%. Along with that, with the calorific value and volumetric energy density of NH_3 being lower than that of gasoline, the brake specific fuel consumption of the engine when using NH_3 increases by almost two times. This is the problem that needs to be solved for the engine's fuel system when converting.

Moreover, with a higher octane value than gasoline, it allows for an increase in the compression ratio of the converted engine. This helps to improve the performance of the NH_3 engine operating on the Miller cycle.

Finally, the research results also show that it is possible to apply the solution of increasing the boost pressure to improve the engine's working parameters. When increasing the initial pressure, the engine's performance and torque increase while the engine's brake-specific fuel consumption decreases. When the boost pressure increases up to 3 bar, the engine's detonation ability is equivalent to that of a gasoline engine without turbocharging.

Based on the results obtained, corresponding solutions will be proposed to improve the engine's working process when converting.

Acknowledgements

This research is funded by the University of Transport and Communications (UTC) under grant number T2026-CK-006.

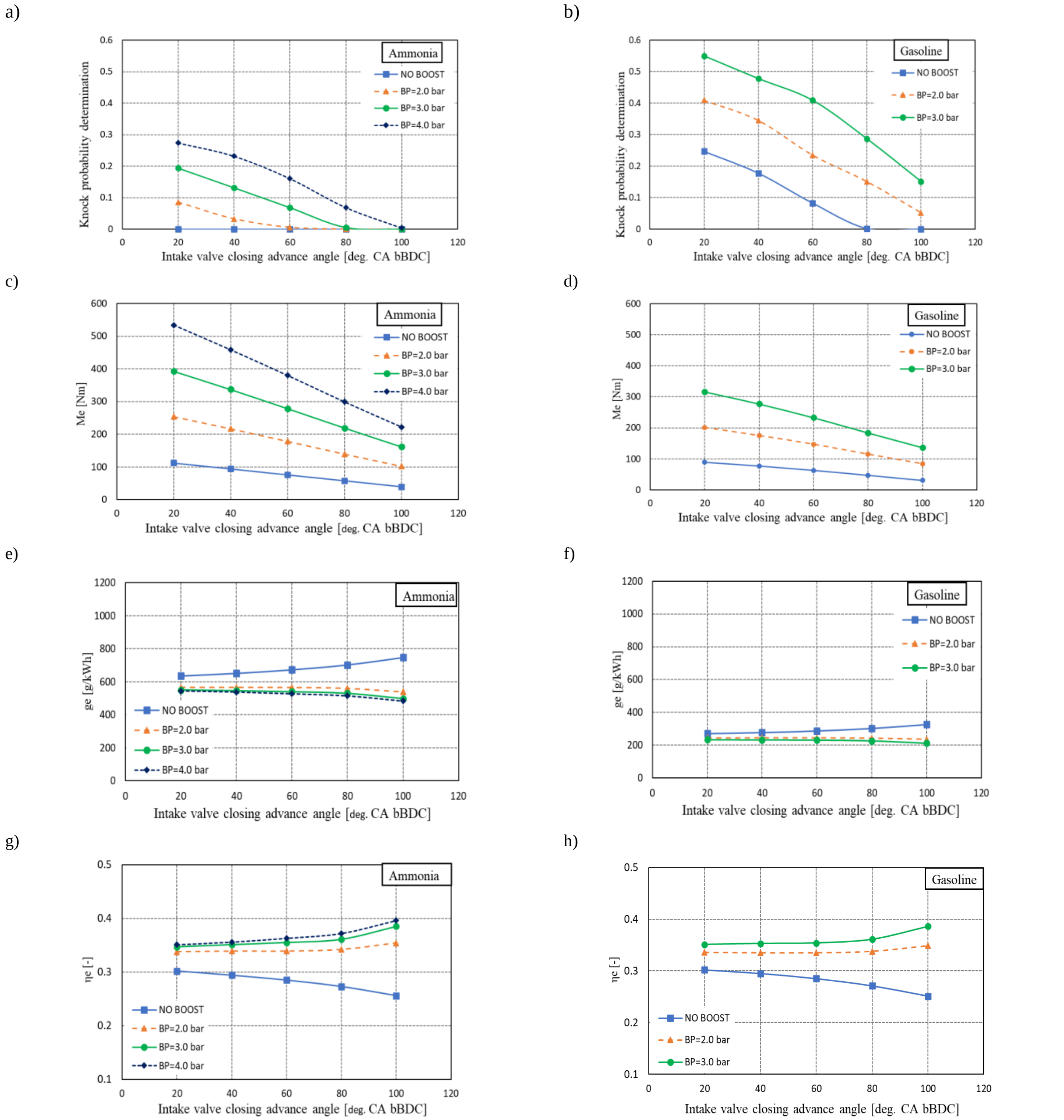

Fig. 7. The operating parameters of the SI 4CH 7.6/7 engine working in the Miller cycle with a variable intake valve closing angle. a, b) probability of knocking occurrence; c,d) brake torque ; e,f) brake specific fuel consumption; g,h) brake efficiency

Nomenclature

AFCs	alkaline fuel cells
bBDC	before bottom dead center
BP	boost pressure
BSFC	brake specific fuel consumption
bTDC	before top dead center
H_2	hydrogen
HC	hydrocarbons
ICEs	internal combustion engines
IT	ignition timing
NH_3	ammonia
NO_x	nitrogen oxides
PM	particulate matter
PEMFCs	proton exchange membrane fuel cells
SI	spark ignition
SOFCs	solid oxide fuel cells

Bibliography

[1] Briceno-Mena LA, Venugopalan G, Romagnoli JA, Arges CG. Machine learning for guiding high-temperature PEM fuel cells with greater power density. Patterns. 2021;2(2). https://doi.org/10.1016/j.patter.2020.100187

[2] Cardoso JS, Silva V, Rocha RC, Hall MJ, Costa M, Eusébio D. Ammonia as an energy vector: current and future prospects for low-carbon fuel applications in internal combustion engines. J Clean Prod. 2021;296:1256562. https://doi.org/10.1016/j.jclepro.2021.126562

[3] Cheng Q, Muhammad A, Kaario O, Ahmad Z, Martti L. Ammonia as a sustainable fuel: review and novel strategies. Renew Sustain Energy Rev. 2025;207:114995. https://doi.org/10.1016/j.rser.2024.114995

[4] Douaud AM, Eyzat P. Four-octane-number method for predicting the anti-knock behavior of fuels and engines. SAE Technical Paper 780080. 1978. https://doi.org/10.4271/780080

[5] Dunin AY, Nguyen Thin Quynh, Golubkov LN, Computational study of the effect of increasing the fuel injection pressure up to 3000 bar on the performance of the diesel engine and its gaseous emissions. International Conference on Engineering Management of Communication and Technology (EMCTECH). 2020:65-70. https://doi.org/10.1109/EMCTECH49634.2020.9261516

[6] Giddey S, Badwal SPS, Munnings C, Dolan M. Ammonia as a renewable energy transportation media. ACS Sustainable Chem Eng. 2017;5(11):10231-10239. https://doi.org/10.1021/acssuschemeng.7b02219

[7] Hong C, Ji C, Wang S, Xin G, Wang Z, Meng H et al. An experimental study of various load control strategies for an ammonia/hydrogen dual-fuel engine with the Miller cycle. Fuel Proc Technol. 2023;247:107780. https://doi.org/10.1016/j.fuproc.2023.107780

[8] Huang Z. Fuel blend combustion for decarbonization. Proc Combust Inst. 2024;40(1-4). https://doi.org/10.1016/j.proci.2024.105776

[9] Kuta K, Nadimi E, Przybyła G, Żmudka Z, Adamczyk W. Ammonia CI engine aftertreatment systems design and flow simulation. Combustion Engines. 2022;190(3):3-10. https://doi.org/10.19206/CE-143158

[10] Kim K, Roh G, Kim W, Chun K. A preliminary study on an alternative ship propulsion system fueled by ammonia: environmental and economic assessments. J Mar Sci Eng. 2020; 8(3). https://doi.org/10.3390/jmse8030183

[11] MacFarlane DR, Cherepanov PV, Choi J, Suryanto BHR, Hodgetts RY, Bakker JM et al. A roadmap to the ammonia economy. Joule. 2020;4(6):1186-1205. https://doi.org/10.1016/j.joule.2020.04.004

[12] Stępień Z. Ammonia as an alternative fuel to internal combustion engines. Combustion Engines. 2025;200(1):117-127. https://doi.org/10.19206/CE-200289

[13] Thin Quynh N, Duc Le H, Dunin, A. Influences of injector geometry parameters on fuel injection characteristics and parameters of a diesel engine. Transport and Communications Science Journal. 2023;74(4):530-543. https://doi.org/10.47869/tcsj.74.4.12

[14] Valera-Medina A, Xiao H, Owen-Jones M, David WIF, Bowen PJ. Ammonia for power. Prog Energy Combust Sci. 2018;69:63-102. https://doi.org/10.1016/j.pecs.2018.07.001

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