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Article

Improving Ammonia Combustion in Spark-Ignition Engines Coupling Hydrogen Enrichment with Multiple Spark Plugs

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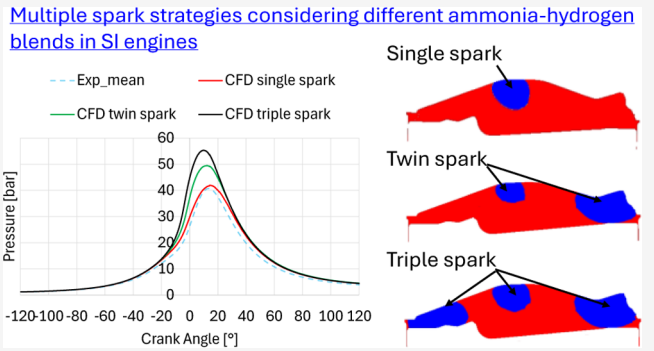
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ABSTRACT: The use of ammonia as a fuel for internal combustion engines has attracted a growing amount of interest in recent years. However, its poor combustion properties have led to extensive research focused on identifying solutions to the challenges posed by ammonia combustion. In particular, several studies have examined the potential of hydrogen enrichment to improve combustion efficiency in spark-ignition engines that use ammonia as a fuel. Recently, the use of multiple spark plugs has also been proposed for the same purpose. The aim of the work is to assess the performance and emissions of a spark-ignition ammonia engine coupling hydrogen enrichment with multiple spark strategies (up to three spark plugs). A 3D-CFD model, coupled with a chemical kinetic approach, already validated based on experimental tests, has been used. Three different configurations have been analyzed: single spark, twin spark, and triple spark. In the analyzed conditions, the gross indicated mean effective pressure ($IMEP_H$), moving from a single to twin spark strategy, increased by 9% and 11% and 6.5%, respectively, for neat ammonia and 5% and 15% of hydrogen by volume in the fuel mixture. The addition of a third spark plug led to negligible improvements of the combustion process considering hydrogen-enriched fuel blends. The multiple spark strategies allow an improvement in terms of gross indicated fuel conversion efficiency, with the maximum increase reached with 5% hydrogen by volume in the fuel blend, equal to about 12%, corresponding to a decrease of the gross indicated specific fuel consumption (GISFC) of about 11%. The emissions of unburned ammonia decreased by about 13%, 8%, and 3%, respectively, for the twin spark and by about 14%, 11%, and 5% for the triple spark, moving from neat ammonia fueling to 15% of hydrogen by volume in the fuel mixture. Considering multiple spark strategies, NO_x emissions decreased even if only a slight reduction in terms of NO emissions has been obtained.



INTRODUCTION

Ammonia[1] is emerging as a viable carbon-free fuel[2] for internal combustion engines and serves as a potential carrier for hydrogen production.[3,4] Its storage at ambient conditions is straightforward, and it finds utility across various applications.[1] However, ammonia exhibits significantly inferior combustion characteristics compared to other fuels; for instance, at stoichiometric conditions, its laminar flame speed is 87% lower than that of gasoline, as shown in Figure 1.[5] This leads to longer combustion durations, more advanced spark angles, and higher cycle-to-cycle variability.[5] To enhance the combustion performance of ammonia in internal combustion engines, blending it with another carbon-free fuel, such as hydrogen, may prove beneficial.[5−8] Current research is focused on ammonia−hydrogen mixtures, with existing literature providing correlations for estimating laminar flame speed[9,10] and ignition delay.[11]

Scharl et al.[12] conducted an investigation into the spray combustion and mixture formation of ammonia−air mixtures using a rapid compression−expansion machine. By integrating one-dimensional analyses with optical studies, they discovered that ammonia flames tend to lack stability postignition. Sun et al.[13] investigated the combustion characteristics of an ammonia spark-ignition engine for different air excess ratios, comparing also the engine performance in case of methane fueling. They achieved optimal combustion performances and the highest power output, considering an air excess equal to 0.9 with a flame speed of 13.7 m/s. Lhuillier et al.[14,15] performed experimental studies on the combustion behavior of ammonia in a gasoline direct injection spark-ignition engine that was modified for port fuel injection. At a constant engine speed of 1500 rpm, they compared the combustion characteristics of the engine

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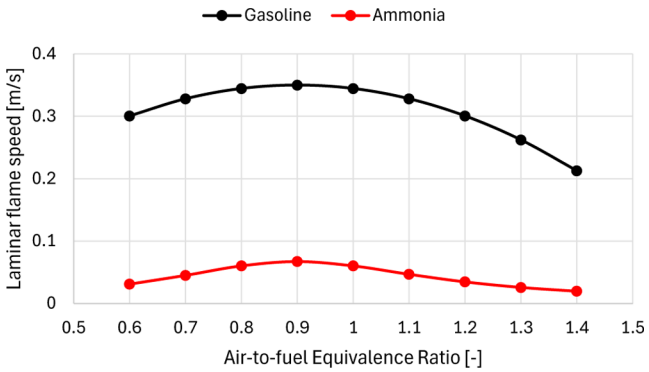


Figure 1. Laminar flame speed at ambient conditions for gasoline−air mixtures and ammonia−air mixtures.

operating on pure ammonia versus those of ammonia−hydrogen blends. Their findings indicated that while ammonia can function as a fuel in a spark-ignition engine, significant advancement of the spark angle is necessary to ensure proper engine operation. In ref 16, through both one-dimensional and three-dimensional analyses, the impact of boost pressure, load conditions, and spark advance on the performance of a light-duty spark-ignition engine fueled by ammonia was evaluated. The results indicated that the operational range of the engine is considerably limited when using pure ammonia with stable combustion primarily occurring at low to medium engine speeds and high boost pressures. Zhang et al.[17] conducted a numerical analysis of the combustion characteristics in a spark-ignition engine operating on ammonia−hydrogen blends, where hydrogen was derived from exhaust-fuel reforming. The researchers reported that incorporating hydrogen up to 10% by volume in the fuel mixture resulted in an elevated cylinder pressure and reduced combustion duration. Klawitter et al.[18] assessed the combustion behavior of an ammonia engine considering partially cracked ammonia. They considered ammonia cracking ratio equal to 0%, 7.5%, and 10% and an equivalence ratio from 0.7 to 1.2 varying the engine speed and the turbulence. They found that the partial cracking of ammonia improved the combustion behavior. Finally, they obtained flame speed values close to those of methane with an ammonia cracking ratio of 10%.

To enhance the performance of an ammonia-fueled engine, not only hydrogen enrichment but also modifications to the engine's structural design are beneficial, such as the implementation of prechambers[19−22] and multiple spark plugs. In ref 23, the authors explored the effects of multiple spark plugs on the combustion dynamics of a compression ignition engine that had been converted to operate as a spark-ignition engine using ammonia. Their findings indicated that the use of multiple spark plugs increased the burn rate and power output, decreased the combustion duration and unburned ammonia emissions, and improved fuel economy. Notably, they observed that the introduction of multiple sparks did not result in elevated NO_x emissions. Udden et al.[24] performed experimental investigations into the combustion and flame characteristics of a spark-ignition engine employing a multiple spark strategy. The results indicated that a single spark plug positioned at the center of the combustion chamber produced a lower heat release rate and a reduced pressure peak, whereas multiple spark plugs facilitated a higher heat release rate, increased in-cylinder pressure, and shorter combustion duration. The authors emphasize the reduction in cyclic variation associated with the heightened pressure in the combustion chamber. Additionally, flame analysis revealed that multiple spark plugs enabled faster flame propagation compared to that of a single spark configuration. In ref 25, the same researchers examined the influence of the equivalence ratio on various spark strategies. Their findings indicated that as the mixture transitioned from stoichiometric to lean conditions, the concentration of NO_x emissions rose, reaching a peak at an air excess ratio of 1.4. However, in this scenario, a subsequent reduction in the NO_x levels was observed, attributed to the decreased pressure and temperature within the combustion chamber.

The presented literature review shows that there is a great interest in the improvement of ammonia combustion using the multiple sparks strategy, but there are only few studies about this topic, and none of them try to couple the effect of both hydrogen enrichment and multiple sparks. The novelty of this study is the assessment of the combined effect of both hydrogen enrichment and multiple spark strategies to improve ammonia combustion in spark-ignition engines. In particular, by means of a 3D model coupled with a chemical kinetic approach already validated based on experimental tests, this study proposes different multiple spark strategies to assess the performances and emissions of a spark-ignition engine considering multiple spark strategies (up to three spark plugs) and different amounts of hydrogen in the fuel blend.

METHODOLOGY

The main features of the engine under study are presented in Table 1. It is a GDI 4-cylinder engine adapted to become a single cylinder engine running with premixed charges. It is characterized by a pent roof chamber with four valves for each cylinder.

Table 1. Engine Characteristics

model	PSA EP6DT
displacement volume [cm^3]	399.5
bore/stroke/conrod [mm]	77/85.8/138.5
compression ratio [-]	10.5

Simulations were conducted using the AVL Fire software, employing a Reynolds-Averaged Navier−Stokes (RANS) methodology. The finite volume method was applied to resolve the partial differential transport equations, while a first-order Euler implicit differencing scheme was utilized for time discretization. Second-order schemes were implemented for the continuity and momentum equations, whereas an upwind scheme was adopted for both the turbulence and energy equations. The turbulence modeling was executed by using the k-z-f approach. The thermodynamic properties of the fluid were determined based on the ideal gas law, and heat transfer through the engine walls was modeled using a standard wall function.

The combustion process of the fuel blends under consideration has been modeled through a chemical kinetic approach, specifically incorporating the mechanism proposed by Stagni et al.[26] into the three-dimensional code. This mechanism, outlined in Table 2, addresses the decomposition of ammonia, hydrogen abstractions, and the dissociation of the HNO intermediate while also accounting for the formation of nitrogen oxide (NO) resulting from both ammonia dissociation and the oxidation of nitrogen present in the air.

The proposed model has been previously validated against the experimental results obtained considering the described engine

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Table 2. Main Characteristics of the Stagni Mechanism[26]

number of species	31
number of reactions	203
pressure [bar]	1
temperature [K]	500−2000
overall air−fuel ratio	1

operating in the conditions reported in Table 3, as detailed in ref 27. The experimental setup has been described in ref 27. In summary, the engine is powered by an electric motor, and the primary shaft is fitted with a Kübler optical encoder, which allows for angular position monitoring with a resolution of 0.1 Crank Angle Degree (CAD). In-cylinder pressure is measured using a water-cooled AVL piezoelectric pressure transducer, also with a resolution of 0.1 CAD, and it operates within a measuring range of 0 to 25 MPa. The temperatures and pressures of the engine's intake and exhaust are monitored through type K thermocouples and piezoresistive absolute pressure transducers, respectively. Fuel flow is measured and regulated by Brooks thermal mass flowmeters, which have an accuracy of ±0.7%, and the fuel is preheated to an intake temperature of 323 K before being mixed in an intake plenum prior to injection. The analysis of wet exhaust gases is conducted using a Gasmet Fourier Transform Infrared (FTIR) spectrometer to determine concentrations of H_2O, NO_x, and NH_3. Following initial validations, the authors estimated a maximum uncertainty of approximately 10% on the full scale for the species concentrations. Additionally, an ADEV gas analyzer is employed to measure H_2 (via thermal conductivity) and O_2 (via paramagnetic methods) in a dry exhaust gas sample. The water vapor measurements obtained from the FTIR analyzer are used for wet correction. The accuracy of the H_2 and O_2 analyzers is ±1% within the ranges of 0 to 20% H_2 and 0 to 21% O_2, respectively. For each experimental test, 100 consecutive pressure cycles were analyzed to assess the cycle-to-cycle variability. The proposed model demonstrated a high degree of accuracy in replicating the combustion development and emissions trends across the various steady-state operating conditions outlined in Table 3, specifically for the engine configured with a single spark plug. These specific experimental operating conditions characterize proper and stable engine conditions. Notably, the computed pressure curves aligned well within the measured variability, as shown in Figure 2, the combustion center was estimated with reasonable precision, and the chemical kinetic model effectively captured the acceleration of the combustion process attributed to hydrogen enrichment.[27] For instance, when a hydrogen volume fraction of 5% was compared to pure ammonia fueling, CA0−10 was reduced by 19%, as indicated by both the average experimental and calculated data. This duration decreased by 39% and 41%, respectively, when 15% hydrogen was compared to neat ammonia.[27]

Table 3. Experimental Test Cases Used for Model Validation

	case 1	case 2	case 3
engine speed [rpm]	1500	1500	1500
H_2 content in fuel blend [vol %]	0	5	15
air−fuel equivalence ratio [-]	1	1	1
intake pressure [bar]	1	1	1
intake temperature [K]	323	323	323
spark angle [CADaTDC]	−43.0	−35.0	−27.0
COV_{IMEP} [%]	1.12	0.60	0.39

Additionally, the model successfully predicted unburned ammonia trends, consistent with experimental findings: the introduction of 15% hydrogen resulted in a 22% reduction in ammonia emissions in simulations and a 30% reduction in experiments. Similarly, transitioning from pure ammonia to a 5% hydrogen blend led to a 12% decrease in unburned ammonia in simulations and a 11% decrease in experimental results. The trends in nitrogen oxide (NO) emissions are also reflected in the computed data. Furthermore, the quantities of calculated N_2O are minimal and closely align with the measured values. Furthermore, both the measured and calculated levels of nitrogen dioxide (NO_2) consistently register as zero.[27] These findings suggest that the presented 3D model can also be used to assess the combustion evolution and emissions of the engine considering multiple spark plugs.

The geometry of the engine lacks symmetry, necessitating the modeling of the entire combustion chamber volume. Specifically, a hexahedral mesh, characterized by an average cell volume of 2.5 mm^3, has been constructed, as illustrated in Figure 3. This mesh comprises approximately 220,000 elements at the top dead center (TDC) (Figure 1, left) and around 390,000 elements at the exhaust valve opening (EVO) (Figure 1, right).

Case Studies. In this work, the same steady-state engine operating points reported in Table 3 (cases 1−3) were chosen to assess the engine operation with multiple spark plugs. Up to three spark plugs, as shown in Figure 4, have been considered. The spark plug S1 is placed at the center of the combustion chamber, while the spark plugs S2 and S3 are placed at the intake and exhaust side, respectively. For each case, three different configurations have been simulated: (i) single spark (S1), which represents the same configuration used for the experimental tests, (ii) twin spark (S1 + S2), and (iii) triple spark (S1 + S2 + S3).

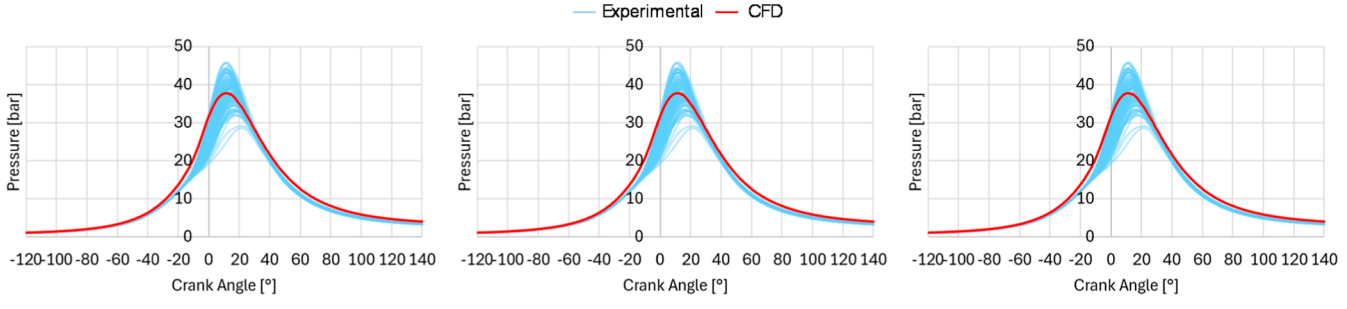


Figure 2. Experimental vs simulated in-cylinder pressure curves. Case 1 (left), case 2 (middle), and case 3 (right).

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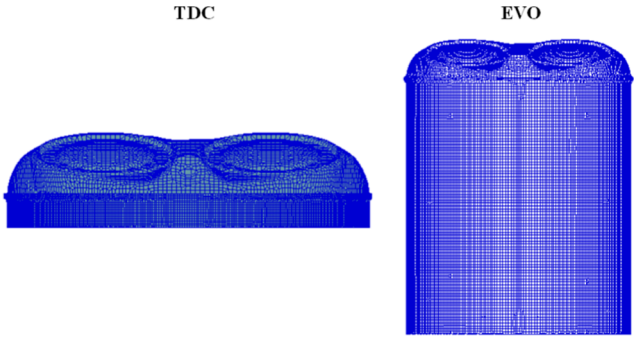


Figure 3. Hexahedral mesh.

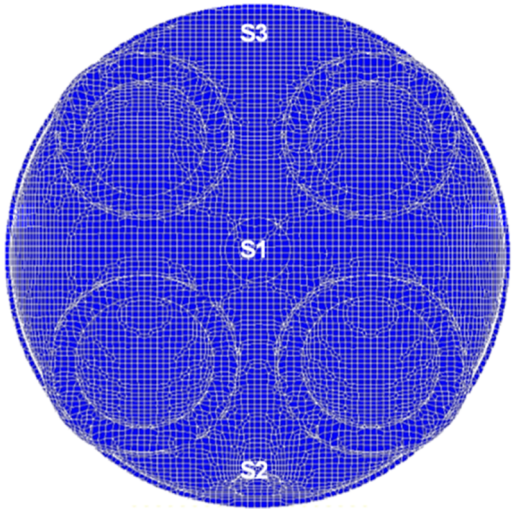


Figure 4. Position of the three spark plugs.

Simulations start at the intake valve closure (IVC), with the assumption of a homogeneous charge within the cylinder. The composition and temperature of the fresh charge have been established based on the measured flow rates entering the cylinder. The initial charge motion has been configured to reflect a singular macrovortex, which emulates the tumble motion typical of this kind of engine. The tumble index, integral length scale, and turbulent intensity have been initialized in accordance with preliminary studies conducted during a motored cycle under constant conditions at the intake manifold. This approach is justified, as dimensionless indices and the integral length scale are primarily influenced by the engine's geometry and exhibit minimal sensitivity to the operating conditions of the engine.[28−30] Additionally, wall temperatures have been assigned constant values derived from the literature pertaining to similar engine types. The initialization parameters for the three cases under investigation are presented in Table 4.

Ignition is simulated by introducing a specific amount of energy, previously utilized for validation in ref 27, quantified at 8.58 W/m^3, into a spherical volume with a diameter of 3 mm. In particular, the ignition energy was used as a tuning parameter to reproduce the in-cylinder pressure trends measured in the experimental tests. This sphere is positioned in accordance with the location of the spark plug electrodes, and the energy release is adjusted based on the ignition duration estimated from experimental tests.[27] For the different multiple spark strategies analyzed, the spark angle was kept constant for each case and equal to the experimental one shown in Table 3.

Table 4. Initial and Boundary Conditions of the Analyzed Cases

	case 1 (neat ammonia)	case 2 (5% of hydrogen by vol.)	case 3 (15% of hydrogen by vol.)
in-cylinder pressure [bar]	1.06	1.06	1.06
in-cylinder temperature [K]	403	396	399
ammonia mass fraction [-]	0.136	0.132	0.123
hydrogen mass fraction [-]	0	0.00081	0.0025
tumble index [-]	0.54	0.54	0.54
head surface temperature [K]	450	450	450
piston surface temperature [K]	493	493	493
liner surface temperature [K]	430	430	430

RESULTS

Combustion Development and Engine Performances.

Figures 5 and 6 compare both the in-cylinder pressure and rate of heat release for each investigated case. With a double spark plug, the pressure peak increases by about 43% considering the engine running with neat ammonia, 28% with 5% of hydrogen by volume, and 18% with hydrogen enrichment equal to 15% by volume. The effect of multiple spark plugs seems to be very sensitive to the amount of hydrogen considered in the fuel mixture, therefore it might be convenient to adopt this strategy with blends with a lower hydrogen enrichment. Figure 6 shows the rate of heat release of each case investigated, and the trends are the same for all the simulated points; furthermore, for all the cases, the in-cylinder temperatures are within feasible values, as shown in Figure 7. Finally, considering the IMEP$_H$ (Table 5), moving from a single to twin spark strategy, it increases by 9% and 11% and 6.5%, respectively, for neat ammonia and 5% and 15% of hydrogen by volume in the fuel mixture. These results confirm that a multiple spark strategy better fits with lower amounts of hydrogen in the fuel mixture.

Referring to the addition of the third spark plug, the improvements are negligible in the case of ammonia−hydrogen blends (cases 2 and 3). In the case of pure ammonia (case 1), there is even a slight reduction in the calculated IMEP$_H$. This is probably due to the nonoptimized position of the third spark plug. In fact, the images reported in Figures 8 and 10 show that the flame front developing in the squish zone located in the exhaust side tends to hinder the development of the front developing in the middle. This effect appears more evident when the engine runs on pure ammonia, where the ignition advance is higher and the development of the fronts is slower (see Figure. 9).

Table 6 presents a comparison related to the calculated combustion characteristics among the different cases investigated. It is worth noting that globally, the adoption of the multiple sparks plugs strategy improves the combustion development leading to a faster combustion process with respect to the case with a single spark plug. As an example, the CA10−90 decreases by about 35%, 22%, and 11% for cases 1−3, respectively, adopting the twin spark strategy; using the third spark plug, it decreases by about 33%, 32%, and 22%, respectively. Figure 11 also clearly depicts that multiple spark plugs lead to an increase in the ammonia consumption curve's slope. Even these results show that the highest improvement is

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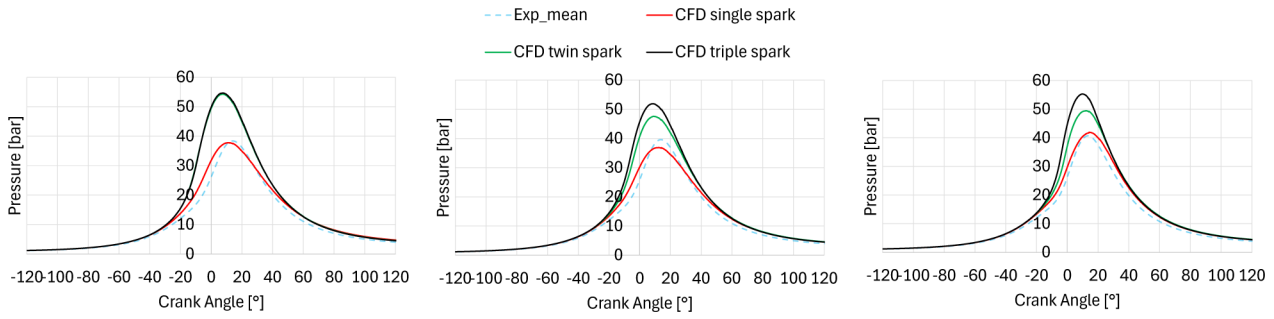


Figure 5. In-cylinder pressure. Case 1 (left), case 2 (middle), and case 3 (right).

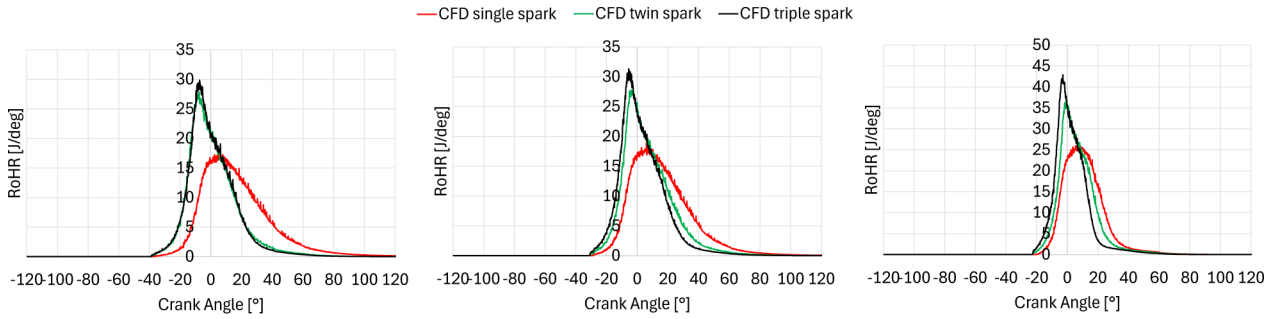


Figure 6. RoHR. Case 1 (left), case 2 (middle), and case 3 (right).

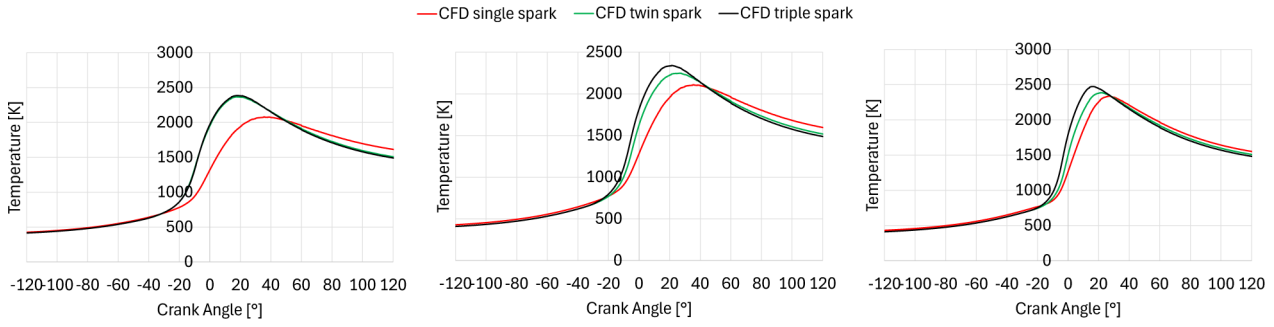


Figure 7. In-cylinder temperature. Case 1 (left), case 2 (middle), and case 3 (right).

Table 5. $IMEP_H$ Comparisons

	case 1 (neat ammonia)		case 2 (5% of hydrogen by vol.)		case 3 (15% of hydrogen by vol.)	
spark strategy	spark angle [CADaTDC]	$IMEP_H$ [bar]	spark angle [CADaTDC]	$IMEP_H$ [bar]	spark angle [CADaTDC]	$IMEP_H$ [bar]
single spark	−43.0	7.94	−35.0	7.76	−27.0	8.25
twin spark	−43.0	8.70	−35.0	8.64	−27.0	8.83
triple spark	−43.0	8.68	−35.0	8.72	−27.0	8.86

related to neat ammonia fueling (case 1), but as also shown above, the higher the amount of hydrogen, the lower is the improvement in the combustion development.

The additional spark plugs also provide an improvement in terms of gross indicated fuel conversion efficiency, as shown in Table 7. Considering neat ammonia (case 1), moving from a single to twin spark, a 9.5% increase in efficiency is obtained. The introduction of a third spark plug leads to a 9.3% improvement, which, while lower than the previous one, is expected due to the corresponding lower increase in $IMEP_H$ (Table 5). For 5% of hydrogen by volume in the fuel mixture (case 2), the efficiency improves by 11% and 12% as the number of spark plugs increases. Conversely, with 15% of hydrogen (case 3), the improvement persists but reduces at around 7% in both configurations. This suggests that employing multiple spark plugs enhances the fuel conversion efficiency, yet the positive impact decreases as the hydrogen concentration in the fuel mixture increases, showing that the optimal scenario occurs with a lower hydrogen amount (case 2). In accordance with the trends obtained in terms of gross efficiency, gross indicated specific fuel consumption (GISFC) shows a progressive reduction for all of the investigated cases when multiple spark plugs are considered, as shown in Table 7. In particular, as also found for the gross efficiency, the maximum improvement is reached with 5% of hydrogen by volume in the fuel mixture (case 2), obtaining a reduction of 10.2% and 10.9% when two and three spark plugs are considered, respectively.

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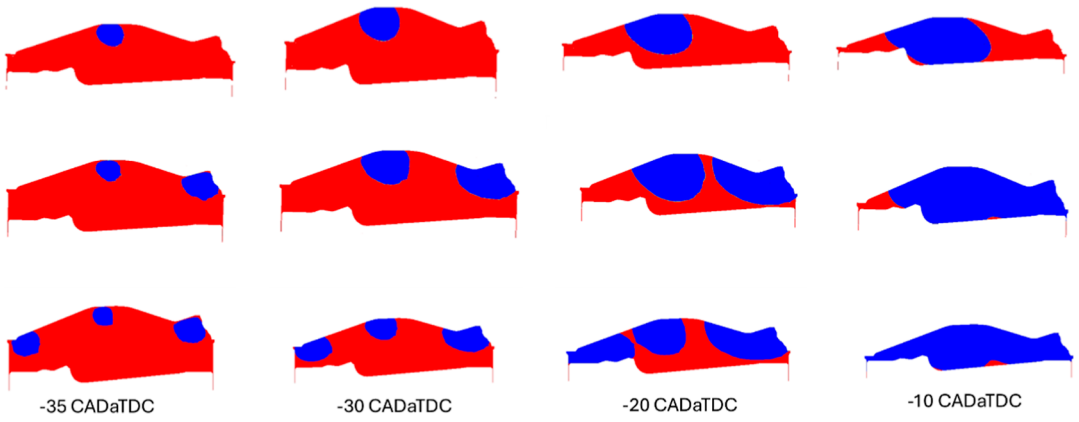


Figure 8. Flame propagation for case 1. Single spark (top), twin spark (middle), and triple spark (bottom).

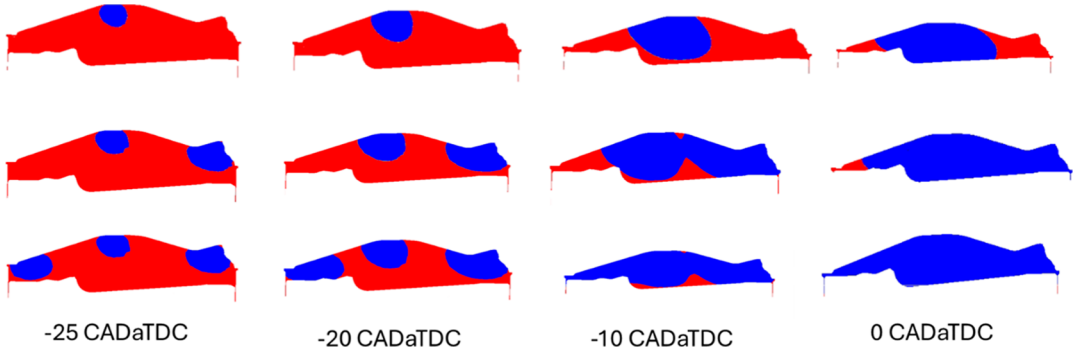


Figure 9. Flame propagation for case 2. Single spark (top), twin spark (middle), and triple spark (bottom).

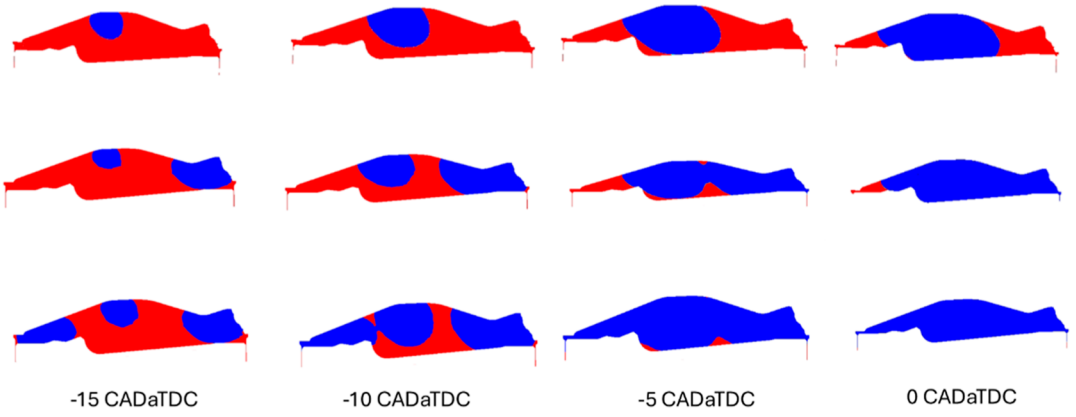


Figure 10. Flame propagation for case 3. Single spark (top), twin spark (middle), and triple spark (bottom).

Table 6. Calculated Combustion Characteristics

	case 1 (neat ammonia)			case 2 (5% of hydrogen by vol.)			case 3 (15% of hydrogen by vol.)		
	single spark	twin spark	triple spark	single spark	twin spark	triple spark	single spark	twin spark	triple spark
CA0−10 [CAD]	36.1	27.2	28.3	29.2	24.9	22.4	21.4	20.4	17.8
CA10−90 [CAD]	52.6	34.3	35.1	46.3	36.1	31.6	28.1	25.1	21.8
CA50 [CAD]	12.5	−2.3	−1.0	12.3	3.4	3.4	9.0	4.2	0.5

EMISSIONS

In this section, a comparison related to the emissions that characterize the investigated points is presented. Considering the main pollutants related to ammonia and ammonia−hydrogen blends, which are unburned NH_3, and NO_x, Table 8 shows a comparison between the results for these pollutants in case of a single, twin, and triple spark strategy, considering wet exhaust gases.

Focusing on unburned ammonia, a reduction in terms of emission is obtained considering the multiple spark strategies. In particular, for each considered fuel mixture, the reduction is about 13%, 8%, and 3%, respectively, for the twin spark and of about 14%, 11%, and 5% for the triple spark. Referring to NO_x

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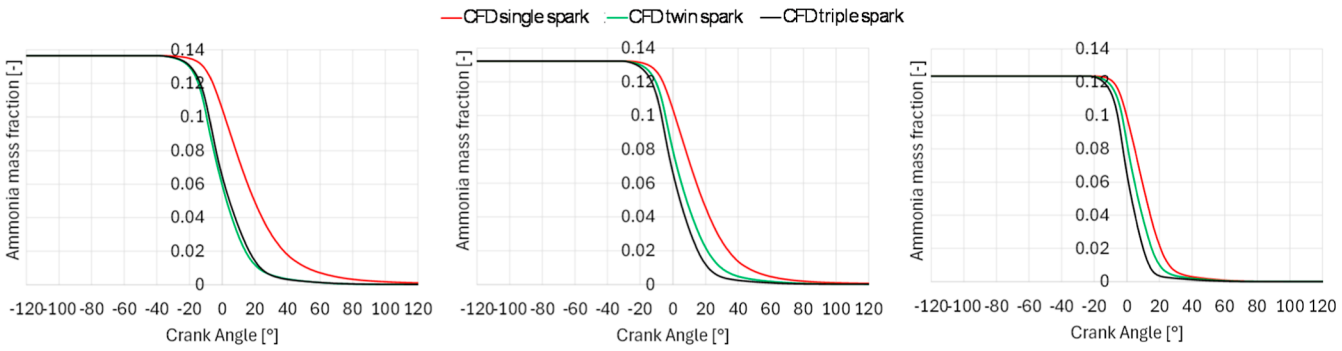


Figure 11. Ammonia consumption for all the investigated cases. Case 1 (left), case 2 (middle), and case 3 (right).

Table 7. Gross Indicated Fuel Conversion Efficiency and GISFC

	gross indicated fuel conversion efficiency [%]			GISFC [g/k W h]		
spark strategy	case 1 (neat ammonia)	case 2 (5% of hydrogen by vol.)	case 3 (15% of hydrogen by vol.)	case 1 (neat ammonia)	case 2 (5% of hydrogen by vol.)	case 3 (15% of hydrogen by vol.)
single spark	37.4	36.7	38.8	511	506	445
twin spark	41	40.8	41.6	467	454	416
triple spark	40.9	41.2	41.7	468	450	414

Table 8. Unburned Fuel and NO_x Concentrations in the Wet Exhaust Gas

	NH_3 [ppm]			NO [ppm]			N_2O [ppm]		
	single spark	twin spark	triple spark	single spark	twin spark	triple spark	single spark	twin spark	triple spark
case 1 (neat ammonia)	6249	5427	5401	1445	1443	1471	12	5	3
case 2 (5% of hydrogen by vol.)	5464	5020	4890	1569	1513	1557	10	5	3
case 3 (15% of hydrogen by vol.)	4885	4721	4665	1732	1675	1667	4	2	1

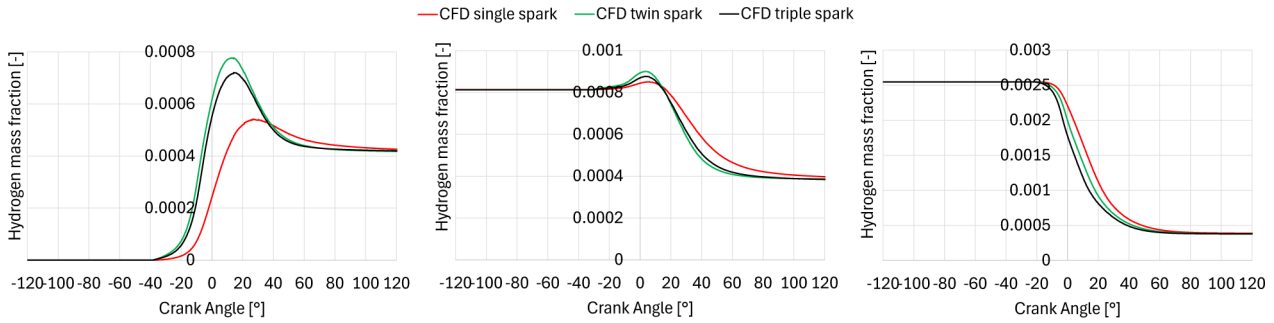


Figure 12. Hydrogen consumption for all the investigated cases. Case 1 (left), case 2 (middle), and case 3 (right).

emissions, an improvement is also reached even if only a slight reduction in terms of NO emissions, which decrease by a maximum of 4%, is obtained. This can be explained by the higher temperature peak reached in the combustion chamber. In fact, considering the multiple spark strategies, the temperature peak in the combustion chamber increases moving from a single spark to a triple spark (Figure 7), affecting the NO_x formation due to the oxidation of the N_2 present in the combustion chamber. Finally, considering both the unburned ammonia trend and temperature trend, it is worth noting that the addition of multiple spark plugs leads to an increase in the ammonia oxidation curve's slope connected to an increase in the combustion chamber temperature. Since the ammonia consumption rate increases, leading to a reduction of the unburned ammonia, the amount of fuel NO_x in the chamber reduces, due to a lower concentration of fuel nitrogen, balancing the bigger thermal NO_x production. This could explain why the differences in terms of NO_x emissions are very small, moving from single to multiple spark configurations.

Another aspect of interest is the amount of hydrogen which remains in the exhaust gas because, for the different fuel blends considered, hydrogen is also a subproduct of ammonia dissociation, as already shown in ref 27. In Figure 12, the consumption of hydrogen, for all the investigated fuel blends considering the three different spark strategies, is presented. From the results obtained for Case 1 (neat ammonia), it is clear that hydrogen is directly generated by ammonia dissociation, in fact it is not present in the combustion chamber before the start of combustion. It is generated during the combustion process and then it is partially consumed. This latter phenomenon occurs in all the investigated cases and, as for ammonia, also hydrogen consumption benefits using the multiple sparks plugs strategy. Similarly to the trends for ammonia, even for hydrogen, the slope of the consumption curves is bigger with respect to the single spark strategy and the amount of hydrogen in the exhaust

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Energy Fuels XXXX, XXX, XXX−XXX

gas reduces by about 1% (twin spark) and 0.5% (triple spark) for neat ammonia (case 1), by about 1.43% (twin spark) and 2.1% (triple spark) considering 5% of hydrogen by volume (case 2), and by about 1.5% (twin spark) and 1.7% (triple spark) when the hydrogen enrichment is equal to 15% (case 3).

CONCLUSIONS

In this work, by means of a validated 3D model, the effects of hydrogen enrichment coupled with the use of multiple spark plugs on the performances and emissions of a light-duty spark-ignition ammonia engine have been assessed. The numerical analysis carried out in this study presents qualitative yet reliable results that allow us to estimate the effect of additional spark plugs in the analyzed engine. The main outcomes are summarized as follows.

- Using a twin spark strategy, the pressure peak increases by about 43% considering the engine running with neat ammonia, 28% with 5% of hydrogen by volume, and 18% with hydrogen enrichment equal to 15% by volume in the fuel blend. The $IMEP_H$, moving from a single to twin spark strategy, increases by 9% and 11% and 6.5%, respectively, for neat ammonia and 5% and 15% of hydrogen by volume in the fuel mixture. These results show that it might be convenient to adopt this strategy with blends with a lower hydrogen enrichment.
- The addition of a third spark plug leads to negligible improvements in combustion development in the case of ammonia−hydrogen blends. In the case of pure ammonia fueling, there is even a slight reduction in the calculated $IMEP_H$, which is probably due to the nonoptimized position of the third spark plug.
- The use of multiple spark plugs improves the combustion development leading to a faster combustion process. Adopting the twin spark strategy, the CA10−90 decreases from about 35% to 11% moving from neat ammonia to a hydrogen enrichment of 15% by volume and from about 33% to 22% using the third spark plug.
- The multiple spark strategies lead to an improvement in terms of gross indicated fuel conversion efficiency. The maximum gain is reached with 5% of hydrogen by volume in the fuel mixture, reaching an increase in gross efficiency of about 11% (twin spark) and 12% (triple spark), corresponding to a reduction of GISFC equal to 10.2% and 10.9%, respectively.
- The emissions of unburned ammonia decrease by about 13%, 8%, and 3%, respectively, for the twin spark and by about 14%, 11%, and 5% for the triple spark, moving from neat ammonia fueling to 15% of hydrogen by volume in the fuel blend.
- Considering multiple spark strategies, NO_x emissions decrease even if only a slight reduction in terms of NO emissions (maximum of 4%) is obtained.

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Notes

The authors declare no competing financial interest.

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NOMENCLATURE

3D	three-dimensional
aTDC	after top dead center
CAD	crank angle degree
CA0−10	0−10% mass fraction fuel burn duration
CA10−90	10−90% mass fraction fuel burn duration
CA50	crank angle at 50% mass burning fraction
CFD	computational fluid dynamics
COV	coefficient of variation
EVO	exhaust valve opening
GDI	gasoline direct injection
GISFC	gross indicated specific fuel consumption
IVC	intake valve closing
IMEP	indicated mean effective pressure
IMEPH	gross indicated mean effective pressure
RANS	Reynolds-Averaged Navier−Stokes
RoHR	rate of heat release
TDC	top dead center

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Energy Fuels XXXX, XXX, XXX−XXX