Experimental study of RCCI engine – Ammonia combustion with diesel pilot injection

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Abstract

Ammonia is seen as one potential carbon-free fuel, especially for maritime applications. Since SI engines require a significant ignition energy for large cylinders, engine manufacturers are targeting the use of ammonia in Compressed Ignition (CI) engines. Because of ammonia's high auto-ignition temperature, to ensure that the combustion occurs in a CI engine, a pilot injection of a higher reactivity fuel must be used, as in Reactivity Controlled Compression Ignition engines. In the present study, the objective was to provide first unique data about the efficiency and pollutant emissions for a single cylinder compression ignition engine with a diesel energy fraction as minimum as possible (down to less than 2%) at a constant 1000 rpm. Experiments cover the impact of a wide variation of equivalence ratios of NH_3-air mixtures from ultra-lean to slightly rich conditions. CO_2, CO, NH_3, NO_X, N_2O, UHC values were measured with a Fourier Transform Infrared (FTIR) spectrometer. Results of CO_2 and N_2O are presented as CO_2-Equivalent (CO_{2eq}) impact. Combustion stability was achieved for most conditions but not for the leanest ones. Furthermore, under lean conditions for a similar ammonia content, the minimum CO_{2eq} is reached with a slightly higher Diesel Energy Fraction than the minimum possible. Finally, both leanest and richest conditions present a higher level of CO_{2eq} compared to the range of ammonia/air mixtures at stoichiometry or just below.

Introduction

Since a few years ago, ammonia has been recognised as a promising chemical storage solution with the potential of delivering green hydrogen over long distances and for heavy loads [1], especially to store and flexibly distribute energy obtained from renewable sources such as wind, solar and marine [2]. In 2018, the International Maritime Organization's Initial Strategy for Green House Gases (GHG) reduction, pointed out that in addition of being a viable 'storage molecule', ammonia will be a fuel necessary to decarbonize the maritime sector by 2050. However, several challenges still remain for the use of ammonia as a fuelling vector. Out of all the main challenges recognised in the transition of a hydrogen-ammonia economy [3], public perception, adequate legislations and a better knowledge on environmental impacts are probably the most complex to deal with. Angeles et al. [4] concluded that internal combustion engine vehicles using biomass-based ammonia could deliver lower carbon footprint profiles than diesel or gasoline fuel. Similarly, Bicer and Dincer [5] analysed various city transportation and power generation systems fuelled with ammonia from nuclear plants. The results indicated that a considerable decrease in GHG emissions could be achieved with a non-negligible impact on global warming potential. However, the results also demonstrated that with the reduction of carbon footprint the increase in nitrogen-based species (ie. NO_X) could be considerably high, shadowing the overall balance in GHG mitigation. Finally, a recent report produced by the Maritime Cleantech Cluster [6] stated that the use of liquid ammonia in internal combustion engines for maritime purposes can lead to 'net zero emissions'. However, careful considerations need to be taken to avoid large GHG emissions during the combustion process itself. The report raises the importance of fully evaluating some of the most environmentally damaging molecules of ammonia combustion, as N_2O, which requires further studies. Thus, it was concluded that ammonia is a free-carbon fuel but not necessarily a 'zero-emission' combustion option. Since Spark Ignition (SI) engines using ammonia require a significant ignition kernel [7] for a large cylinder like the case of X40DF-1.0 [8], most marine engine manufacturers announced that ammonia will be used as a fuel in Compression

Ignition (CI) engines. Due to the high auto-ignition temperature and the low reactivity of ammonia, dual-fuel concepts seem to be more adapted for this application as investigated in most recent studies [9-11]. However, in order to achieve zero carbon emissions, the amount of diesel has to be drastically reduced. As highlighted in [12-13], the Reactivity Controlled Compression Ignition Combustion (RCCI) concept could perhaps mitigate these issues. This technology relies on the ignition of a low-reactivity premixture, in the case of ammonia/air mixture, with a higher reactivity pilot fuel such as dodecane. Only two recent studies are focused on this ignition of ammonia engine by diesel pilot injection [14, 15]. Therefore, the objective of the present study is to evaluate the potential of RCCI for an ammonia engine in terms of stability and performances, i.e. to use a diesel energy fraction as low as possible - below 2% for certain operating conditions - and to characterize the exhaust emissions. Experimental measurements cover the impact of a wide variation of the premixed equivalence ratio (ultra-lean to rich premixed ammonia-air mixtures) and of the Diesel Energy Fraction (DEF) (maximum 12.5%) on the performances and emissions in a single cylinder 0.5L engine. Special attention will be given to N_2O due to its large global warming effect: at equivalent mass, N_2O has a global warming potential 265 times more potent than CO_2, as recently indicated in the 5th assessment of Intergovernmental Panel on Climate Change (IPCC) [16]. For the first time, some quantitative values about the global warming impact of this type of engine without any exhaust post-treatment system are also provided.

Experimental set-up and operating conditions

Engine tests were conducted on a four-cylinder Compression Ignition PSA DW10 light duty engine converted into a 0.5L single cylinder engine. All experimental set-up details are shown in Fig. 1, with more details specified in Table.1. The engine speed was maintained constant at 1000 rpm. The intake and exhaust temperature and pressure were acquired by K thermocouples and resistive absolute pressure transducers, respectively. The in-cylinder pressure from a relative quartz pressure transducer Kistler 6045A was acquired via a LabView data acquisition system, with a resolution of 0.1 CAD, provided by a Kubler optical encoder.
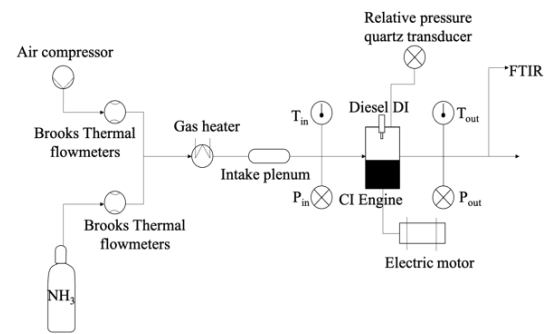

Fig. 1. Global view of the experimental set-up.

Ammonia, the 'low reactive' fuel, was stored in liquid form in a tank and heated to be evaporated before entering inside the intake plenum to be mixed with air. Ammonia and air flow rates were controlled by two Brooks thermal mass flowmeters. The premixture was heated to be admitted at a desired intake temperature (T_{in}=80°C). For each operating condition, 200 consecutives cycles of pressure data are recorded to analyze cycle-to-cycle variations and compute averaged pressure values. From the average pressure data, as a function of the crank-angle degree (each 0.1 CAD), the apparent fuel-combustion heat release is computed from the first law of thermodynamics, first by considering a constant γ value as,

$$\frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

With γ being the heat capacity ratio, P the in-cylinder pressure, V the in-cylinder volume and θ the crank angle degree. Then, the average in-cylinder temperature as a function of the crank-angle degree is estimated. By using the Cp function of the temperature, Cv and γ are then computed each 0.1 CAD.

Table 1. Engine specifications.

Number of cylinders	1
Bore x Stroke (mm x mm)	85 x 88
Conn. Rod length (mm)	141
Displacement (l)	0.5
Compression Ratio	16.4:1
Bowl type	Re-entrant
Valves per cylinder (Int./Exh.)	(2/2)
Engine speed (rpm)	1000
Intake valve opening/closing	344ATDC/ 117 BTDC
Exhaust valve opening/closing	134/396 ATDC
Swirl ratio (50 CAD BTDC)	2.0
Port intake pressure (bar)	1
Port intake temperature (°C)	80

Then the heat release rate is calculated a second time with this variable γ. The Indicated Mean Effective Power (IMEP) was evaluated by integrated the measured pressure data with the volume variation. The blowby and the internal recirculation of gases were not considered here.

Exhaust emissions measurement

A fast Fourier Transformed InfraRed (FTIR) DX2000 spectrometer was used to measure the exhaust gases concentration of the following species H_2O, CO_2, CO, NH_3, NO_X, N_2O, UHC. Special attention was given to the accuracy of N_2O measurement at the exhaust especially at low concentrations (lower than 50 ppm). For that, a calibrated bottle of N_2O/N_2 was used to evaluate the interferences with other gases, for example CO_2. Thus, the optimized wavelength range 2,300-2,450 cm^{-1} for N_2O and 2,100-2,350 cm^{-1} for CO_2 was selected and validated for a mixture of $CO_2/N_2O/N_2$. During experiments, the exhaust gases samples were heated up to 190°C or above before being analyzed by the FTIR software. Each sample was collected and analyzed every five seconds. For each condition, 20 consecutives samples were acquired after stabilization. For N_2O calculations, among these samples, only those with acceptable residuals were considered. Standard deviations were computed but only N_2O's are presented in this paper due to low standard deviation values for other measurements.

Ammonia and Diesel amount conditions

The 'high reactivity' fuel, diesel, was directly injected inside the cylinder with a CRI Bosch injector. The pressure of injection was maintained at 200 bar to ensure sufficient flexibility on the injection duration to adjust to a as low quantity as possible. The characterization of the injected diesel mass was done as a function of different Duration of Injections (DOI). A second order fit was used afterwards to know the mass injected in each case of the experiment. At each ammonia/air premixture condition and for each DOI, first using higher DOI than DOI$_{min}$, the Start Of Injection (SOI) parameter was carefully chosen. This was done to ensure and optimize as much as possible the stability of the combustion, therefore minimizing the variation of the IMEP. After this point, the DOI was carefully reduced until a duration where it can no longer be reduced without affecting the engine performance - DOI$_{min}$. Three other different DOI were investigated to experiment the effect of the diesel energy fraction on both engine performances and exhaust emissions: the DOI$_{min}$ (ie. corresponding to the minimum amount of diesel injected that ensures stable combustion), DOI$_{min}$ + 50, 100 and 150 μs. For the leanest ammonia/air premixture conditions, the dead-zone of the injector imposed a DOI$_{min}$ of 400μs. The DOI values used as a function of the

ammonia/air equivalence ratio are shown in Figure 2. This DOI$_{min}$ starts to increase only after an equivalence ratio $\phi_{premix} > 0.9$ due to the greater diesel demand to ensure a stable combustion for each cycle. From calibration, the Diesel Energy Fraction (DEF) was determined as below:

$$DEF = \frac{\dot{m}_{diesel} \cdot LHV_{diesel}}{\dot{m}_{diesel} \cdot LHV_{diesel} + \dot{m}_{NH_3} \cdot LHV_{NH_3}}$$

Results

Combustion development and characteristics

As it can be seen in Figure 2, DOIs of diesel were set in order to ensure maximum engine stability as much as possible. A minimum of 2% of DEF was possible only for NH_3/air at equivalence ratios between 0.8-0.9. As expected, due to the low flammability range for ammonia, it was necessary to increase the diesel injection amount up to 12.5% for the leanest mixtures for an ultra-lean equivalence ratio (0.25).
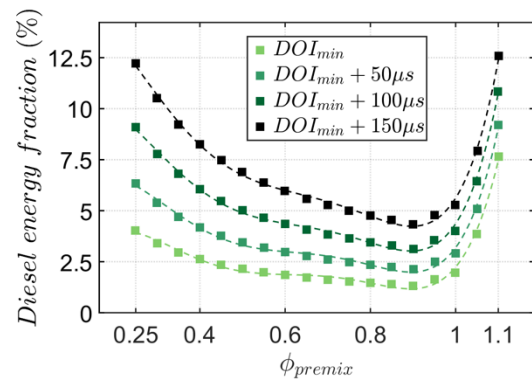

Fig. 2. Different DEF corresponding to the different DOI as a function of ϕ_{premix}

In Figure 3, the Indicated Mean Effective Pressure (IMEP) was plotted for all conditions with their corresponding variability (COV$_{IMEP}$). From 0.8 to 1.1 of ϕ_{premix}, a maximum of IMEP is obtained with a very good engine stability. When the duration of diesel injection is slightly increased, it was possible to ignite leaner ammonia/air mixture.
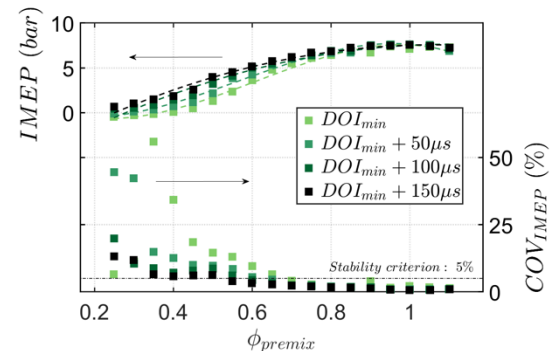

Fig. 3. Effect of Diesel amount on IMEP as a function of ϕ_{premix}

Figure 4.a highlights the combustion delay as a function of the equivalence ratio of the ammonia/air mixture when the minimum diesel amount to ensure ignition of the premixed charge is injected. Moreover, the richer ϕ_{premix}, the greater the heat release rate. These results agree with those of Wüthrich et al. [13]. In Fig. 4.b, the estimated averaged thermodynamic temperature along the cycle is plotted, indicating how this temperature could be high when a stoichiometric mixture is used. Yet, lean conditions induce a temperature peak below 1000 K due to very low heat release rates, therefore a poor ammonia combustion efficiency.
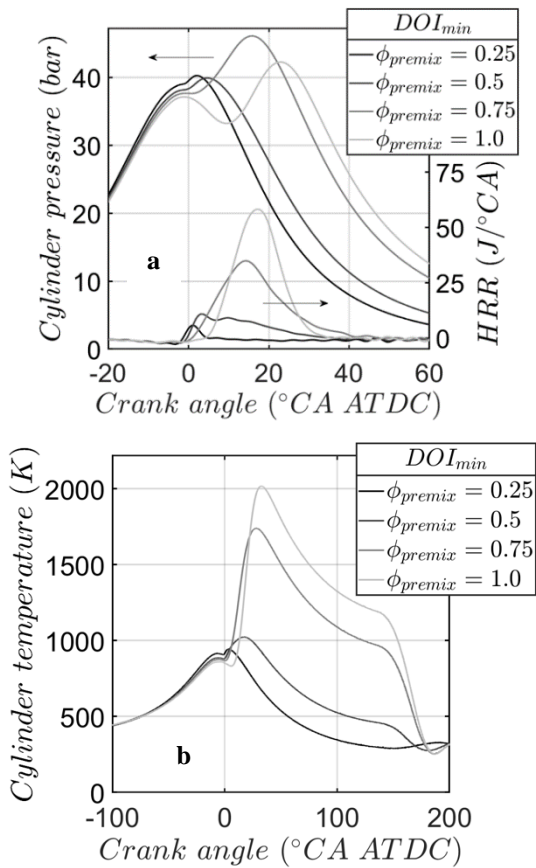

Fig. 4. **(a)** In-cylinder pressure and heat release rate and **(b)** average thermodynamic temperature evolution as a function of crank angle degree for different air/ammonia mixtures with minimum diesel content necessary to guarantee ignition and combustion stability.

Last, an example of the effect of diesel amount on the combustion development is shown in Fig. 5 for the equivalence ratio 0.9. The different durations of diesel injection induce different amounts of diesel energy fraction (1.5, 2.5, 3.5 and 5%). When the diesel content is not minimum, the shape of the heat release rate and pressure does not really change with the increase of diesel, affecting only the maximum reached. The temperature profile itself is similar

except for DOI_{min}. It is worth noting that with this strategy, a longer combustion duration relates to the visibly longer heat release rate during the cycle, as it can be seen in Fig. 5.a. A shift of the temperature profile is noticed as compared to other DOI strategies and both pressure and heat release rate profiles are reduced using DOI_{min}.
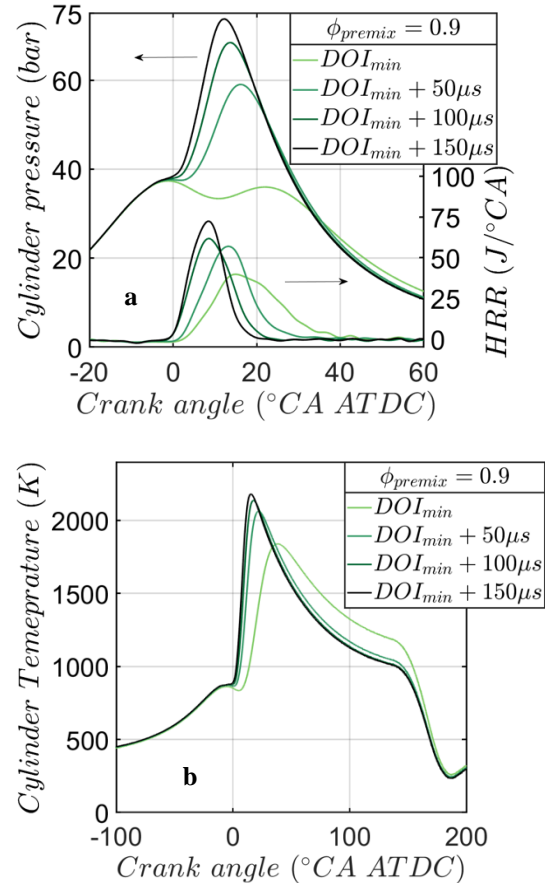

Fig.5. **(a)** In-cylinder pressure (the left axis) and heat release rate (the right axis) and **(b)** as a function of crank angle degree for different DOI at ϕ_{premix} =0.9

Exhaust emissions : unburnt species and CO_2-CO
Unburned ammonia emissions are a real issue due to the environmental impact if no after-treatment is added [3]. In Fig. 6, one can notice that the maximum unburnt ammonia level is obtained at ultra-lean conditions, and this maximum peak decreases with the increase of diesel content due to the combustion improvement caused by a longer diesel injection that induces more stratification in the cylinder (stratification in total equivalence ratio and local diesel/ammonia ratio). Moreover, when going to the leanest ammonia/air mixtures, the unburnt ammonia emissions decrease strongly as ammonia combustion is optimized due to the relative importance of diesel energy fractions.
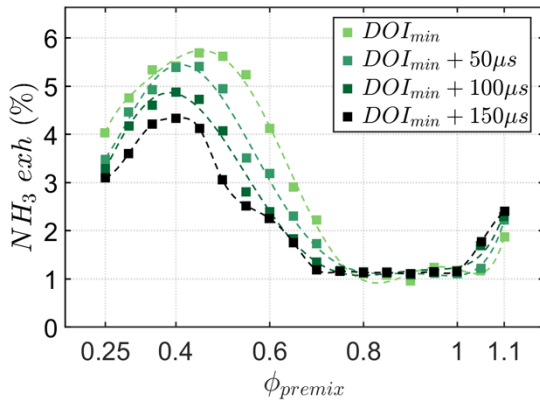
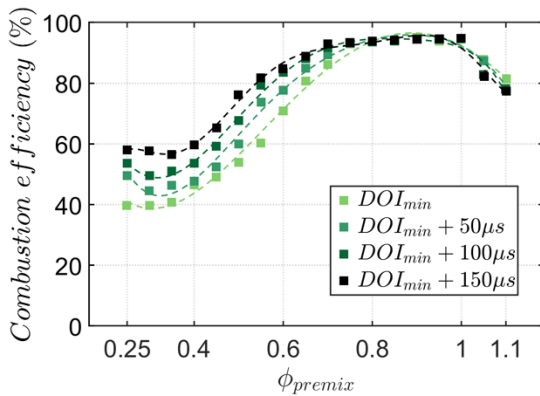**Fig.6.** emissions evolution of unburnt ammonia for all conditions.

Fig.7. Ammonia combustion efficiency as a function of equivalence ratio for all conditions

Longer DOI might increase the local turbulence thus improving combustion. Besides, the longer DOI strategies, the greater number of high reactivity zones are present inside the chamber and located at different zones creating more ignition spots to enhance ammonia combustion. This could be explained by the dynamics of the diesel spray and the entrainment of the fresh ammonia/air mixtures. From 0.7 to 1.0 of ammonia/air equivalence ratio, NH_3 exhaust emissions reached a minimum, without any impact of the change in diesel content. This level remains relatively important (1%) and is due to the design of the piston bowl as indicated in [17, 18] but not to the ignition of the amount of injected diesel. This trend is highlighted in Figure 7, where ammonia combustion efficiency reaches a peak around 95% at these conditions and neither a slight change in equivalence ratio nor a longer DOI impact the combustion efficiency of ammonia. In Fig. 6, as expected, under slightly rich condition, emissions increase due to the excess of fuel and imply a decrease of ammonia combustion efficiency present in Fig. 7. Another important aspect is that the decrease of ammonia at the exhaust at ultra-lean ammonia/air premixture from 0.5 to 0.25 ϕ can be

explained by the quasi constant combustion efficiency of ammonia for the same DOI strategy. On one hand, the increase of ammonia content in the mixture towards the flammability limit increases the ammonia combustion efficiency. On the other hand, adding ammonia in the mixture increases the inhibiting effect of ammonia on diesel. It is worth noting that the second effect, the non-intuitive one, can be highlighted in Fig. 7 and Fig. 8.b. In Fig. 7, for a specific and ultra-lean ϕ_{premix}, increasing the diesel amount l increases the combustion efficiency of ammonia as well as of the. In fact, in Fig. 8.b, at these conditions, UHC ratio decreases by increasing the diesel amount, indicating the greater the diesel amount is injected, the less the negative ammonia impact. Therefore, the better diesel burns, the better the ammonia mixture burns.
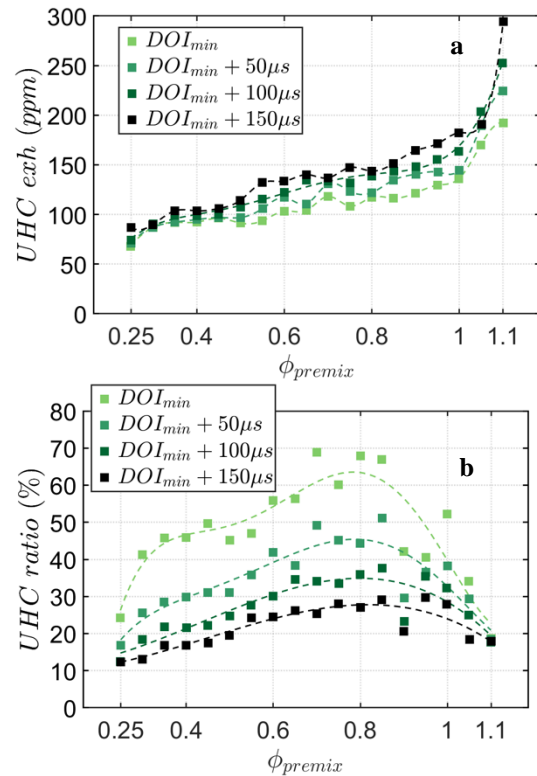

Fig.8. Evolution of UHC emissions **(a)** and UHC ratio **(b)** (exhaust UHC (mass)/diesel amount (mass))

By considering Fig. 8.a, it is worth to notice that Unburnt Hydro-Carbons (UHC) levels increase with both ammonia and diesel content. To better visualize these effects, the ratio between UHC at the exhaust (in mass) and the mass of diesel injected at the intake is estimated (by assuming UHC at a similar molecular weight than diesel), Fig. 8.b. A peak of UHC (i.e., less efficient combustion of diesel) occurs at ϕ_{premix}=0.8, whatever the fraction of diesel, but the combustion of diesel is improved when increasing DOI. Above this ϕ_{premix}=0.8, UHC ratio

decreases with ammonia content and strongly with diesel content. It is seen that the UHC ratio strongly decreases after the stoichiometric value mostly due to the minimum amount of diesel required to ensure a good combustion. Therefore, it can be concluded that the combustion of diesel in an ammonia/air mixture under engine operating conditions is improved when the amount of diesel is not set at the possible minimum. In fact, ammonia has an inhibiting effect on diesel combustion especially in the case of a small diesel amount. This was observed during the experiments near the minimum diesel fraction needed to burn the premixed charge. Just below this limit ammonia inhibits diesel ignition and increases auto-ignition delay as concluded in [19] with n-heptane/ammonia blend.
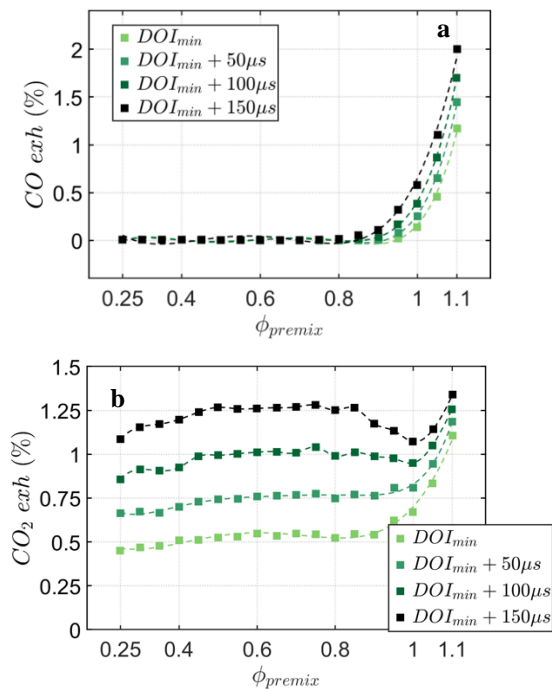

Fig.9. evolution of **(a)** CO and **(b)** CO_2 emissions for all conditions.

In Fig. 9.a, it can be noted also that for the same threshold value, ϕ_{premix}=0.8, CO emissions change. Below this value, no CO emissions were observed, and above this value, greater levels of CO are emitted, despite the constant or even higher diesel amount introduced, compared to leaner ϕ_{premix} cases. Meanwhile, for ϕ_{premix}>0.9 the CO levels increase very rapidly and reach higher levels than CO_2 at the exhaust. By increasing the diesel content at the input, the CO above ϕ_{premix}=0.9 is greater than for poorer diesel content due to less oxygen available for the greater diesel content. This evolution of CO between 0.8 and 1.0 is related to the uncomplete oxidation of diesel which leads to a decrease of CO_2 at the same conditions. Except from those

conditions, CO_2 levels are greater for longer DOI strategies as expected.

Exhaust emissions : NO_x species
As previously measured in [17, 18] with the same engine but ignited by a spark, NO_X emissions, Fig. 10, reach a maximum level near an equivalence ratio of 0.8. Adding diesel changes only very slightly the equivalence ratio corresponding to the maximum; more diesel induced a peak of NO_X for leaner conditions but with a slight decrease of the peak. For example, at ϕ_{premix}=0.9, for DOI_{min} the NO_X emission is 4,211 ppm for a peak average temperature of 1,839 K, while for a greater diesel content with DOI_{min}+150μs NO_X levels are at 3506 ppm for a peak average temperature of 2,180 K. But for all cases at ϕ_{premix} = 1.1, the level remains at 1,000 ppm, contrary to previous studies where zero NO_x emissions are obtained, certainly due to the pilot diesel injection. Figure 11.a confirms that NO_2 remains negligible - almost zero above ϕ_{premix}=0.6 and below 50ppm for ultra-lean conditions, with a positive effect on the diesel amount. Adding diesel will decrease even more these levels mostly due to higher temperature reached compared to DOI_{min}. In fact, when NO_2 is plotted as a function of the maximum average thermodynamic temperature, Fig. 11.b, the effect evidences that NO_2 emissions increase when this maximum is low due to the lean ammonia/air mixture.
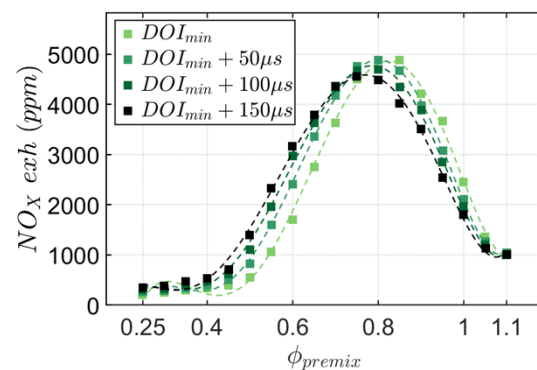

Fig.10. evolution of NO_X emissions for all conditions.

N_2O emissions
For all operating conditions, N_2O levels remain below 150 ppm but present two maximum peaks, one at ultra-lean conditions (ϕ_{premix}=0.45), and a lower one around stoichiometry, with a minimum reached at slightly rich conditions (ϕ_{premix}=1.1). This specific trend of N_2O emissions as a function of ammonia/air mixture is in good agreement with that obtained in a swirl burner [21]. The increase of diesel amount for a same ϕ_{premix} decreases the levels of N_2O at lean conditions, while for DOI_{min} the standard deviation indicates less stability.
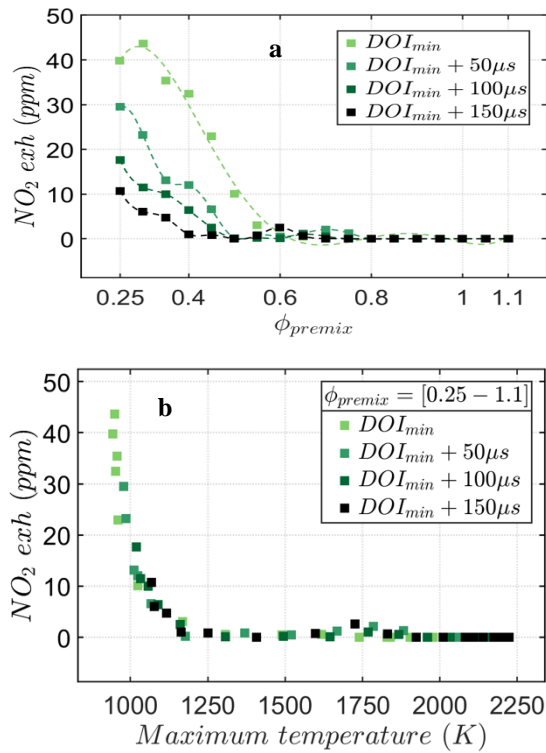
Fig.11. Evolution of NO_2 emissions as a function of equivalence ratio **(a)** and maximum average in-cylinder temperature **(b)** for all conditions.
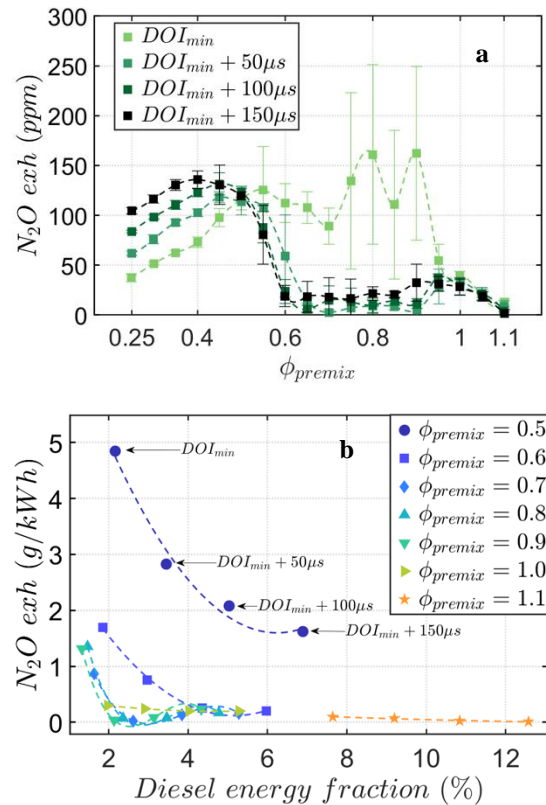

Fig.12. Evolution of N_2O emissions as a function of ammonia/air equivalence ratio **(a)** and diesel energy fraction **(b).**

Figure 12.b illustrates N_2O emissions (g/kWh) at the exhaust from ϕ_{premix}=0.5 to ϕ_{premix}=1.1. First, the lower the indicated power due to lean ammonia/air mixtures, the higher the levels of N_2O. For each premixed equivalence ratio, the longer the DOI, the higher the DEF. It seems that to reduce N_2O as much as possible a good strategy is to run at slightly rich conditions (ϕ_{premix}=1.1) or lean conditions (ϕ_{premix}= 0.7-0.9), as ultra-lean and near stoichiometry cases increase significantly N_2O at the exhaust. For lean conditions (ϕ_{premix}= 0.7-0.9), it seems that to limit the diesel energy fractions (below 2%) to reduce N_2O as much as possible is not an optimized strategy.

To understand better the key parameters that drive N_2O emissions, the impact of both the combustion duration and the maximum average in-cylinder temperature on N_2O emissions is provided for 3 premixed equivalence ratio ranges: ultra-lean conditions (a) ϕ_{premix}= 0.25-0.5, lean conditions (b) ϕ_{premix}= 0.55-0.8, near-stoichiometric conditions (c) ϕ_{premix}= 0.85-1.1, as ranked in Fig. 13. First, the decrease of the peak average in-cylinder temperature appears when the combustion duration is longer.

At ultra-lean conditions, it might be emphasized that except from the lowest Temperature Peak (T_{peak}<1,100 K), N_2O emissions are only affected by the combustion duration since levels rise constantly with longer combustion durations.

Under the lowest T_{peak}, it has to be added that all low levels of N_2O emissions (<100 ppm) are obtained with only less than half of the ammonia input being correctly burned. For lean conditions, no distinction can be done between the impact of combustion duration and the peak of in-cylinder temperature due to their inter-link. However, a short combustion duration and a high peak temperature are required to ensure the lowest levels of N_2O. It is also interesting to notice that the highest variations tend to increasingly appear during longer combustion duration and lower T_{peak}, except at DOI_{min} where the standard deviation is considerable. Finally, at near-stoichiometric conditions, both factors have a huge impact on N_2O emissions since with shorter combustion duration than 13°CAD and T_{peak}>2,000 K, the levels of N_2O are constant under 50 ppm. It is also observed that a narrow window of T_{peak} (2,050 K<T_{peak}< 2,100 K) seams to minimize N_2O below 20 ppm at the exhaust. For these operating conditions, combustion durations vary a lot (between 14.3 CAD and 9.2 CAD) indicating that no optimized range for N_2O under 20 ppm is observed for this combustion duration.
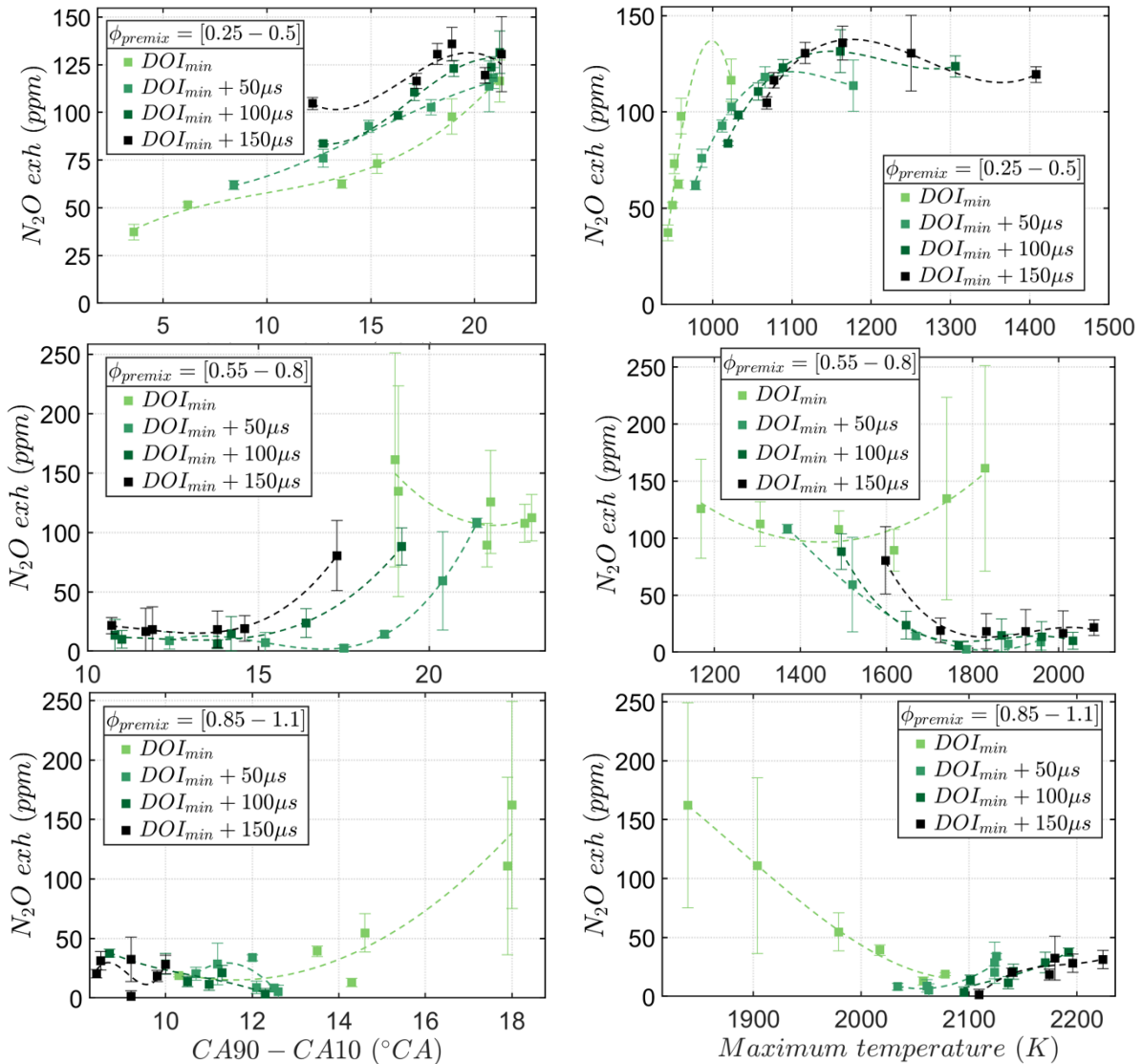
Fig.13. N₂O emissions as a function of combustion duration and maximum averaged in-cylinder temperature for ultra-lean (**a**), lean (**b**) and near stoichiometry conditions (**c**)

Discussion. Global warming impact for RCCI ammonia engine

Since 1 kg of N_2O has been assessed in [16] to be equivalent as 265 kg of CO_2, an equivalent CO_2 (g/kWh) was computed as below:

$$CO_{2\,eq} = CO_2 + 265 \times N_2O$$

for conditions from ϕ_{premix}= 0.5 to ϕ_{premix}= 1.1.

In Fig. 14.a, it can be observed that equivalent CO_2 emissions are minimized for lean conditions, between ϕ_{premix}=0.6 and ϕ_{premix}=1.0, except for $DOI_{min.}$ The best DOI strategy seems to be DOI_{min}+50μs ~ 2.5% of diesel energy fraction. By operating at stoichiometric or slightly rich conditions, the increase of GHG levels is mainly due to the increase of diesel content to ensure a stable combustion. To operate at leaner conditions increases drastically the level of GHG. Finally, it can be noticed that GHG levels can be reduced by ten times compared to a regular light duty diesel engine assessed in [20]. This can be accomplished by optimizing the ammonia/air equivalence ratio and diesel energy content. This is highlighted in Fig. 14.b, where the same results are plotted as a function of Diesel Energy Fraction.

Contrary to what might be expected at first sight, the increase of diesel content does not directly increase the GHG equivalent as N_2O share in GHG at the exhaust is not neglectable despite their low levels. It is clear that for this engine and operating conditions, to minimize the GHG, the premixed equivalence ratio has to be between ϕ_{premix}= 0.7 and ϕ_{premix}= 0.9 since these values are the only ones reaching under 100g/kWh of GHG levels.
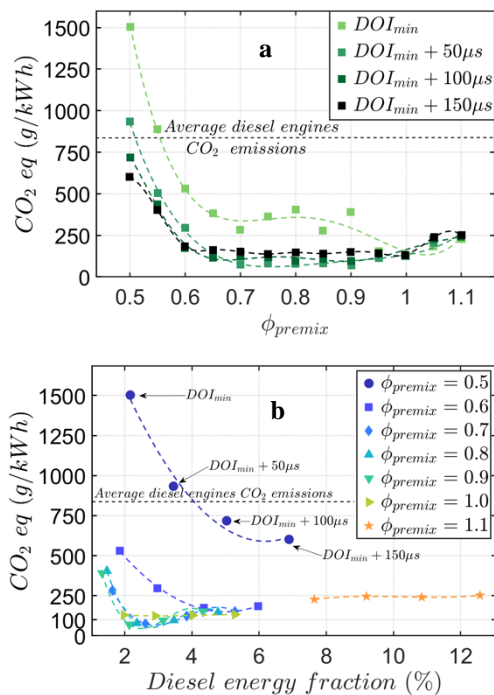
Fig.14. CO_2 equivalent emissions as a function of ϕ_{premix} **(a)** and diesel energy fraction **(b)**.

But even by choosing a lean case, it is observed that reducing as much as possible the diesel content will increase drastically GHG levels in comparison to slightly richer diesel contents.

An interesting trend is seen only for lean conditions since an optimal DEF, between 2% and 4%, seems to minimize CO_{2eq} under 100g/kWh. For each ϕ_{premix} with increasing diesel content, first GHG levels are high due to high N_2O levels with low CO_2 content, then CO_2 levels increase slightly but N_2O decreases drastically. Finally, for DEF >=4% the GHG rises due to increasing levels of CO_2 despite low levels of N_2O. At stoichiometric or slightly rich conditions, at fixed ϕ_{premix}, the N_2O levels are always so low that the slight increase of GHG levels is related to the DEF increase inducing lower GHG increase than at lean conditions. It has to be underlined that in this study, the intake pressure was maintained constant, inducing a strong decrease of IMEP at lean conditions.

Conclusions

In this study the impact of diesel and ammonia content on the performance and emissions in a RCCI engine have been investigated for the first time. The combustion properties of ammonia and its impact on the oxidation of diesel make difficult the control of this combustion mode as a function of the injection start and the duration of injection. The diesel energy fraction was minimized up to 2% with stable operating conditions (5% of IMEP variation maximum) over a wide variation of the premixed ammonia/air equivalence ratio (from 0.6 to 1.1). It was possible to operate at ultra-lean conditions (up to ϕ_{premix}=0.25) even if the combustion was not efficient or the engine stable. Both the ammonia/air premixed equivalence ratio and the diesel energy fraction affect the combustion development and the exhaust emissions. The change of the diesel energy fraction, done with four different DOI strategies, has a major role in emissions. A trade-off has to be made with this technology since all pollutants and equivalent GHG are at their lowest value for a premixed ammonia/air equivalence ratio around 0.8 with the correct DOI strategy. This comes with a trade-off for NO_X that reaches its maximum. Finally, this study has highlighted the major importance of considering N_2O emissions, even if their absolute level is low, by considering its equivalent global warming impact. The study demonstrates that reducing the diesel amount of the pilot injection as low as possible might not be necessarily the best strategy to minimize carbon footprint. An optimization of both the ammonia/air premixed equivalence ratio and the diesel energy fraction has to be found to ensure this minimum. Finally, the combustion duration and the maximum of average in-cylinder temperature have been proven to be two key indicators, depending of the premix equivalence ratio, on the levels of N_2O at the exhaust.

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