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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 important ignition energy for large cylinders, engine manufacturers are targeting the use of ammonia in 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 engine. 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 constant 1000 rpm. Experiments cover the impact of a wide variation of the equivalence ratio of NH₃-air mixture, from ultra-lean to slightly rich conditions. CO₂, CO, NH₃, NO_x, N₂O, UHC values were measured with a Fourier Transform Infrared (FTIR) spectrometer. Results of CO₂ and N₂O are presented as CO₂ equivalent impact. The combustion stability was achieved for most of conditions but not 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. Last, both leanest and richest conditions present a higher level of CO_{2eq} compared to the range of ammonia/air mixture at stoichiometry or just below.

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Keywords: Ammonia, Diesel pilot ignition, Emissions, performances.

1 Introduction

2 Ammonia is more and more recognised as one
3 chemical storage solution with the potential of
4 delivering green hydrogen over long distances and
5 for heavy loads [1], especially to store and flexibly
6 distribute energy obtained from renewable sources
7 such as wind, solar and marine [2]. The IMO's
8 Initial Strategy for Green House Gases Reduction,
9 in 2018, pointed out ammonia will be not only a
10 'storage molecule' but one fuel necessary to
11 decarbonize the maritime sector for 2050. However,
12 several challenges still remain for the use of
13 ammonia as a fuelling vector. Out of all the main
14 challenges recognised in the transition of a
15 hydrogen-ammonia economy [3], public perception,
16 adequate legislations and a better knowledge on
17 environmental impacts are probably the most
18 complex to deal with. Angeles et al. [4] concluded
19 that internal combustion engine vehicles using
20 biomass-based ammonia could deliver lower carbon
21 footprint profiles than diesel or gasoline fuel.
22 Similarly, Bicer and Dincer [5] analysed various city
23 transportation and power generation systems fuelled
24 with ammonia from nuclear plants. The results
25 denoted a considerable decrease in greenhouse gas
26 emissions with an acute impact on global warming
27 potential. However, the results also demonstrated
28 that with the reduction of carbon footprint the
29 increase in nitrogen-based species (ie. NO_x) could
30 be considerably high, shadowing the overall balance
31 in greenhouse gases mitigation. Finally, a recent

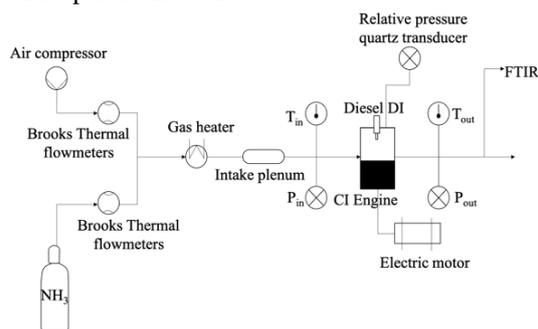
32 report produced by the Maritime Cleantech Cluster
33 [6] stated that the use of liquid ammonia in internal
34 combustion engines for maritime purposes can lead
35 to "net zero emissions". However, careful
36 considerations need to be taken to avoid large GHG
37 emissions during the combustion itself. The report
38 raises the importance of fully evaluating some of the
39 most environmentally damaging molecules of
40 ammonia combustion, as N₂O, which requires
41 further studies. Thus, it was concluded that ammonia
42 is a free-carbon fuel but not necessarily a "zero-
43 emissions" combustion fuel. Since Spark Ignition
44 (SI) engines using ammonia require important
45 ignition energy [7] for a large cylinder like the case
46 of X40DF-1.0 [8], most marine engine
47 manufacturers announced that ammonia will be used
48 as fuel in Compression Ignition (CI) engines. Due to
49 the high auto-ignition temperature and the low
50 reactivity of ammonia, dual-fuel concept seems to
51 be more adapted as in [9-11]. But in order to reach
52 zero carbon emissions as well as possible, the
53 amount of diesel has to be limited. For that RCCI
54 (Reactivity Controlled Compression Ignition
55 Combustion) concept is the most adapted as
56 underlined in [12, 13] : a pilot fuel with a higher
57 reactivity ensures the combustion ignition and
58 stability of the premixture of ammonia/air. The high
59 reactivity fuel chosen is the diesel one as a current
60 fuel on board vessels. Only two recent study is
61 focused on this ignition of ammonia engine by diesel
62 pilot injection [14, 15]. Therefore, the objectives of
63 the present study are to evaluate the potential of
64 RCCI for ammonia engine in terms of stability and

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1 performance, i.e. to use a diesel energy fraction as
 2 minimum as possible - below 2% for certain
 3 operating conditions - and to characterize the
 4 exhaust emissions. Experimentations cover the
 5 impact of a wide variation of the premixed
 6 equivalence ratio (Ultra-lean to rich premixed
 7 ammonia-air mixture) and of the Diesel Energy
 8 Fraction (DEF) (maximum 12.5%) on the
 9 performances and emissions in a single cylinder 0.5
 10 l engine. A special attention will be given to N_2O due
 11 to its large global warming effect: at equivalent
 12 mass, N_2O has a global warming potential 265 times
 13 more important than CO_2 as recently indicated the 5th
 14 assessment of IPCC (Intergovernmental Panel on
 15 Climate Change) [16]. For the first time, some
 16 quantitative values about the global warming impact
 17 of this kind of engine without any exhaust post-
 18 treatment system are provided.

19 Experimental set-ups and operating 20 conditions

21 Engine tests were conducted on a four-cylinder
 22 Compression Ignition PSA DW10 light duty engine
 23 converted into a 0.5 l single cylinder engine. All
 24 experimental set-up details are shown in Fig. 1, with
 25 more details specified in Table.1. The engine speed
 26 was maintained constant at 1000 rpm. The intake
 27 and exhaust temperature and pressure were acquired
 28 by K thermocouples and resistive absolute pressure
 29 transducer respectively. The in-cylinder pressure
 30 from a relative quartz pressure transducer Kistler
 31 6045A was acquired via a LabView data acquisition
 32 system, with a resolution of 0.1 CAD, provided by a
 33 Kubler optical encoder.
 34



35
 36 **Fig. 1** : Global view of experimental set-up

37 Ammonia, the 'low reactive' fuel was stored in
 38 liquid form in a tank and heated to be evaporated
 39 before entering inside the intake plenum to be mixed
 40 with air. Ammonia and air flow rates were
 41 controlled by two Brooks thermal mass flowmeters.
 42 The premixture was heated to be admitted up to the
 43 intake desired temperature ($T_{in}=80^{\circ}C$).
 44 For each operating condition, 200 consecutive
 45 cycles of pressure data are recorded to analyze
 46 cycle-to-cycle variations and compute averaged

47 pressure values. From the average pressure data, as
 48 a function of the crank-angle degree (each 0.1
 49 CAD), the apparent fuel-combustion heat release is
 50 computed from the first law of thermodynamic, first
 51 by considering constant γ value as :

$$52 \quad \frac{dQ}{d\theta} = \frac{\gamma}{\gamma - 1} P \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dP}{d\theta}$$

53 With γ the heat capacity ratio, P the in-cylinder
 54 pressure, V the in-cylinder volume and θ the crank
 55 angle degree. Then, the average in-cylinder
 56 temperature as a function of the crank-angle degree
 57 is estimated. By using the C_p function of the
 58 temperature, C_v and γ are then computed each 0.1
 59 CAD. Then the heat release rate is calculated a
 60 second time with this variable γ . The indicated work
 61 (IMEP) was evaluated by integrated the measured
 62 pressure data with the volume variation. The blowby
 63 and the internal recirculation of gases were not
 64 considered here.

65 **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
Bowl type	Re-entrant
Valves per cylinder (Int./Exh.)	(2/2)
Engine speed (rpm)	1000
Intake valve opening/closing	-117/344 ATDC
Exhaust valve opening/closing	134/396 ATDC
Swirl ratio (50 CAD BTDC)	2.0
Engine speed (rpm)	1000
Port intake pressure (bar)	1
Port intake temperature ($^{\circ}C$)	80

66

67 Exhaust emissions measurement

68 A fast Fourier Transformed InfraRed (FTIR)
 69 DX2000 spectrometer was used to measure the
 70 exhaust gases concentration of following species
 71 H_2O , CO_2 , CO , NH_3 , NO_x , N_2O , UHC. Special
 72 attention was given to the accuracy of N_2O
 73 measurement at the exhaust especially at low
 74 concentrations (lower than 50 ppm). For that, a
 75 calibrated bottle of N_2O/N_2 was used to evaluate the
 76 interferences with other gases, as for instance CO_2 .
 77 Thus, the optimized wavelength range 2300-2450
 78 cm^{-1} for N_2O and 2100-2350 cm^{-1} for CO_2 was
 79 selected, validated for a mixture $CO_2/N_2O/N_2$.
 80 During experiments, the exhaust gases samples are
 81 heated up to $190^{\circ}C$ before being analyzed by the
 82 FTIR software. Each sample was collected and
 83 analyzed every five seconds. For each condition, 20
 84 consecutive samples were acquired after
 85 stabilization. For N_2O calculations, among these
 86 samples, only ones with acceptable residuals were

1 considered. Standard deviations were computed but
2 only N_2O ones are presented in this paper.

3 Ammonia and Diesel amount conditions

4 The ‘high reactivity’ fuel, the diesel, was directly
5 injected inside the cylinder with a CRI Bosch
6 injector. The pressure of injection was maintained at
7 200 bar to ensure sufficient flexibility on injection
8 duration to adjust very low quantity as well as
9 possible. The characterization of injected diesel
10 mass was done as a function of different Durations
11 of Injections (DOI). A second order fit was used
12 afterwards to know the mass injected in each case of
13 the experiment. At each condition of ammonia/air
14 premixture and for each DOI, the Start Of Injection
15 (SOI) was carefully chosen to ensure the stability of
16 the combustion and therefore to minimize the
17 variation of the Indicated Mean Effective Pressure
18 (IMEP). 4 different DOI were investigated to
19 experiment the effect of the diesel energy fraction
20 value on the engine performance and exhaust
21 emissions: the DOI_{min} corresponding to the
22 minimum amount of diesel injected which ensures
23 stable combustion, $DOI_{min} + 50$, 100 and 150 μs . For
24 the leanest ammonia/air premixture conditions, the
25 dead-zone of the injector imposed a DOI_{min} of
26 400 μs . The values of DOI minimum as a function of
27 the ammonia/air equivalence ratio are shown in
28 Figure 2. This DOI_{min} starts to increase only after
29 $\Phi_{premix} > 0.9$ due to the greater diesel demand
30 to ensure a stable combustion at each cycle.

31 From a calibration, a Diesel Energy Fraction (DEF)
32 was determined as below:

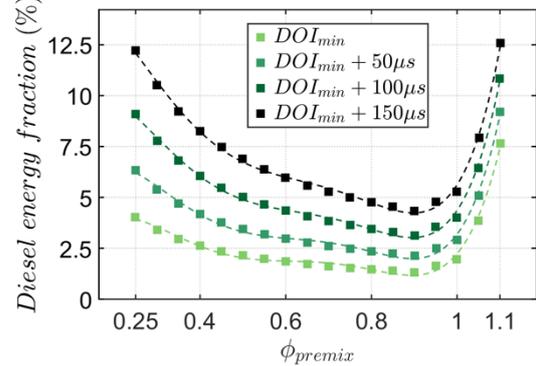
$$33 \quad DEF = \frac{m_{diesel} \cdot LHV_{diesel}}{m_{diesel} \cdot LHV_{diesel} + m_{NH_3} \cdot LHV_{NH_3}}$$

34 Results

35 Combustion development and characteristics

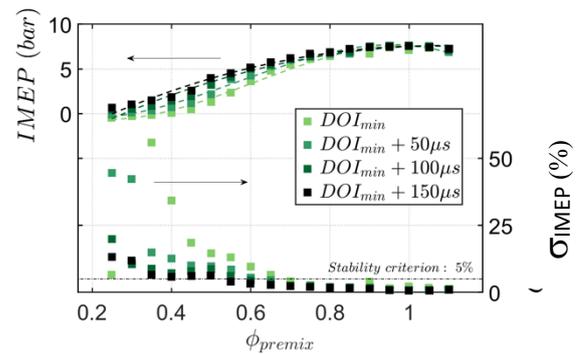
37 As it can be seen in Figure 3, the durations of Diesel
38 injection were set in order to ensure maximum
39 engine stability as well as possible. A minimum of
40 2% of DEF was possible only for NH_3 /air
41 equivalence ratio between 0.8-0.9. As expected, due
42 to the low flammability range for ammonia, it was
43 necessary to increase the diesel injection amount for
44 the leanest mixtures up to 12.5% but for a very low

45 equivalence ratio (0.25).

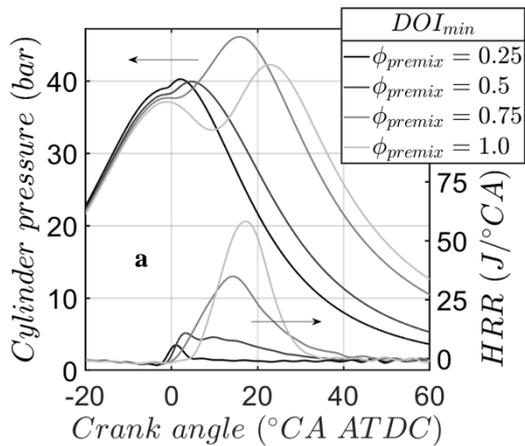


46
47 **Fig. 3:** Different DEF corresponding to the
48 different DOI as a function of ϕ_{premix}

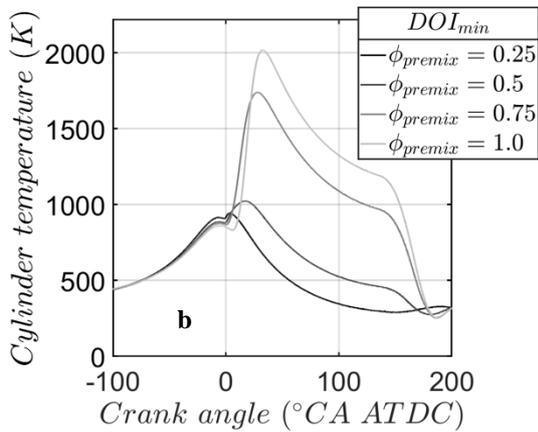
49 In Figure 4, the Indicated Mean Effective Pressure
50 (IMEP) was plotted for all conditions with the
51 corresponding variability (σ_{IMEP}). From 0.8 to 1.1 of
52 Φ_{premix} , a maximum of IMEP is obtained with a very
53 good engine stability. When the duration of diesel
54 injection is slightly increased, it was possible to
55 ignite leaner ammonia/air mixture. Figure 5a
56 highlights the delay of the combustion as a function
57 of the equivalence ratio of the ammonia/air mixture
58 when the minimum of diesel amount to ensure
59 ignition of the premixed charge is injected.
60 Moreover, the more ϕ_{premix} is rich, the more the heat
61 release rate is important. These results agree with
62 those of Wüthrich et al. [13]. In Figure 5b, the
63 estimated averaged thermodynamic temperature
64 along the cycle is plotted and indicates how this
65 temperature could be high when stoichiometric
66 mixture is used. Yet, lean conditions induce a
67 temperature peak below 1000 K due to very low heat
68 release rate and therefore a poor ammonia
69 combustion efficiency.



70
71 **Fig. 4:** Effect of Diesel amount on IMEP as a
72 function of ϕ_{premix}



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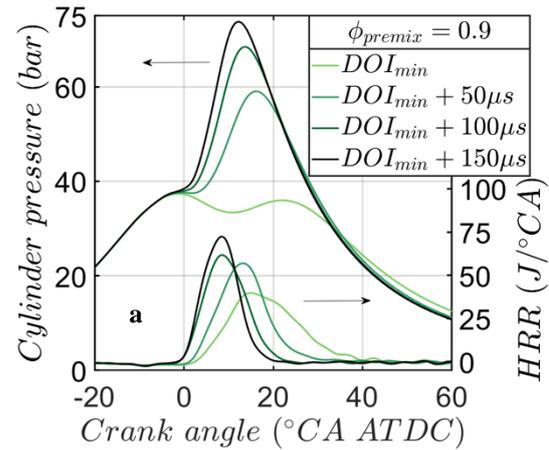
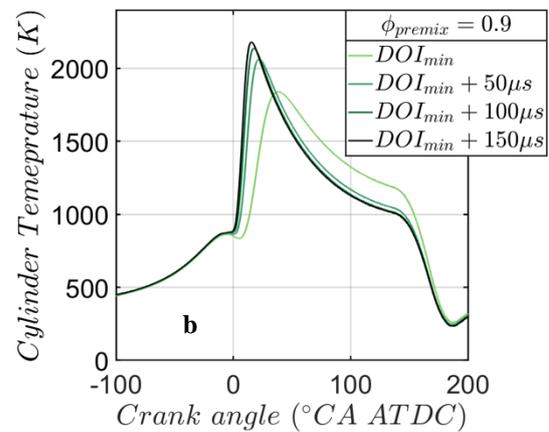


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4 **Fig. 5: (a)** in-cylinder pressure and heat release rate
 5 and **(b)** average thermodynamic temperature
 6 evolutions as a function of crank angle degree for
 7 some different air/ammonia premixtures with
 8 minimum diesel content to guarantee ignition and
 9 combustion stability

10
 11 Last, an example of the effect of diesel amount on
 12 the combustion development is shown in Figure 6,
 13 for the equivalence ratio of 0.9. The different
 14 durations of diesel injection induce different amount
 15 of diesel energy fraction (1.5, 2.5, 3.5 and 5%).
 16 When the diesel content is not minimum, the shape
 17 of the heat release rate and pressure does not really
 18 change with the increase of diesel amount, this only
 19 affects the maximum reached. The temperature
 20 profile itself is similar.

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Fig.6: (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 $\phi_{premix}=0.9$

Exhaust emissions : unburnt species and CO₂-CO

The emissions of unburnt ammonia are a real issue due to the environmental impact if no after-treatment is added [3]. In Figure 7a, one can notice that the maximum of unburnt ammonia level is obtained at ultra-lean conditions, and this maximum peak decreases with the increase of diesel content, due to the improvement of the combustion for these mixture as a longer diesel injection 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 the combustion of ammonia is optimized due to the relative importance of diesel energy fractions.

Furthermore, with longer DOI strategies, more high reactivity zones are present inside the chamber and located at different zones thus creating more local ignition points.

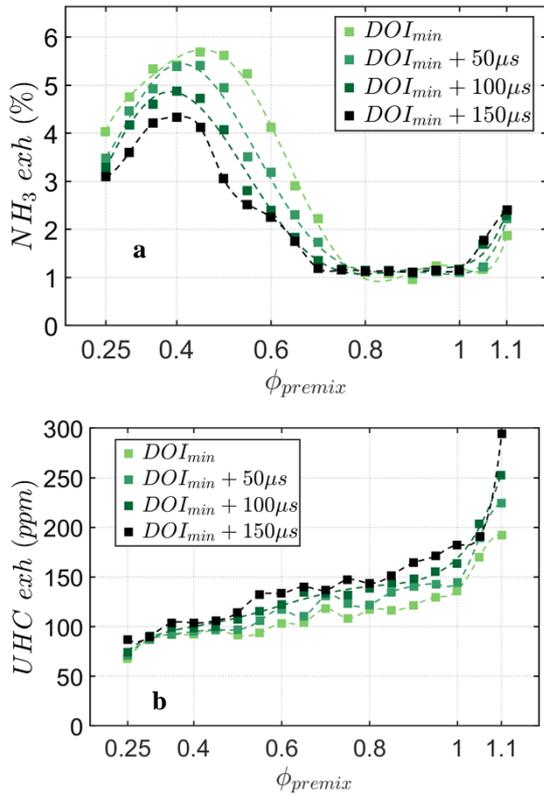


Fig.7: emissions evolution of unburnt (a). ammonia and (b) hydrocarbons

Longer DOI might increase the local turbulence thus improving the combustion. Besides, with longer DOI strategies, more high reactivity zones are present inside the chamber and located at different zones creating more ignition spot 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 by a diesel pilot injection. As expected, in rich condition, even slightly, the emissions increase due to the excess of fuel. By considering Figure 7b, it is worth to notice that Unburnt HydroCarbons levels increase with both ammonia and diesel content. To better visualize these effects, the ratio between the UHC at the exhaust (in mass) and the mass of diesel injected at the intake is estimated (by assuming UHC as similar molecular weight than diesel) and plotted in Figure 8. A peak of UHC, i.e., less efficient combustion of diesel occurs at $\phi_{premix}=0.8$, whatever the fraction of diesel but the combustion of diesel is improved when increasing the DOI. Above this $\phi_{premix}=0.8$, UHC ratio decreases with ammonia content and strongly

with diesel content. Therefore, it can be concluded that the combustion of the diesel in an ammonia/air mixture in engine operating conditions is improved when the amount of diesel is not set at the minimum possible. In fact, ammonia has an inhibiting effect on the diesel combustion especially in the case of small diesel amount. This was observed during the experiments near the minimum of diesel fraction needed to burn the premixed charge: just below this limit ammonia inhibits diesel ignition and increase auto-ignition delay as concluded in [19] with n-heptane/ammonia blend.

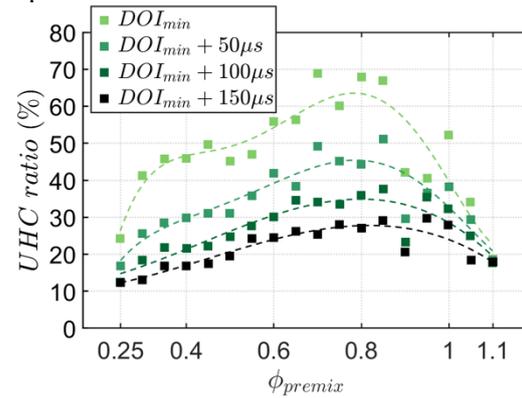


Fig.8: evolution of unburnt hydrocarbons ratio (exhaust UHC (mass)/diesel amount (mass))

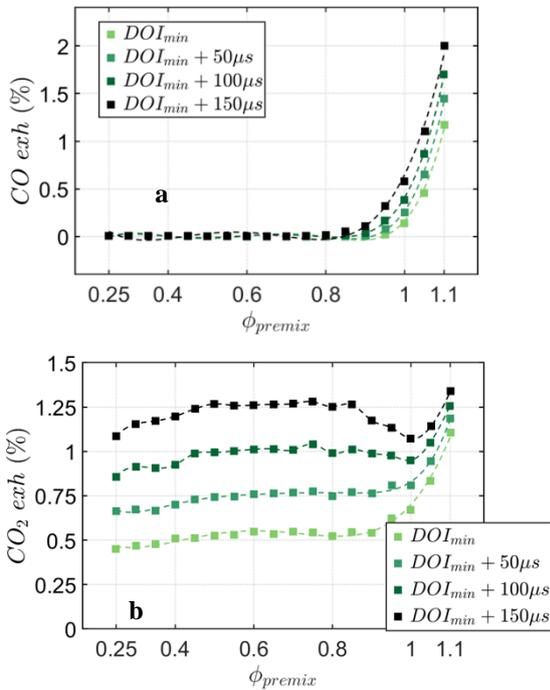
In Figure 9, it can be noted also that for the same threshold value, $\phi_{premix}=0.8$, the emissions of CO change: below this value, no CO emissions and above this value, more and more CO is emitted, despite the constant or even higher diesel amount introduced, compared to leaner ϕ_{premix} . While for $\phi_{premix}>0.9$, 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 $\phi_{premix}=0.9$ 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 which leads to the decrease of CO_2 . As expected CO_2 levels are greater for longer DOI strategies.

Exhaust emissions : NO_x species

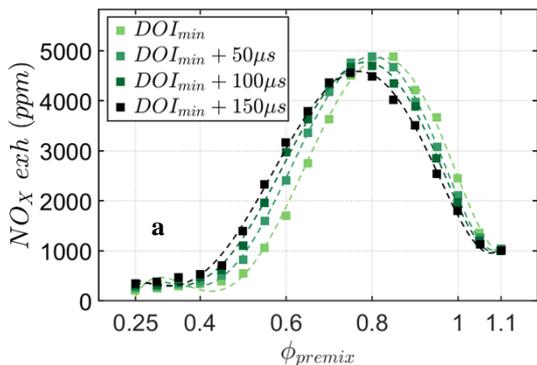
As previously measured in [17, 18] with the same engine but ignited by a spark, NO_x emissions, plotted in Figure 10.a, reach a maximum level near an equivalence ratio of 0.8. Adding diesel changes only very slightly the equivalence ratio corresponding to the maximum: more diesel induced a peak of NO_x for leaner conditions but with a slight decrease of the peak. For example, at $\phi_{premix}=0.9$, for DOI_{min} the NO_x emission is 4211 ppm for a peak average temperature of 1839 K and for greater diesel content with $DOI_{min}+150\ \mu s$ NO_x level is at 3506

1 ppm for a peak average temperature of 2180 K. But
 2 for all cases at $\phi_{\text{premix}} = 1.1$, the level remains at 1000
 3 ppm, contrary to previous studies where zero
 4 emission of NO_x is obtained, certainly due to the
 5 pilot diesel injection.
 6



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 10 **Fig.9:** evolution of (a) CO and (b) CO₂ emissions
 11 for all conditions

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 13 Figure 10.b confirms that NO₂ remains very
 14 neglectable: almost zero above $\phi_{\text{premix}}=0.6$ and
 15 below 50ppm for ultra-lean conditions, with a
 16 positive effect of diesel amount. Adding diesel will
 17 decrease even more these levels mostly due to higher
 18 temperature reached compared to DOI_{min}. Indeed,
 19 when NO₂ is plotted as a function of the maximum
 20 average thermodynamic temperature (Figure 11), it
 21 puts in evidence that NO₂ emissions increase when
 22 this maximum is low due to the lean ammonia/air
 23 mixture.



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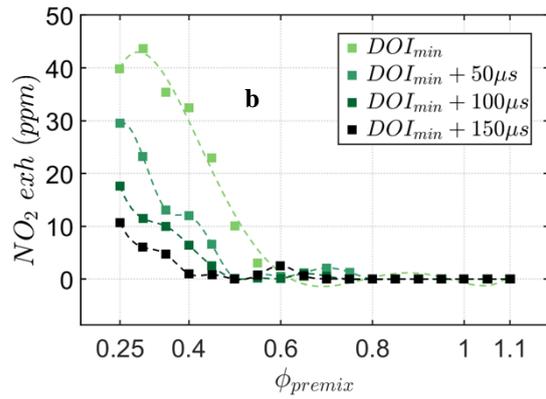
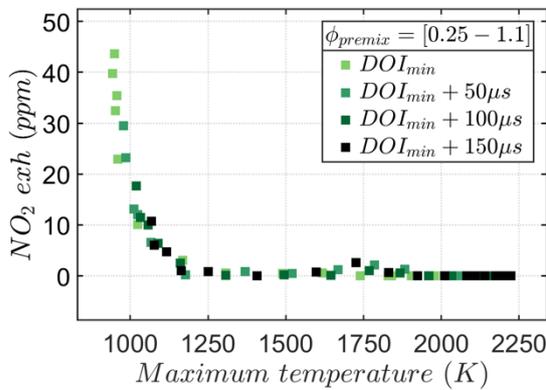


Fig.10: evolution of (a) NO_x and (b) NO₂
 emissions for all conditions



28
 29 **Fig.11:** evolution of NO₂ emissions as a function of
 30 maximum average in-cylinder temperature.

31 **N₂O emissions**

32 For all operating conditions, N₂O levels remain
 33 below 150 ppm but present two maximum peaks,
 34 one at ultra-lean conditions ($\phi_{\text{premix}}=0.45$), and
 35 another lower one around stoichiometry, with a
 36 minimum achieved at slightly rich conditions
 37 ($\phi_{\text{premix}}=1.1$). This specific trend of N₂O emissions as
 38 a function of ammonia/air mixture is in a good
 39 agreement with that obtained in a swirl burner [21].
 40 The increase of diesel amount for a same ϕ_{premix}
 41 decreases the levels of N₂O, except for DOI
 42 minimum, where the standard deviation indicates
 43 less stability. Figure 12b illustrates N₂O emissions
 44 (g/kWh) at the exhaust from $\phi_{\text{premix}}=0.5$ to
 45 $\phi_{\text{premix}}=1.1$. First, the lowest indicated power due to
 46 the leanest ammonia/air mixture leads to the highest
 47 levels of N₂O. For each premixed equivalence ratio,
 48 the longer the DOI the higher the DEF. It seems that
 49 to reduce N₂O as much as possible the good strategy
 50 is to run at slightly rich conditions ($\phi_{\text{premix}}=1.1$) or
 51 lean conditions ($\phi_{\text{premix}}= 0.7-0.9$), as ultra-lean and
 52 near stoichiometry cases increase significantly N₂O
 53 at the exhaust. For conditions ($\phi_{\text{premix}}= 0.7-0.9$), it
 54 seems that to reduce N₂O as much as possible
 55 limiting the diesel energy fractions as well as
 56 possible (below 2%) is not the optimised strategy.

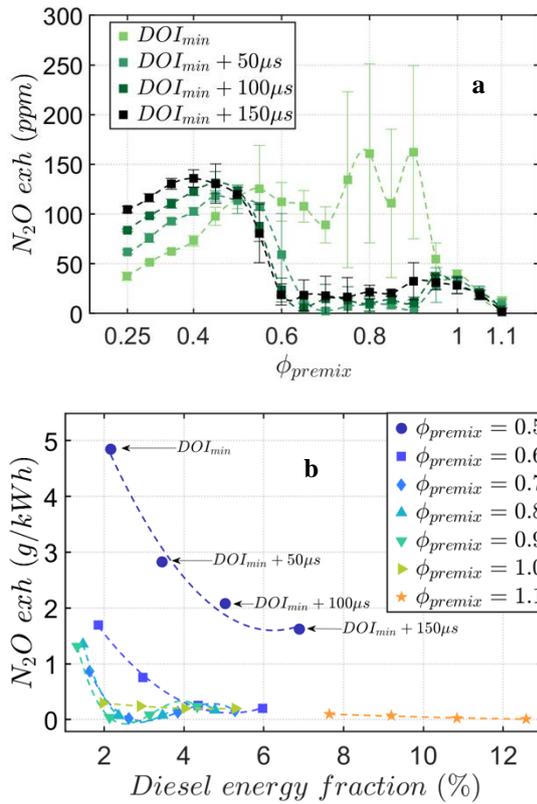


Fig.12: evolution of N_2O emissions as a function of ammonia/air equivalence ratio (a) and diesel energy fraction (b)

To understand better the key parameters which drive N_2O emissions, the impact of the combustion duration and of the maximum average in-cylinder temperature on N_2O emissions is provided for 3 ranges of premixed equivalence ratio: ultra-lean conditions (a) $\phi_{premix} = 0.25-0.5$, lean conditions (b) $\phi_{premix} = 0.55-0.8$, near-stoichiometric conditions (c) $\phi_{premix} = 0.85-1.1$ as ranked in Figure 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 assessed that except for the lowest peak of temperature ($T_{peak} < 1100$ K), N_2O emissions are only affected by the combustion duration since levels are rising constantly with longer combustion duration. 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 at the input 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. But, both, 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 appear increasingly at longer combustion duration and lower T_{peak} , except at DOI_{min} where the standard deviation is really too important. Finally, at near-stoichiometric conditions, both factors have a huge impact on N_2O emissions since with shorter combustion duration than $13^\circ CAD$ and $T_{peak} > 2000$ K, the levels of N_2O are constant under 50 ppm. It is also observed that a narrow window of T_{peak} ($2050 K < T_{peak} < 2100 K$) seems to minimize under 20 ppm N_2O at the exhaust for near-stoichiometric conditions. 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 the combustion duration.

Discussion about Global warming impact for RCCI ammonia engine

Since 1 kg of N_2O has been assessed in [17] 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 $\phi_{premix} = 0.5$ to $\phi_{premix} = 1.1$.

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

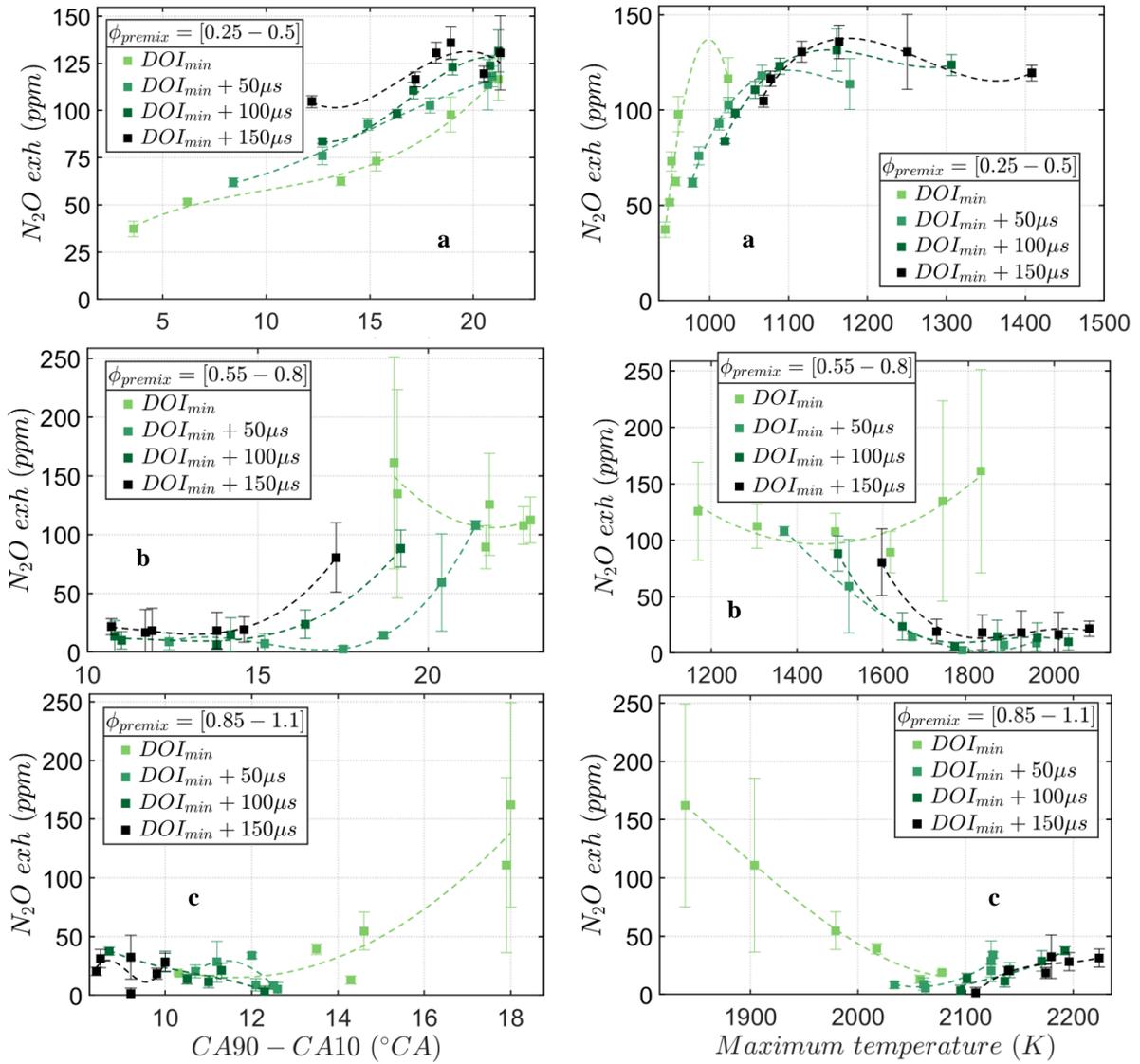


Fig.13. N_2O emissions as a function of combustion duration and maximum averaged in-cylinder temperature for ultra-lean (a), lean (b) and near stoichiometry conditions (c)

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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 to be retained has to be between $\phi_{premix}=0.7$ and $\phi_{premix}=0.9$ since they are the only ones reaching under 100g/kWh of GHG levels. 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 content. Finally, an interesting trend is seen only for lean conditions since an optimal DEF, between 2% and 4%, seems to allow 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 levels, then CO_2 levels increase slightly but N_2O ones decrease drastically for DEF between 2% and 4% and for DEF $\geq 4\%$ the GHG rises due to increasing levels of CO_2 despite low levels of N_2O . Otherwise 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 and induces higher GHG increase than at lean conditions, but it has to be underlined that in this study, the intake pressure was maintained constant, inducing a strong decrease of IMEP at lean conditions.

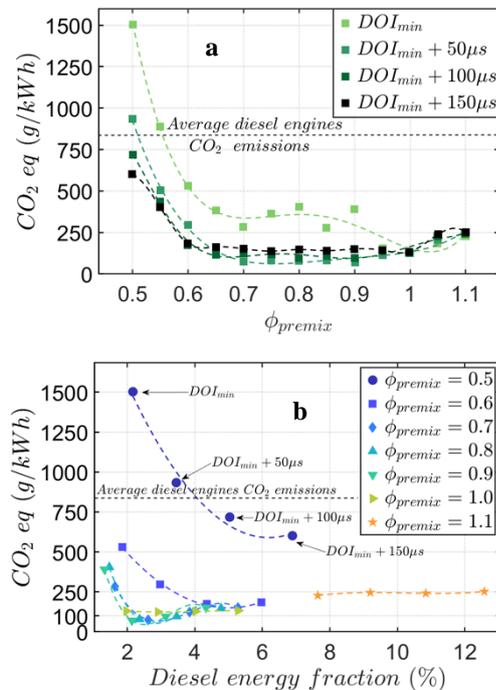


Fig.14 CO₂ equivalent emissions as a function of ϕ_{premix} (a) and diesel energy fraction (b).

Conclusions

In this study the impact of diesel and ammonia content on performances and emissions have been investigated in a RCCI engine 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 start of injection and the duration of the injection. The diesel energy fraction had been minimized until 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 (until $\phi_{premix}=0.25$) even if the combustion was not efficient and 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 except NO_x which reaches its maximum. Last, this study has highlighted the major importance of considering N_2O emissions even if its absolute level is low, by considering its equivalent global warming impact. This study demonstrates that reducing the diesel amount of the pilot injection as well as possible is not necessarily the best strategy to minimize the 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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