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Experimental investigation of the effects of simultaneous hydrogen and nitrogen addition on the emissions and combustion of a diesel engine[☆]

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ABSTRACT

Overcoming diesel engine emissions trade-off effects, especially NO_x and Bosch smoke number (BSN), requires investigation of novel systems which can potentially serve the automobile industry towards further emissions reduction. Enrichment of the intake charge with H₂ + N₂ containing gas mixture, obtained from diesel fuel reforming system, can lead to new generation low polluting diesel engines.

This paper investigates the effect of simultaneous H₂ + N₂ intake charge enrichment on the emissions and combustion of a compression ignition engine. Bottled H₂ + N₂ was simultaneously admitted into the intake pipe of the engine in 4% steps starting from 4% (2% H₂ + 2% N₂) up to 16% (v/v).

The results showed that under specific operating conditions H₂ + N₂ enrichment can offer simultaneous NO_x, BSN and CO emissions reduction. Apart from regulated emissions, nitrogen exhaust components were measured. Marginal N₂O and zero NH₃ emissions were obtained. NO/NO₂ ratio increases when speed or load increases. Under low speed low load operation the oxidation of NO is enhanced by the addition of H₂ + N₂ mixture. Finally, admission of H₂ + N₂ has a detrimental effect on fuel consumption.

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1. Introduction

Reduction of diesel exhaust emissions can be accomplished either with in-cylinder emissions formation suppression [1] or with aftertreatment technologies [2]. The existence of inverse relationship between emissions, especially NO_x and PM, makes the in-cylinder emissions control a very challenging

task. Hence, novel solutions are required in order to further reduce the levels of pollutants leaving the cylinder.

Some parameters affecting emissions formation which can be adjusted relatively easy through the ECU are injection timing, injection pressure and EGR volume fraction. Buyuk-kaya and Cerit [3] performed experiments on a low heat rejection diesel engine and found that retardation of injection timing lowers NO_x emissions at the expense of PM. A negative

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relationship between NO_x and PM emissions is observed when intake charge is diluted with EGR or injection pressure is varied [4]. Reduction of intake charge's oxygen concentration lowers NO_x at the expense of PM [5].

Various techniques that simultaneously suppress the formation of both NO_x and PM emissions have been investigated but only few of them have been put into massive production. Mkilaha et al. [6] modified a four cylinder indirect injection diesel engine to supply compressed air into the pre-chamber during combustion. At no load conditions they demonstrated simultaneous NO_x and PM reductions. They concluded that NO_x reduction is attributed to flame quenching while improved mixing resulting from air injection lowers PM. It has been reported by many researchers that dual fuelling is a feasible solution to the reduction of emissions which could also lead to sustainable transportation. Dual fuelling can affect various combustion parameters such as flame propagation, ignition delay, adiabatic flame temperature, flammability limits, reduction of intake's charge oxygen concentration and combustion duration which in turn influence engine's emissions and performance. Papagiannakis and Hountalas [7] investigated the effect of diesel-natural gas dual fuelling on performance and emissions of a DI diesel engine. They showed that in most of the operating points tested admission of natural gas resulted in simultaneous NO and PM emissions reduction. They also reported that fuel consumption, CO and HC emissions deteriorated under dual-fuel compared to baseline operation. Simultaneous NO_x and PM emissions reduction can also be realised when feeding the engine with combined hydrogen and EGR [8]. Bika et al. [9] reported that part substitution of diesel fuel with syngas introduced into the intake manifold of a compression ignition engine considerably increases CO emissions. Moreover, they detected unburned hydrogen into the exhaust gas and they also showed that due to poor gaseous fuel utilisation thermal efficiency decreased compared to baseline operation. Saleh [10] experimentally proved that admission of propane into the inlet manifold of a diesel engine increases fuel conversion efficiency when the mass fraction of the gaseous fuel supplied is up to 40%. He also compared the fuel conversion efficiency obtained when the engine was fuelled on neat propane and propane-butane blends and demonstrated that when the butane composition in the gas mixture is increased fuel conversion efficiency drops. Nevertheless, increasing butane fraction has a beneficial effect on NO_x emissions.

Roy et al. [11] combined in-cylinder and aftertreatment emissions reduction techniques in a single cylinder, naturally aspirated DI diesel engine. NO_x formation was suppressed using EGR whereas the increased PM emissions, associated with EGR, were treated by a cyclonic separator fitted into the exhaust pipe. The particles were subject to centrifugal forces enabling this way the separation of soot from the exhaust gas. Nakatani et al. [12] reported over 80% reductions in both NO_x and PM emissions by developing an aftertreatment system called DPNR (Diesel Particulate – NO_x Reduction System). In addition to the aftertreatment technology installed, the engine required periodical alternation from lean to rich combustion.

Johnson presented a review on NO_x control and PM reduction technologies [13]. Diesel particulate filter is the technology currently used to remove PM from the exhaust gas, whereas

NO_x emissions are treated with the aim of a selective catalytic reduction system. The integration of both in the exhaust pipe allows simultaneous NO_x and PM aftertreatment. Yoshinobu et al. [14] proposed a simultaneous electrochemical reduction system to reduce NO_x and PM pollutants from diesel engines. They claimed NO_x and PM reductions over 90%.

The implementation of new PM emission regulations, i.e. introduction of the particulate number, reveals the importance to lower soot emissions. Valavanidis et al. [15] reviewed the results of the latest epidemiological and toxicological studies. They summarized that small PM are more harmful to human, compared to those having greater size. In addition, further NO_x emissions reduction will be beneficial to both human and environment since NO_x contribute, among others, to ozone depletion, acid rain and respiratory problems.

The objective of this paper is to present the effect of simultaneous hydrogen and nitrogen addition on the emissions and combustion of an HSDI diesel engine. Bottled gases, simulating diesel fuel reforming product gas were introduced into the engine through the intake port. Carbon monoxide at relatively high concentration is typically present in diesel reforming product gases, nevertheless, through the water gas shift reaction using a good low-temperature catalyst CO concentration can be reduced to ppm levels. The effect of combined syngas and nitrogen addition, aiming to build on the knowledge obtained on previous [16] and current study, is currently investigated by the authors and the results will be reported in a separate paper. In the current research effort, along with the regulated emissions, measurements of nitrogen exhaust components such as NO , NO_2 , NH_3 and N_2O are provided.

2. Experimental setup

Fig. 1 shows the experimental setup which has been extensively described in previously published work [16]. At this point only a brief description of the experimental setup and experimental procedure is provided.

The experiments were carried out in a Ford Puma HSDI diesel engine. Its main specifications are: 4 cylinders, 2.0 L, 16 valves, turbo charged (not employed during the tests), water cooled, fuelled by ULSD, bore 86 mm, stroke 86 mm, compression ratio 18.2:1.

Speed and load variations were achieved through a Schenk eddy current dynamometer connected to engine's output shaft. In-cylinder pressure as a function of crank angle was recorded through the interaction of a LabView software, a Kistler 6125A pressure transducer, a Kistler 5001 charge amplifier and a shaft encoder. Throughout the experiments fuel consumption was measured by a Coriolis flow meter and a glass burette. The latter was actually employed for fuel flow verification purposes. Injection timing was controlled by a software that gave direct access to the ECU and allowed the user to program it.

Introduction of bottled gases into the engine through the intake port suggests that an equivalent volume of intake air was replaced by the gas mixture. The ambient air volume flow was measured by a positive displacement air flow meter while the volume of $\text{H}_2 + \text{N}_2$ entering the engine was measured by glass tube flowmeters. The gases were stored in separate

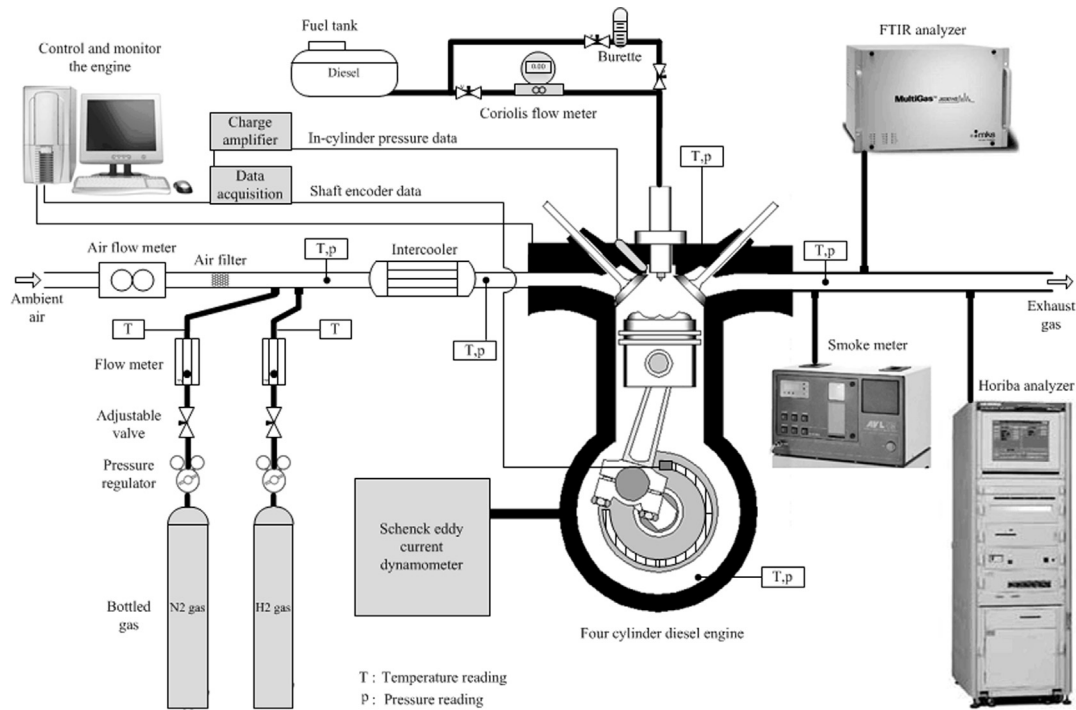


Fig. 1 – Experimental setup.

bottles and all the safety precautions were taken, i.e. backfire arrestor and relief valve. A pressure regulator was mounted at each cylinder to control the line pressure and track the remaining gas. Access points on the intake manifold allowed collection of intake charge samples which were analysed using a gas chromatograph, in order to verify whether the gases are thoroughly mixed.

The exhaust gas was analysed using an FTIR MultiGas 2030, a Horiba Mexa 7170DEGR and an AVL smoke meter. Nitrogen exhaust components were measured by the FTIR. Carbon monoxide and smoke number were obtained by the Horiba and smoke meter, respectively. All the instruments were fed directly from the exhaust pipe as schematically shown in Fig. 1.

Four operating conditions representative for low and medium duty diesel engines were tested in the current experimental investigation. The test matrix, which is shown in Table 1, includes variations in engine speed (1500 and 2500 rpm), load (2.5 and 5 bar BMEP), start of injection (3–12 CAD BTDC in 3° steps) and mixture concentration. The latter progressively replaced intake air starting from 4% replacement (volume of intake charge) up to 16%, in 4% increments.

The diesel fuel was delivered into the cylinder using a single injection event having the pressure set at 800 bar.

Moreover, the pressure in $H_2 + N_2$ lines was fixed at 3 bar. This is actually the pressure required in order to achieve the maximum flow rate of the admitted gases.

3. Hydrogen and nitrogen enrichment

This work extends beyond separate hydrogen and nitrogen addition [16], by simultaneously feeding the engine with the two gases in order to assess the effect on emissions and combustion. The study aims to simulate the supply of oxygen free reformer products into the engine. Depending on the reactions promoted, hydrogen can be the main combustible product of diesel fuel reformation, while nitrogen simulates intake's charge oxygen concentration reduction, i.e. increased N_2 to O_2 ratio. Table 2 illustrates the intake charge composition at various $H_2 + N_2$ fractions supplied.

The effect on the combustion of a diesel engine when separate H_2 and N_2 gas is delivered into the engine has been extensively discussed in previously published paper [16]. For the sake of completeness, however, a brief discussion is provided in the following paragraph.

Table 1 – Test matrix with the parameters varied.

Condition	Speed [rpm]	Load [bar BMEP]	Start of injection [CAD BTDC]	H_2 and N_2 mixture [% volume]
1	1500	2.5	3–12	4–12
2	1500	5	3–12	4–16
3	2500	2.5	3–12	4–8
4	2500	5	3–12	4–12

Table 2 – Intake charge composition.

Bottled $H_2 + N_2$ [%vol. of intake air]	Intake charge composition		
	[% N_2]	[% O_2]	[% H_2]
0	79	21	0
2 + 2	77.84	20.16	2
4 + 4	76.68	19.32	4
6 + 6	75.52	18.48	6
8 + 8	74.36	17.64	8

Nitrogen being an inert gas is not participating in the combustion process but effectively displaces oxygen lowering combustion temperature and thus NO_x emissions at the expense of CO, PM and THC. On the other side hydrogen's increased flame temperature contributes to NO_x formation, but its combustion products do not contain CO, PM and THC. Simultaneous H₂ + N₂ intake charge enrichment allows realisation (under certain operating conditions) of the benefits the two gases offer, namely simultaneous NO_x, BSN and CO emissions reduction. It is anticipated that H₂ + N₂ rich intake charge could lead to new advances in diesel engines with the reforming-based H₂ + N₂ serving as the enabling technology for developing new frontier emissions reducing engines.

Table 3 shows the percentage of energy released by the combustion of hydrogen at various H₂ + N₂ gas mixture fractions whereas Equation (1) details how it was calculated.

$$E_{H_2} = \frac{CV_{H_2} \times (\dot{V}_{H_{2i}} - \dot{V}_{H_{2e}}) \times \rho_{H_2}}{CV_{diesel} \times \dot{V}_{diesel} \times \rho_{diesel} + CV_{H_2} \times (\dot{V}_{H_{2i}} - \dot{V}_{H_{2e}}) \times \rho_{H_2}} \quad (1)$$

Where CV_{H₂} = hydrogen calorific value (kJ/kg), $\dot{V}_{H_{2i}}$ = hydrogen volume flow rate in the inlet (m³/s), $\dot{V}_{H_{2e}}$ = hydrogen volume flow rate in the exhaust (m³/s), ρ_{H_2} = hydrogen density (kg/m³), CV_{diesel} = diesel calorific value (kJ/kg), \dot{V}_{diesel} = diesel volume flow rate (m³/s), ρ_{diesel} = diesel density (kg/m³).

In order to maintain speed and load constant, at a given operating condition, diesel fuel was gradually reduced as H₂ + N₂ fraction increased.

4. Results

4.1. NO_x-BSN trade off

As already mentioned, N₂ rich intake charge lowers NO_x emissions at the expense of BSN, whereas H₂ enrichment typically causes the opposite effect. Simultaneous in-cylinder NO_x-BSN emissions suppression can be achieved from the combination of the two distinct gases. In this section, among others, it will be shown that addition of H₂ + N₂ gas mixture in the intake charge can break the trade-off effects of diesel engine, under low speed low load operation. Figs. 2 and 3 show the NO_x-BSN values when H₂ + N₂ gas mixture is supplied into the engine. It should be made clear that due to zero BSN values in Fig. 2(a), and in order to better illustrate the results, a secondary y axis was used.

Table 3 – Percentage of energy from the combustion of hydrogen.

H ₂ + N ₂ [%vol.]	Intake air [%vol.]	Energy from H ₂ [%] at 1500 rpm		Energy from H ₂ [%] at 2500 rpm	
		2.5 bar BMEP	5 bar BMEP	2.5 bar BMEP	5 bar BMEP
0	100	0	0	0	0
2 + 2	96	13.60	9.50	11.70	7.77
4 + 4	92	27.52	18.91	24.38	15.78
6 + 6	88	45.00	28.72	–	25.12
8 + 8	84	–	38.94	–	–

Fig. 2(a) reveals that up to 4% H₂ + N₂ admission causes only minor NO_x reductions. Possibly intake charge is not diluted to the degree that will considerably affect the formation of NO_x. Increase the volume of the admitted gas mixture beyond 4% results in substantial NO_x formation suppression. As presented in previous work [16] fuelling on H₂-diesel hardly affects NO_x emissions under low speed low load operation, therefore, the reduction of NO_x when H₂ + N₂ is used, (running the engine at the same operating condition) is mainly attributed to the reduction of intake charge's oxygen concentration. Comparison to neat diesel operation shows 71.5% reduction in NO_x formation when 12% H₂ + N₂ is supplied into the engine. It is worth mentioning that at 12% H₂ + N₂ (6% H₂ and 6% N₂) addition, in three out of four different injection events, slightly lower NO_x values were obtained compared to those collected at 6% separate N₂ [16]. The lower oxygen concentration in the former case (18.48% as compared to 19.74% in the latter) possibly compensated hydrogen's higher adiabatic combustion temperature which does not greatly affect NO_x under this particular operating condition.

As oxygen demand at this operating condition is not high, no adverse effects on CO were observed. The inherent lower carbon concentration of H₂ + N₂-diesel dual-fuel operation resulted in BSN reductions which become finally zero (or below the detection limit of the smoke meter). It should be noted that the BSN lines at SOI 3 and 6 CAD coincide.

Substantial BSN reduction was recorded at low speed medium load, Fig. 2(b). At the same operating condition, substituting up to 8% of inlet air with H₂ + N₂ increased NO_x emissions at a rate of 0.6–9%, depending on the injection timing chosen. Of interest is the 16% H₂ + N₂ addition – start of injection 12 CAD BTDC case where unexpected values, compared to current trends, were observed. NO_x reduction along with BSN increase may have been caused due to diesel misfire. When injection timing is advanced, diesel is injected at lower cylinder gas temperature and together with the lower diesel quantity and oxygen replacement may have led to poorer fuel oxidation which is reflected on emissions with higher BSN (unoxidised soot) and lower NO_x (lower temperature due to less fuel being utilised). The above speculation is confirmed by examining the in-cylinder pressure values shown in Fig. 7 as well as by the increased CO emissions (under the same test point) which are presented and discussed in Section 4.6.

Considerable reduction of soot and relatively unaffected NO_x emissions emerged at high speed low load operation, Fig. 3(a). Simultaneous NO_x-BSN reduction could have been achieved if the percentage of hydrogen and nitrogen supply was not equal. Less hydrogen supply (for instance either 6% N₂-2% H₂ or 5% N₂-3% H₂) would probably result in less NO_x breaking again the NO_x-BSN trade off.

Under high speed medium load operation NO_x emissions were found to be very sensitive when H₂ + N₂ percentage was over 8%. Below that gas mixture fraction an interesting behaviour was observed. When diesel injection was commenced on 12 or 9 CAD BTDC increasing the amount of H₂ + N₂ resulted in NO_x emissions increase. Retarding the injection timing on 6 or 3 CAD BTDC causes the production of less NO_x emissions, when H₂ + N₂ percentage is increased.

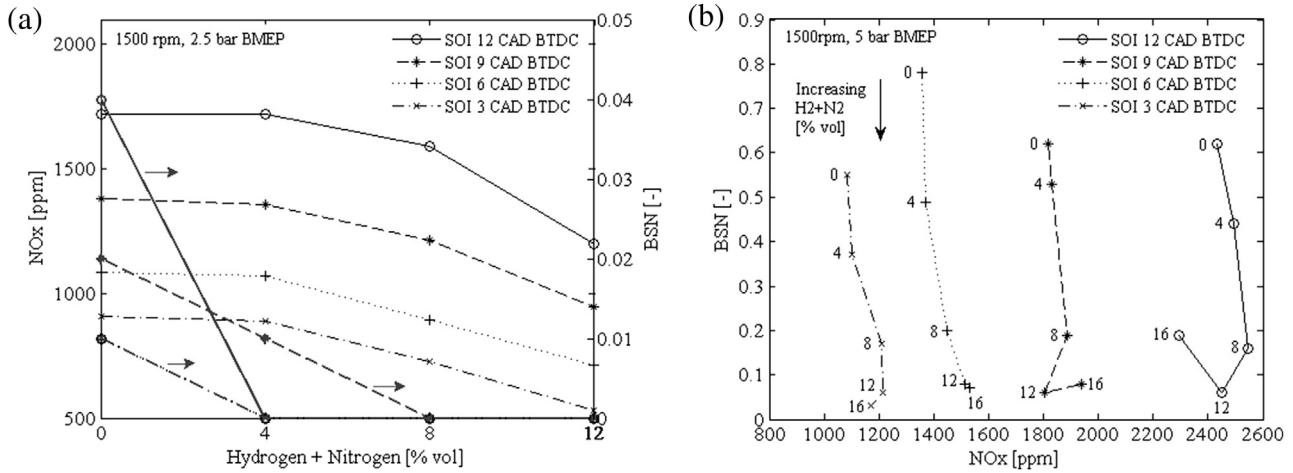


Fig. 2 – Effect of simultaneous H₂ + N₂ addition on NO_x-BSN trade off. Engine speed 1500 rpm, load: (a) 2.5 bar BMEP, (b) 5 bar BMEP.

Soot is reduced at almost all the points tested when gas mixture fraction is increased.

4.2. Combustion analysis

From Figs. 2 and 3 six points were selected in order to examine the effect of H₂ + N₂ on the in-cylinder pressure and rate of heat release (ROHR). The two figures on top correspond to 1500 rpm low and medium load (Fig. 4(a) and (b) respectively), while the figure on bottom shows data recorded during the 2500 rpm 5 bar BMEP run. The start of injection is the same, i.e. 6 CAD BTDC, at the three graphs illustrated in Fig. 4.

Looking now at each graph separately, Fig. 4(a) shows the case where increase of H₂ + N₂ fraction (from 8% to 12% H₂ + N₂) resulted in less NO_x emissions. The smaller premixed burn fraction under 12% H₂ + N₂ justifies the drop in NO_x emissions. Higher proportion of premixed burning often correlates with higher NO_x emissions [17]. Moreover, although more fuel was burned during the diffusion combustion phase under 12% H₂ + N₂ plus the combustion temperature was lower (reflected on lower NO_x values) BSN was maintained at zero levels which

indicates that the reduction of carbon/hydrogen ratio is the dominant BSN reduction factor at this operating condition. Comparison of the heat release when the introduction of different H₂ + N₂ fractions (12% and 16% H₂ + N₂) did not significantly affect NO_x and BSN values reveals a very good similarity between the two curves, Fig. 4(b). Finally a NO_x increase – BSN reduction case when the amount of gas mixture supplied into the engine was increased (from 8% to 12% H₂ + N₂) is indicated in Fig. 4(c). Again, higher NO_x are emitted from the combustion with the larger premixed burn fraction. The reduction of smoke emissions is attributed to the following reasons. Reduction of carbon/hydrogen ratio as hydrogen fraction is increased, better soot oxidation due to higher combustion temperature (reflected on increased NO_x), enhancement of soot oxidation from the production of OH radicals and less fuel burned late in the expansion cycle [18–21].

Fig. 5 depicts a 10%, 50% and 90% mass fraction burned (obtained from the integration of heat release rate) comparison for the cases shown in Fig. 4. The start of combustion, considered as the point the heat release curve intersects the x-axis, is also indicated in the graphs.

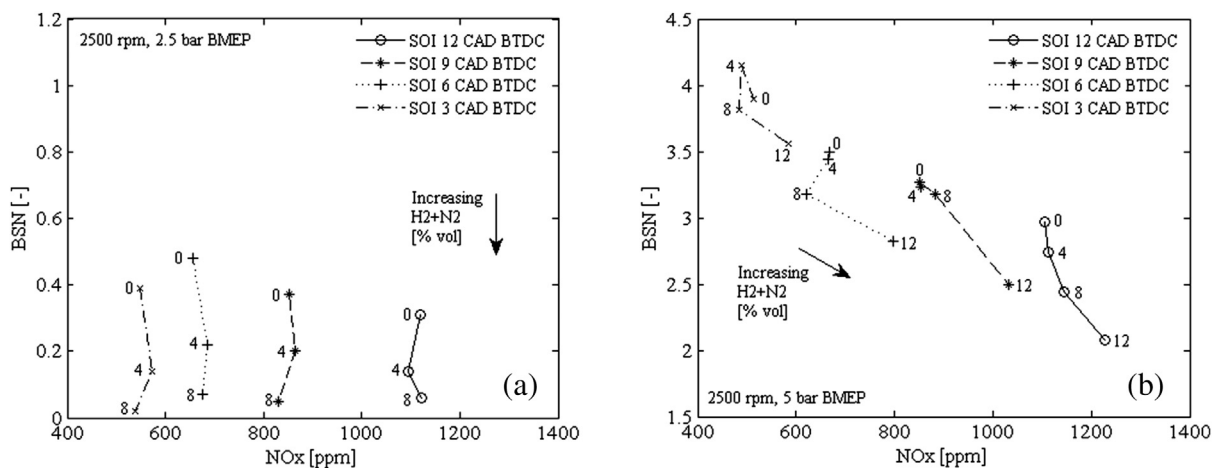


Fig. 3 – Effect of simultaneous H₂ + N₂ addition on NO_x-BSN trade off. Engine speed 2500 rpm, load: (a) 2.5 bar BMEP, (b) 5 bar BMEP.

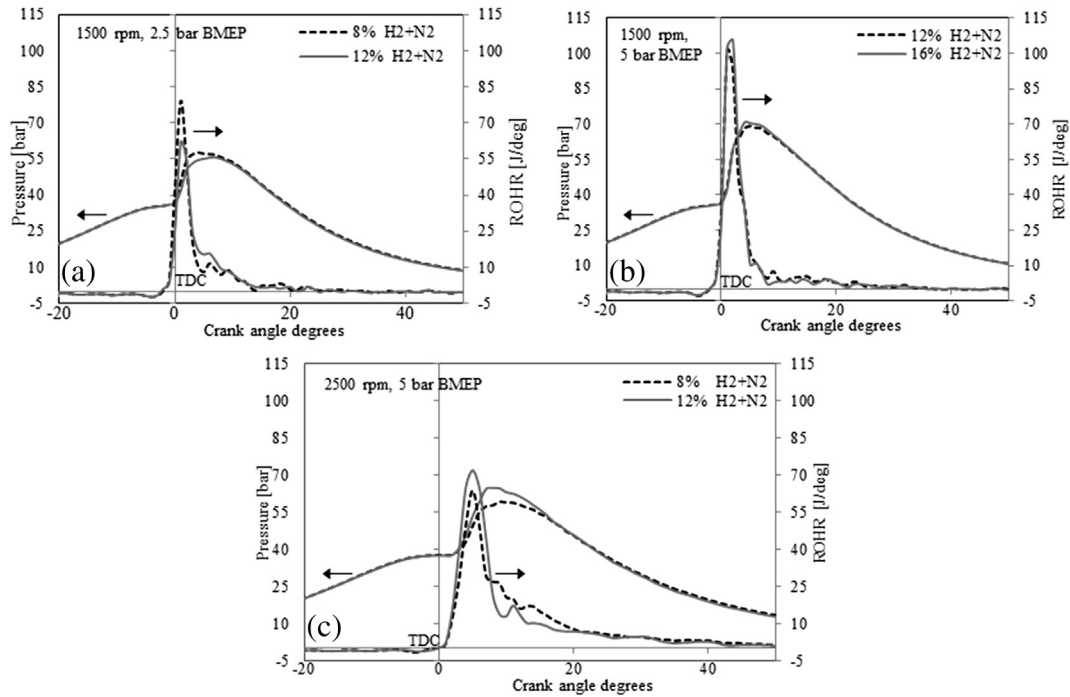


Fig. 4 – Effect of simultaneous H₂ + N₂ addition on the in-cylinder pressure and rate of heat release: (a) 1500 rpm, 2.5 bar BMEP; (b) 1500 rpm, 5 bar BMEP; (c) 2500 rpm, 5 bar BMEP.

Increase of the gas mixture fraction at low speed does not significantly affect the MFB patterns, as shown in Fig. 5(a) and (b). In Fig. 5(c) a quite noticeable change on combustion duration was observed. Particularly, the 50% and 90% MFB took place over a shorter period when H₂ + N₂ fraction was increased from 8 to 12%. Peirce et al. conducted experiments

on a compression ignition engine fuelled by diesel and bio-diesel and proved that the fuel which is burned faster is the most NO_x emitting one [22]. The same holds true when diesel is part substituted by hydrogen [23]. Referring back to Fig. 3(b) it is obvious that the 12% H₂ + N₂-diesel mixture generates higher NO_x emission. The increase of H₂ fraction may

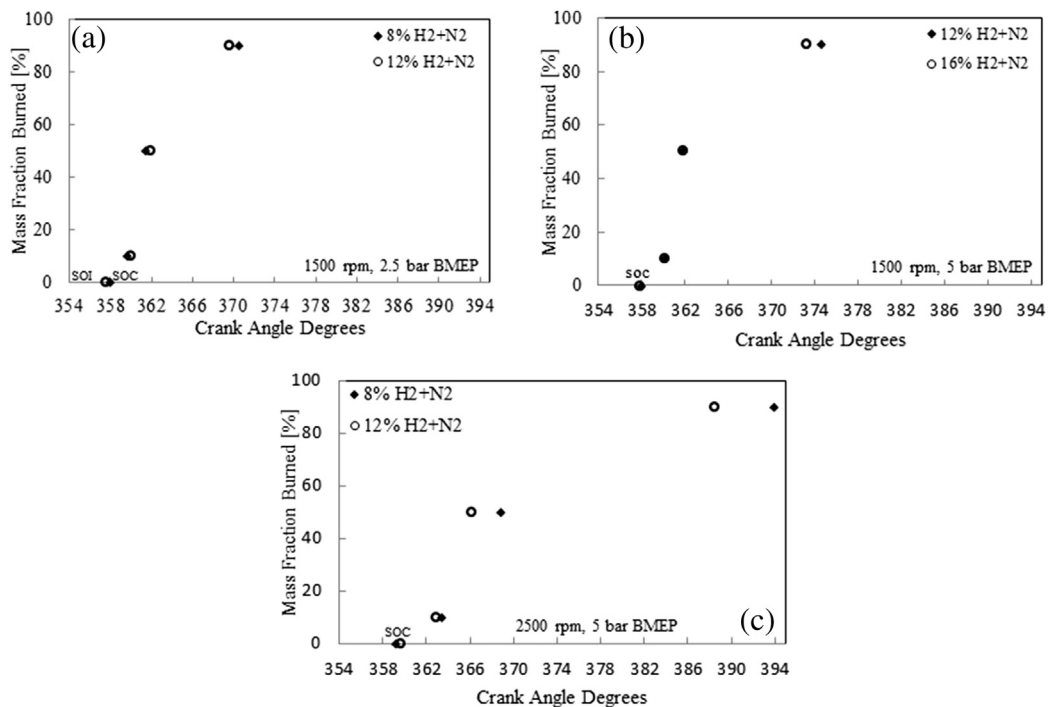


Fig. 5 – Comparison of mass fraction burned as a function of crank angle: (a) 1500 rpm, 2.5 bar BMEP; (b) 1500 rpm, 5 bar BMEP; (c) 2500 rpm, 5 bar BMEP.

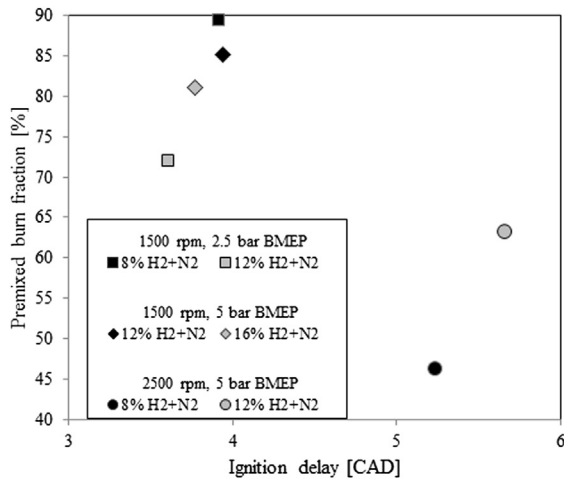


Fig. 6 – Premixed burn fraction as a function of ignition delay.

contribute to more explosive burn under this operating condition.

From the above it is obvious that enrichment of the intake charge with $H_2 + N_2$ does not always contribute to faster combustion which in turn would increase thermal efficiency as a result of a cycle closer to the ideal one. Also, dilution of intake charge along with incomplete hydrogen combustion has usually a detrimental effect on thermal efficiency, as presented in Section 4.7.

From Fig. 6 we may conclude that when the premixed burn fraction is over 70%, further dilution of intake charge with $H_2 + N_2$ is not necessarily associated with increased ignition delay. Examination of the high speed operation reveals the inherent diesel engine trend of increased ignition delay as intake charge is further diluted. In contrast to low speed, under high speed operation the premixed burn fraction is below 65%. From the above it could be deduced that dilution is not dominating the ignition delay when the premixed burn fraction is high enough.

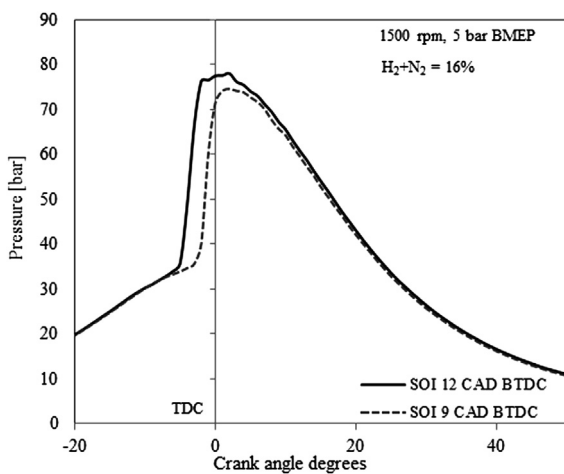


Fig. 7 – Comparison of in-cylinder pressure data. Engine speed 1500 rpm, load 5 bar BMEP, SOI 12 and 9 CAD BTDC, $H_2 + N_2$ vol. fraction 16%.

Fig. 7 shows the in-cylinder pressure data obtained while running the engine under the following operating conditions; low speed medium load, 16% gas mixture fraction, diesel injection commencement at 9 and 12 CAD BTDC. This figure verifies what has been assumed in Section 4.1 i.e. when diesel injection was commenced at 12 CAD BTDC the engine probably misfired. Comparison of the two curves (the dotted one corresponds to same engine conditions but different injection timing, 9 CAD BTDC) shows that the peak of the solid line is not as smooth as the dotted one. This abnormality implies an issue with the combustion which is reflected on the emission values, see Section 4.1 and 4.6.

4.3. Nitric oxide and nitrogen dioxide

The FTIR analyser is capable of measuring, among others, nitrogen exhaust components in the exhaust gas. In this paper apart from the regulated emissions a discussion on NO, NO_2 , N_2O and NH_3 emissions is also provided.

NO_2 in combustion is produced through the oxidation of NO by oxidative radicals and has been demonstrated that the type of fuel used and engine operating condition affects the conversion [19,24]. Moreover, the production of HO_2 radicals, occurring at low temperatures through H atoms reaction with O_2 enhances the conversion of NO to NO_2 [9].

From the four operating conditions tested the maximum NO_2 fraction was measured at low speed low load whereas the minimum at high speed medium load, Figs. 8 and 9 respectively. This is not unanticipated since increase of speed or load requires further fuel supply, lowering the A/F ratio and hence the excess oxygen concentration. An interesting point concerning the effect of fuel type on NO oxidation arises by examining Fig. 8. When $H_2 + N_2$, at any percentage, is supplied into the engine the NO_2 fraction becomes higher as compared to the neat diesel operation, although oxygen availability is lower due to air replacement. In particular, the fraction of NO_2 when the engine was running on neat diesel was found around 6% whereas under $H_2 + N_2$ rich intake charge that value varied from 11 up to 27%. It is worth mentioning that when the exhaust pipe is equipped with diesel oxidation catalyst (DOC) the NO_2/NO ratio may increase downstream of the catalyst [25].

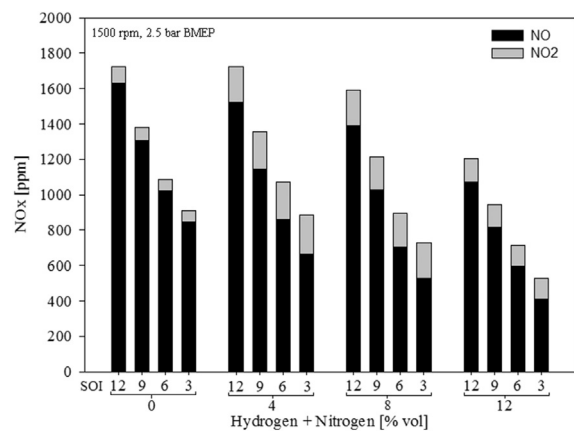


Fig. 8 – NO– NO_2 fraction. Engine speed 1500 rpm, load 2.5 bar BMEP.

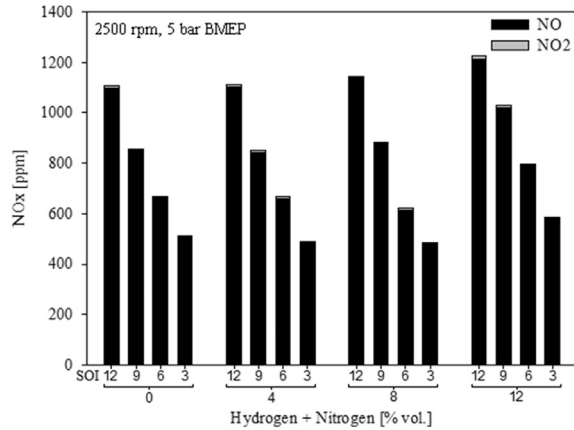


Fig. 9 – NO–NO₂ fraction. Engine speed 2500 rpm, load 5 bar BMEP.

4.4. Nitrous oxide

N₂O is a greenhouse gas and air pollutant and has been found that this chemical compound exists in automobiles' treated exhaust gases [26,27]. Fig. 10 illustrates that raw exhaust gases contain marginal N₂O emissions which, although low, tend to further reduce as H₂ + N₂ fraction is increased. Only the low speed low load operating condition is illustrated here as the same emission levels were observed at the rest of the operating conditions tested. Lipman and Delucchi [28] also reported zero N₂O emissions from cars without catalytic converter, however, the installation of catalytic converter may enhance nitrous oxide formation.

4.5. Ammonia emissions

Zero ammonia emissions were recorded at both neat diesel and H₂ + N₂ intake charge enrichment operation. In essence the values obtained from the FTIR analyser, indicated in Fig. 11, are slightly below zero and this has possibly been

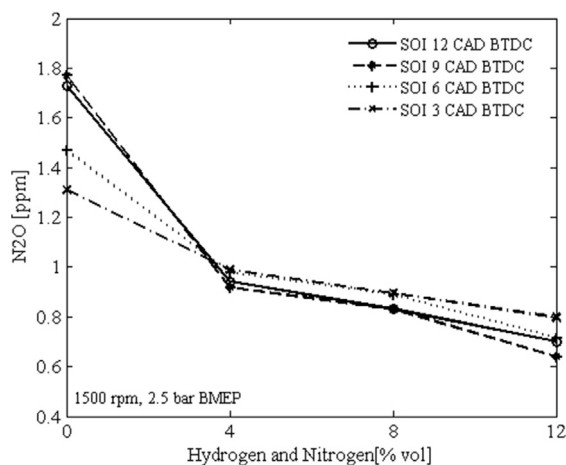


Fig. 10 – N₂O emissions. Engine speed 1500 rpm, load 2.5 bar BMEP.

caused by electrical noise. As with Fig. 10, only the measurements collected at low speed low load operation are presented as similar results were obtained at the rest operating conditions.

4.6. Carbon monoxide

Under low load operation (cond. 1 and 3) simultaneous H₂ + N₂ intake charge enrichment resulted in considerable CO emissions reduction. Comparison of neat diesel operation with the values obtained at the maximum H₂ + N₂ supply (12% at low speed and 8% at high speed operation) revealed up to 273% and 77% CO reduction at low and high speed, respectively. For the sake of brevity and due to the linear reduction of CO as H₂ + N₂ fraction increase, at the entire start of injection range tested, measurements are not presented in graphs.

It is interesting though to present the CO graphs obtained under medium load operation (cond. 2 and 4), Fig. 12. Under low speed medium load up to 8% H₂ + N₂ addition does not significantly affect CO emissions. Introduction of gas mixture over 8% increased the CO values when diesel injection was commenced at 9 CAD BTDC whereas at the rest injection timings tested CO emissions reduced apart from the 12 CAD BTDC – 16% gas mixture addition event. Based on the data presented in previous Sections (4.1 and 4.2) diesel misfire is probably the main reason of obtaining that unexpected value. As concerns high speed medium load operation 4% H₂ + N₂ addition decreased CO emissions when injection was commenced anywhere before 3 CAD BTDC. Addition of gas mixture over 4% increased CO emissions compared to the baseline values.

4.7. Brake thermal efficiency

A brake thermal efficiency comparison is illustrated in Figs. 13 and 14. Equation (2) presents the calculation process for the H₂ + N₂-diesel dual-fuel engine operation [29],

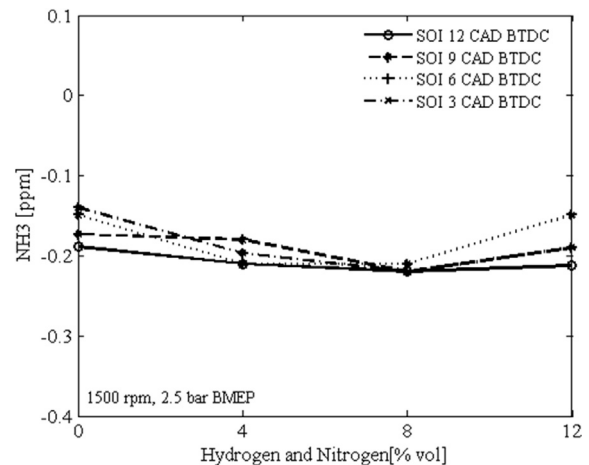


Fig. 11 – Ammonia emissions. Engine speed 1500 rpm, load 2.5 bar BMEP.

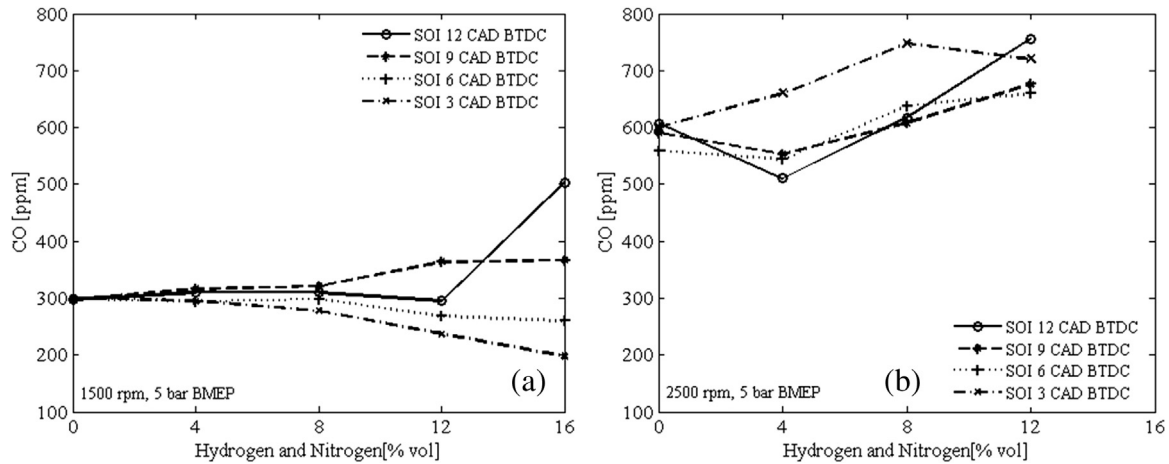


Fig. 12 – Effect of simultaneous $H_2 + N_2$ addition on carbon monoxide. Load: 5 bar BMEP, Engine speed: (a) 1500 rpm, (b) 2500 rpm.

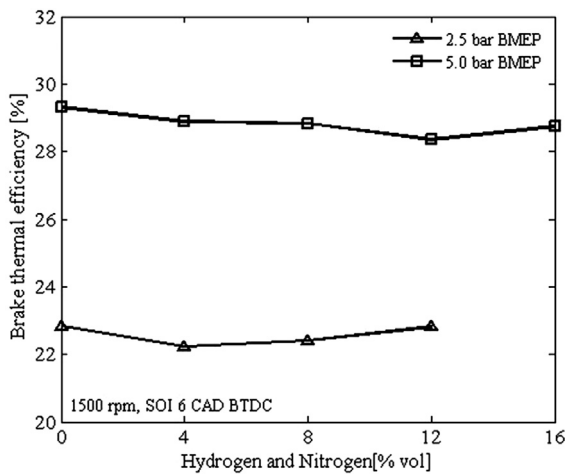


Fig. 13 – Brake thermal efficiency comparison. Engine speed 1500 rpm, load 2.5 and 5 bar BMEP, SOI 6 CAD BTDC.

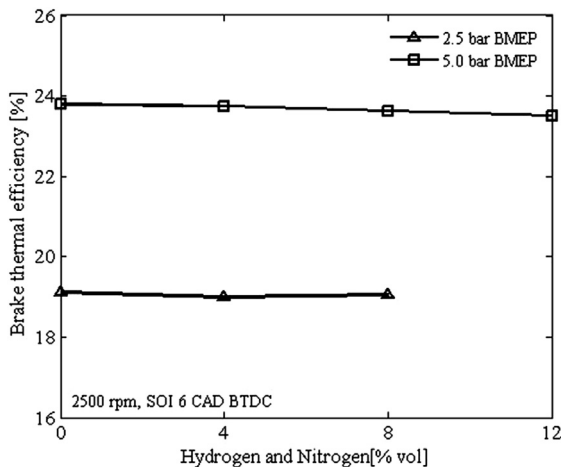


Fig. 14 – Brake thermal efficiency comparison. Engine speed 2500 rpm, load 2.5 and 5 bar BMEP, SOI 6 CAD BTDC.

$$\eta_{th} = \frac{\dot{W}_b}{\dot{m}_{df}LHV_{df} + \dot{m}_{H_2}LHV_{H_2}} \quad (2)$$

Where \dot{W}_b is the brake power, \dot{m}_{df} is the mass flow rate of diesel, \dot{m}_{H_2} is the mass flow rate of hydrogen, LHV_{df} is the lower heating value of diesel and LHV_{H_2} is the lower heating value of hydrogen.

The engine is more fuel efficient under low speed (1500 rpm) compared to high speed (2500 rpm) operation. Taking into account that thermal efficiency is improved as peak in-cylinder pressure occurs closer to TDC [30] and examining Fig. 4 it is obvious that in-cylinder pressure is better utilized during the power stroke under low speed operation. Previous study has revealed that over 98.7% of the hydrogen supplied into the engine is burned under high speed medium load operation [16]. Examination of Fig. 14 (medium load operation) shows a gradual thermal efficiency reduction as $H_2 + N_2$ concentration is increased. This implies that under this operating condition the major contributor in the reduction of thermal efficiency is the partly oxidised diesel fuel emerging from the dilution of the intake charge, since almost all the hydrogen supplied is consumed.

5. Conclusions

The influence of simultaneous $H_2 + N_2$ intake charge enrichment on the emissions and combustion of an HSDI diesel engine was studied and the main findings are presented below.

- NO_x -BSN trade off: Simultaneous NO_x and BSN emissions reduction was achieved under low speed low load run. Comparison to baseline operation shows a 71.5% reduction in NO_x emissions when 12% $H_2 + N_2$ gas mixture is supplied into the engine. Under both low speed medium load and high speed low load operation considerable BSN reductions with minor changes on NO_x emissions were recorded. Finally, under high speed medium load operation the rate of NO_x change appears to be very sensitive when

introducing over 8% H₂ + N₂ mixture. As concerns the NO_x-BSN values the optimum start of diesel injection is typically at 3CAD BTDC when the engine operates under conditions 1–3.

- Combustion analysis: Depending on the operating conditions, NO_x can be reduced, stay relatively unchanged or increased when the fraction of H₂ + N₂ mixture is increased. Reduction of premixed combustion is accompanied with lower NO_x emissions. When similar heat release curves are obtained (at two distinct H₂ + N₂ fractions) NO_x remain relatively unchanged. Rise of premixed combustion when increasing gas mixture fraction is accompanied with NO_x increase.
- Nitrogen exhaust components: When speed or load is increased the oxidation of NO is reduced as more oxygen is consumed in the combustion process. Under low speed low load operation, admission of H₂ + N₂ resulted in higher NO₂/NO ratio as compared to baseline operation. Raw exhaust gases contain marginal N₂O and zero NH₃ emissions. N₂O, although low, tend to further reduce as H₂ + N₂ fraction is increased.
- Carbon monoxide emissions: Under low load operating conditions (cond. 1 and 3) simultaneous H₂ + N₂ intake charge enrichment resulted in considerable CO emissions reduction. Relatively unaffected CO emissions emerged under low speed medium load run. As concerns high speed medium load operation, H₂ + N₂ addition over 4% produces increased CO emissions compared to baseline values.
- Brake thermal efficiency: H₂ + N₂ rich intake charge has a detrimental effect on brake thermal efficiency. In general, the engine is more fuel efficient under low speed (1500 rpm) compared to high speed (2500 rpm) operation. A factor contributing to better fuel economy under low speed – SOI 6 CAD BTDC is that peak cylinder pressure takes place closer to TDC.

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Nomenclature

A/F	air to fuel ratio
BMEP	brake mean effective pressure
BTDC	before top dead centre
CAD	crank angle degrees
DI	direct injection
DOC	diesel oxidation catalyst
DPNR	diesel particulate -NO _x reduction system
ECU	engine control unit
EGR	exhaust gas recirculation
FTIR	fourier transform infrared spectroscopy
HSDI	high speed direct injection
MFB	mass fraction burned

ROHR	rate of heat release
rpm	revolutions per minute
SCR	selective catalytic reduction
SOC	start of combustion
SOI	start of injection
TDC	top dead centre

Gases and fuels

H ₂	hydrogen
LPG	liquefied petroleum gas
N ₂	nitrogen
O ₂	oxygen
ULSD	ultra low sulphur diesel

Emissions

BSN	Bosch smoke number
CO	carbon monoxide
CO ₂	carbon dioxide
HO ₂	hydroperoxyl radical
NH ₃	ammonia
NO	nitric oxide
NO _x	nitrogen oxide
NO ₂	nitrogen dioxide
N ₂ O	nitrous oxide
OH	hydroxyl radical
PM	particulate matter
THC	total hydrocarbon emissions

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