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HYDROGEN-ENHANCED GASOLINE STRATIFIED COMBUSTION IN SI-DI ENGINES

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ABSTRACT

Experimental investigations were carried out to assess the use of hydrogen in a Gasoline Direct Injection (GDI) engine. Injection of small amounts of hydrogen (up to 27% on energy basis) in the intake port creates a reactive homogeneous background for the direct injection of gasoline in the cylinder. In this way, it is possible to operate the engine with high EGR rates and, in certain conditions, to delay the ignition timing as compared to standard GDI operation, in order to reduce NOx and HC emissions to very low levels and possibly soot emissions.

The results confirmed that high EGR rates can be achieved and NOx and HC emissions reduced, showed significant advantage in terms of combustion efficiency and gave unexpected results relative to the delaying of ignition, which only partly confirmed the expected behavior.

A realistic application would make use of hydrogen-containing reformer gas produced on board the vehicle, but safety restrictions did not allow using carbon monoxide in the test facility. Thus pure hydrogen was used for a best-case investigation. The expected difference in the use of the two gases is briefly discussed.

INTRODUCTION

One of the most promising long-term alternative fuels appears to be hydrogen. It is "clean" burning (the major combustion product being water), is easily ignited, and has wide flammability limits. Nevertheless, some important issues arise, such as on-board storage, safety concern, pre-ignition and backflash, combustion control, emission of NO_x, unburned H_2 and H_2O_2, power density and some more, not least lack of infrastructure for distribution. In the mid-term time frame, combustion of mixtures of hydrogen and gasoline appears to be a good opportunity to combine the major advantages given by both fuels. This would avoid many problems, especially if small amounts of hydrogen are produced on-board directly from gasoline by means of a reformer. Use of hydrogen and gasoline blends seems to be especially suitable for part load operation and reduction of emissions during cold start.

Interest about addition of hydrogen or reformer gas to conventional fuels has been growing over the last years. From large gas engines for stationary applications to passenger car engines, the investigation into this concept has been so far concentrated on spark-ignited homogeneous combustion engines. Our previous works gave a comprehensive characterization of the global engine behaviour and of the combustion process through detailed heat release analysis [1], of the capabilities of this concept in the homologation cycles [2] and of the influence of reformer gas on laminar and turbulent premixed flame speed in engine conditions [3].

Our own work stemmed from earlier investigations, which assessed from a global point of view the performance of a real prototype of reformer and showed promising results in terms of emissions [4][5][6]. Other investigations also showed the combustion characteristics of hydrogen or reformer gas blends with isooctane or other primary reference fuels in research burners or with simulation tools, mainly at standard conditions ([7], for example).

During our investigations into premixed combustion, the idea arose of looking at whether the concept was applicable also to the stratified combustion of gasoline in a direct-injection spark-ignition engine. The idea is that small amounts of hydrogen can be injected into the intake port to create a reactive background for the combustion of the gasoline injected into the cylinder. The expected effects were:

• Allow delaying the ignition timing as compared to standard gasoline operation, still having stable combustion;

• Allow the recirculation of large amounts of exhaust gas still having stable combustion.

Both measures are important to reduce the tendency of GDI engines to produce high quantities of NOx. In fact, they both make the combustion temperature decrease, the first by giving gasoline more

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time to evaporate and form a local mixture whose stoichiometry is leaner than the one giving the highest NOx formation ($\lambda \sim 1.1$); the second by limiting the temperature increase through the introduction of high heat capacity inert gas to absorb the combustion heat as well as by reducing the oxygen concentration in the oxidizer stream.

In normal gasoline operation, the possible ignition delay and the EGR rate are very limited, because they cause ignition and combustion stability to deteriorate. This results in a sudden increase of partial combustion products (unburnt hydrocarbons) and poor engine drivability. In principle, due to the wide flammability limits of hydrogen, both EGR rate and ignition delay can be increased while the combustion process remains stable. NOx emissions can therefore be reduced. Unburnt hydrocarbons are also significantly reduced and the brake efficiency increased. As side effect, also a reduction of the soot emissions can be expected, but this was not measured in our experiments.

EXPERIMENTAL SET UP

In this investigation, a single-cylinder research engine was used. The engine is a modification of a Mercedes Benz M111, whose specifications are listed in Table 1. The cylinder head (Figure 1) was originally equipped with 4 valves. For this application, one exhaust valve seat was shut. The spark plug was moved from the central position to a side position in the space freed by the shut valve, close to the gasoline injector, which was placed in the central position. The hollow-cone injector is from Siemens. The system is spray-guided. The spark plug is a NGK Iridium. This was chosen for its better performance as opposed to a conventional Bosch plug. The very thin electrode of the NGK plug is believed to offer a smaller surface for soot to deposit and compromise the performance of the ignition source.

Gasoline is injected at high pressure (80 bar) through a common rail injection system. For the injection of hydrogen in the intake port, two special injectors from Bosch and suitable drivers were used. Each injector was placed in one of the two channels leading to the two intake valves.

One engine speed (2000 rpm), two loads (3 and 5 bar IMEP – corresponding to slightly more than 2 bar and 4 bar BMEP) and different combinations of EGR rate and hydrogen additions were investigated. Measurements were carried out at pure gasoline operation, 17% and 27% hydrogen content (energy fraction). At 3 bar IMEP use of EGR was also investigated, up to 30% at 27% H_2.

Every measurement consisted of a sweep of the ignition timing with fixed injection end. The injection duration of gasoline and hydrogen was varied according to the desired load and the relative fraction of the two fuels. In particular, for every combination of hydrogen enrichment and EGR, the total amount of fuel was determined as to have always the same IMEP at the best ignition timing for IMEP. The amount of the two fuels was then kept constant during the ignition timing sweep. The injection timing of hydrogen in the intake port was set at -320°CA (during the intake stroke). The injection timing of gasoline was set to end injection always at about -55.6 °CA (electronic). The start of injection was varied depending on the amount of fuel to be injected.

The pressure indication was averaged over 290 cycles. The heat release rate was calculated using our internal code, which is based on a detailed 2-zone model.

In what follows the amount of hydrogen is always expressed as percent fraction of the total energy yielded by hydrogen.

Table 1: Engine specifications.

| Engine	Daimler Benz Aggregat Basis M 111 1-cylinder Stroke: 86.6 mm, Bore: 89.9 mm Compression Ratio: 10:1 2 intake valves, 1 exhaust valve
Injection	Intake channel injection from Bosch Direct injection from Siemens:(hollow-cone injector); Rail pressure: 80 bar
Ignition system	Coil from BMW, Transistor from Bosch Spark plug from NGK
Brake	ELIN, asynchronous motor/generator
EGR	Water-cooled, one-way valve, water trap
Oil and water conditioning	Oil- and water pump: external actuation electrical heating with water heat exchanger with monitoring elements for temperatures and pressures
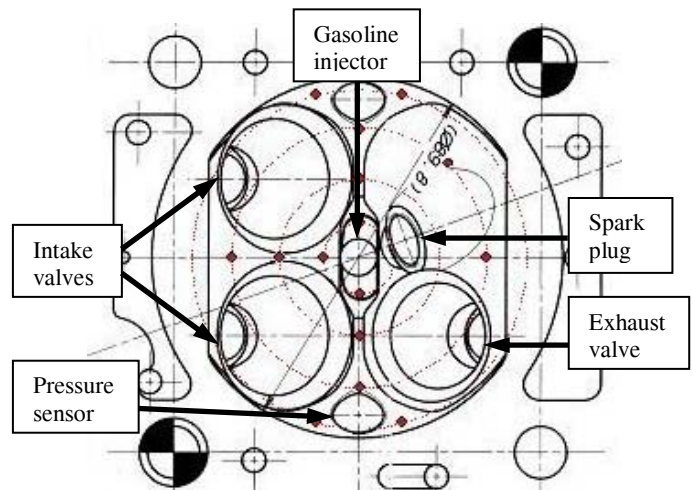
Figure 1: Cylinder head.

RESULTS

3 bar IMEP, without EGR

Figure 2 illustrates engine stability and power output versus ignition timing at pure gasoline operation and at 17% and 27% hydrogen addition. In general, hydrogen addition improves the stability of the combustion process. When the fraction of hydrogen in the fuel blend becomes larger, the optimal ignition point for best stability (lowest COV) lies earlier in the cycle. As expected, hydrogen addition widens the ignition window, but the effect is clearly stronger for early ignition timings than for late ignition. In all cases the deterioration of the combustion stability is very sharp for too early ignition and more gradual for late ignition.
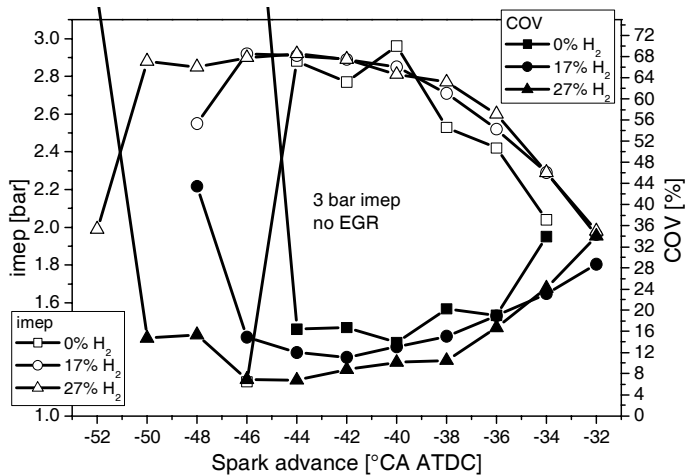
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Figure 2: IMEP and COV_IMEP versus ignition timing at gasoline operation, 17% and 27% hydrogen enrichment, without EGR.

It is in general known that addition of hydrogen to gasoline increases the flame speed and widens the flammability limits, but the effect is much stronger in rich mixtures than in lean mixtures. This means that rich mixtures between hydrogen and gasoline are more reactive and easily ignitable than lean mixtures. Figure 3 illustrates in a qualitative way how the mixture stoichiometry changes with time at the spark plug location during injection and evaporation of the gasoline spray. This kind of stoichiometry evolution was demonstrated through CFD simulation of a gasoline spray in the same GDI combustion chamber and the optimal ignition window was confirmed by the measured engine stability [8]. A gasoline/air mixture can be ignited in a narrow window around the optimal point, which is when $\phi \approx 1$ at the ignition source. The addition of hydrogen widens the stoichiometry window in which the mixture can be ignited, but mainly on the rich side. This is reflected in a widening of the ignition window more to earlier timing than to late timing. This agrees well with the results in Figure 2. In general the engine load changes the stoichiometry profile at the spark plug, because the curve tends to a higher value for large t, due to the larger amount of fuel for roughly the same air. This effect changes significantly the result in terms of extension of the ignition window, as it will be shown later.
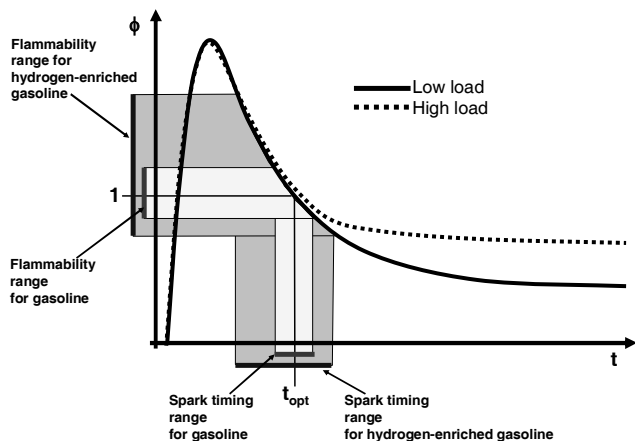

Figure 3: Qualitative evolution of the mixture stoichiometry with time at the ignition source, for gasoline and hydrogen addition, at low load.

Figure 4 illustrates how NOx and HC emissions vary with the ignition timing at pure gasoline operation and at two different degrees of enrichment. The emissions are normalized by the IMEP. HC emissions are generally lowered by hydrogen enrichment in the whole ignition window. This is due to better combustion stability and to the lower fraction of gasoline in the fuel mixture. Instead NOx emissions increase in the whole range, due to the generally higher combustion temperatures produced by hydrogen. The experience from homogeneous combustion of gasoline-reformer gas blends [1] suggests that use of reformer gas would give same or slightly lower NOx formation than at pure gasoline operation, because the reformer gas contains large amounts of inert N_2, whose thermal capacity limits the temperature increase due to hydrogen. The increase of NOx is particularly evident for early ignition, when a richer mixture is burnt. This confirms the previous observations regarding the stronger effect of hydrogen in rich mixtures. Also for late ignition timings hydrogen addition increases NOx formation as compared to pure gasoline operation. The little extension of the ignition window to late timings is not enough to bring down NOx to lower values than at gasoline operation. However the HC-NOx trade-off is clearly better that at pure gasoline operation. In fact, significant HC emission reduction can be achieved with H_2-enrichment at same NOx levels as at pure gasoline operation.
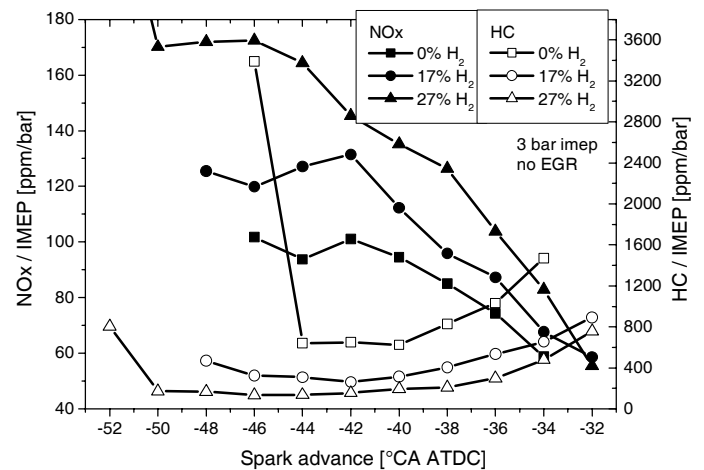

Figure 4: NOx and HC emissions versus ignition timing at 3 bar IMEP, without EGR.

Figure 5 shows NOx emissions versus engine stability, expressed as coefficient of variation of the IMEP (COV_IMEP). This allows comparing emissions when using hydrogen enrichment at the same level of stability as at pure gasoline operation. Although the present system is not a state-of-the-art GDI engine, it is clear that hydrogen addition improves the quality and repeatability of combustion, but at the cost of higher NOx emissions. This is true also if the ignition timing is delayed as much as possible, having as constraint that the same engine stability as at pure gasoline operation is met.

In synthesis, hydrogen enrichment *per se* increases NOx emissions. The widening of the ignition window is larger for early ignition than for late ignition. Late ignition is not enough, alone, to reduce NOx production from pure gasoline levels. HC emissions are lowered.
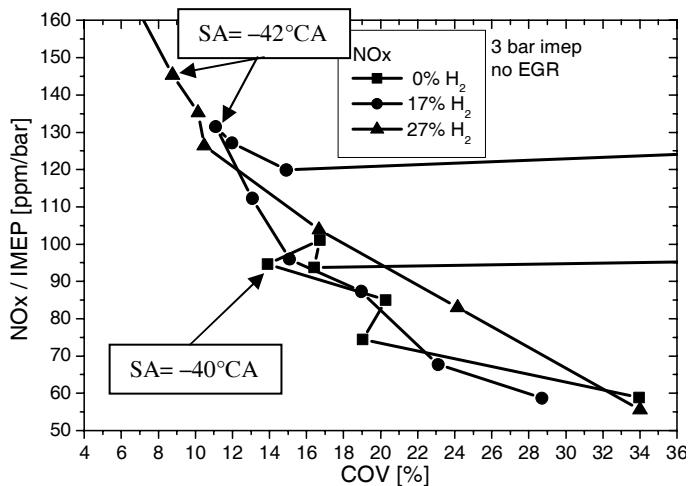
Figure 5: NOx emissions versus engine stability (COV_IMEP) at 3 bar IMEP, without EGR.

Exhaust gas recirculation at 3 bar IMEP

Hydrogen enrichment allows using higher EGR rates than at pure gasoline operation for the same engine stability. At pure gasoline operation, already a 10% EGR rate causes the combustion stability to deteriorate, but the power output is slightly higher, because of better efficiency. At 17% hydrogen operation, the deterioration of the combustion stability caused by EGR is smaller, and less so for even higher amount of hydrogen. This allowed reaching 20% EGR at 17% hydrogen operation and 30% EGR at 27% hydrogen operation. HC emissions are only slightly affected by the EGR rate increase, unless very high EGR rates (30% at 27% hydrogen) are used. On the other hand, 10% EGR is already very effective in reducing NOx formation and 30% EGR at 27% hydrogen operation can abate NOx by a factor of 5 as compared to pure gasoline without EGR. In Figure 6 the NOx emissions for all the strategies illustrated so far are plotted. Once again, it is clear that, at same EGR rate, hydrogen addition increases NOx formation as compared to pure gasoline operation and that the later ignition allowed by its use is never enough to compensate the higher NOx formation. On the other hand, the more hydrogen is used, the higher EGR rates can be achieved, with significant abatement of NOx production.
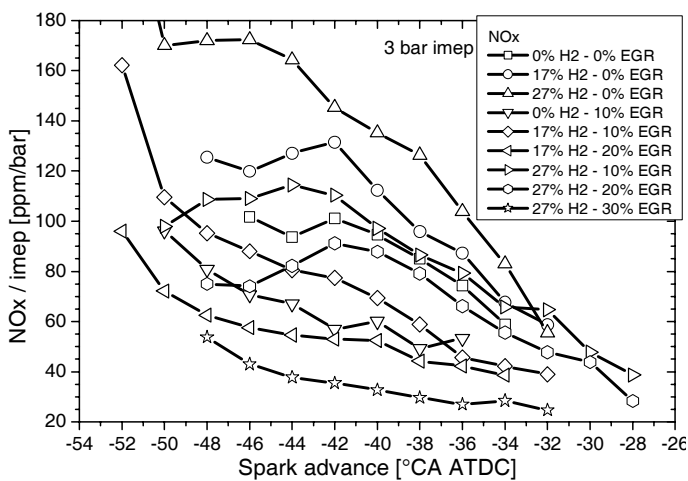

Figure 6: Comparison of NOx emission for different EGR rates and hydrogen enrichment degrees.

Figure 7 illustrates the trade-off between NOx and combustion stability, for some combinations of hydrogen enrichment and EGR rates. If the same engine stability as in the most stable point at pure gasoline operation is taken as target, NOx emissions can be reduced by adding hydrogen and at same time increasing EGR by a factor of 3 in the best case, which is 27% hydrogen enrichment with 30% EGR.

Figure 8 shows the trade-off between indicated efficiency and NOx emissions. The indicated efficiency, which is calculated as ratio between IMEP and fuel energy, increases noticeably with hydrogen addition. Also EGR increases the engine efficiency because the lower combustion temperatures reduce the thermal losses. When the 27% hydrogen/30% EGR operation point is compared to the standard gasoline operation without EGR, the advantage in terms of both efficiency and NOx is considerable.
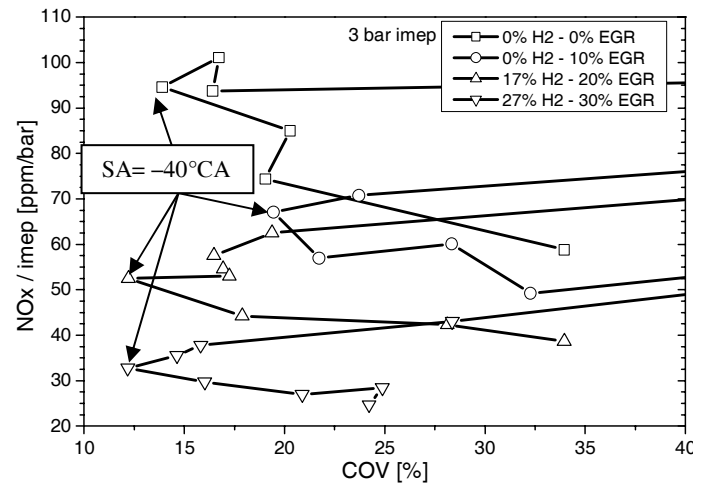

Figure 7: NOx-COV trade-off for some combinations of hydrogen enrichment and EGR rates.
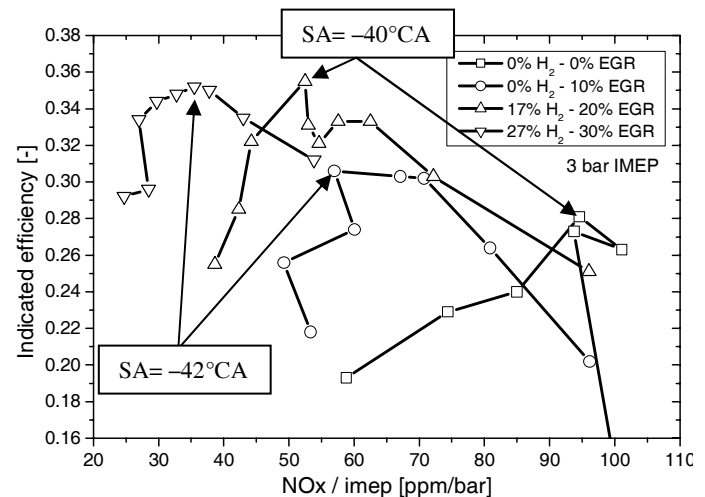

Figure 8: Trade-off between indicated efficiency and NOx for some combinations of hydrogen enrichment and EGR rates.

The extent of the efficiency increase given by hydrogen addition is certainly astonishing and requires some more discussion. At this regard, measurements were carried out with homogeneous charge (throttled, $\lambda=1$), at same load and level of hydrogen addition, to compare the efficiency increase. Table 2 summarizes the most

4

important results. First of all, it appears that, at homogeneous operation, hydrogen addition gives a much smaller efficiency increase, stemming from reduced pumping losses and shorter combustion, which might be compensated by higher heat losses due to higher combustion temperatures. The second important aspect is that the engine efficiency at pure gasoline homogeneous operation is higher than that at pure gasoline stratified operation, although the pumping losses are 4 times higher. This leads to the conclusion that the combustion quality of this system is not optimal at stratified operation.

Table 2: Comparison of stratified and homogeneous combustion, without EGR. For each strategy, the ignition timing giving the highest IMEP is selected.

type	H_2	PMEP	PMEP / IMEP	COV	Indicated efficiency	HC/CO_2	CO/CO_2
	[%]	[bar]	[%]	[%]	[−]	[%]	[%]
Strat.	0	-0.134	4.5	13.9	0.280	6.4	6.6
Strat.	17	-0.207	7.2	11.09	0.326	2.9	4.6
Strat.	27	-0.225	7.7	6.75	0.359	1.7	3.4
Homog.	0	-0.643	21.9	2.5	0.296	0.5	5.2
Homog.	28	-0.55	18.6	2.06	0.318	0.3	2.4

The efficiency increase given by hydrogen addition must therefore be ascribed to improved combustion quality to the largest extent. If one considers the ratio between HC emissions (measured as C_1) and CO_2 emissions, it appears that the fraction of fuel which is not or only partially burnt is rather large and can account for the missing energy, more so if one considers that part of the unburnt hydrocarbons is also oxidized in the exhaust tract and does not reach the gas analysis. The ratio between HC and CO_2 decreases with hydrogen addition and reaches a level comparable to homogeneous combustion. If one now compares stratified and homogeneous combustion with same hydrogen addition, the difference in efficiency is, as expected, directly linked to the decreased pumping losses (PMEP), as the combustion quality is now roughly comparable. In conclusion, this analysis reveals that the observed increase of combustion efficiency is to some extent specific to the GDI system used here and should not be generalized. Nevertheless it is clear that, besides the effect on NOx –which is the main focus of this investigation– small amounts of hydrogen have a striking effect on combustion quality and also in a modern engine could help to reduce HC and soot emissions remarkably.

3 bar IMEP - Combustion analysis

Figure 9 illustrates the rate of heat release at 3 bar IMEP, without EGR, for the ignition timing giving best stability. Also the heat release rate at homogeneous operation for the ignition timing giving best stability is plotted for comparison. All the curves are shifted to account for the different ignition timings. In the homogeneous case, hydrogen addition gives a faster start of combustion as compared to combustion of pure gasoline. In stratified combustion instead, the heat release rate in the first phase of combustion appears to be similar for all hydrogen enrichment degrees. This is because in stratified combustion the limiting factor is the evaporation of gasoline and the formation of an ignitable mixture with air and hydrogen. After combustion has started, the heat release rate is higher and combustion slightly faster with increasing H_2-enrichment. As it will be shown later, at higher load this behaviour is different, because hydrogen alone is already in an

ignitable stoichiometry with air, therefore the evaporation and mixing of gasoline is less a limiting factor.

Figure 10, Figure 11 and Figure 12 show the duration in crank angle degrees of the three main combustion phases, 0 to 5%, 5 to 50% and 50 to 90% of the total energy release (Integral of the Heat Release Rate, IHRR), at pure gasoline operation, 17% and 27% hydrogen enrichment, without EGR. In all cases delaying the ignition timing has a major effect on the first phase of combustion. The more the ignition is delayed, the more the liquid fuel has time to evaporate and mix with air. When the mixture is finally ignited, the combustion rate is higher, especially in the region closer to the ignition source, where the first 5% is burnt. If one compares how the duration of combustion changes with the hydrogen amount, it results that for increasing quantities of hydrogen the 0 to 5%, the 5 to 50% and the 50 to 90% become faster in the whole range of investigated spark timings, but the duration decrease is not large, especially from 0% to 17% H_2-enrichment degree.
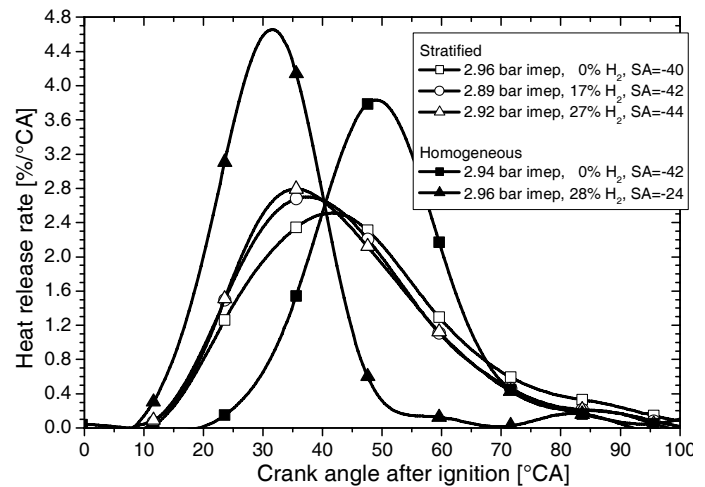

Figure 9: Heat release rate for stratified operation, without EGR. For comparison, also the heat release rate at homogeneous operation is plotted.
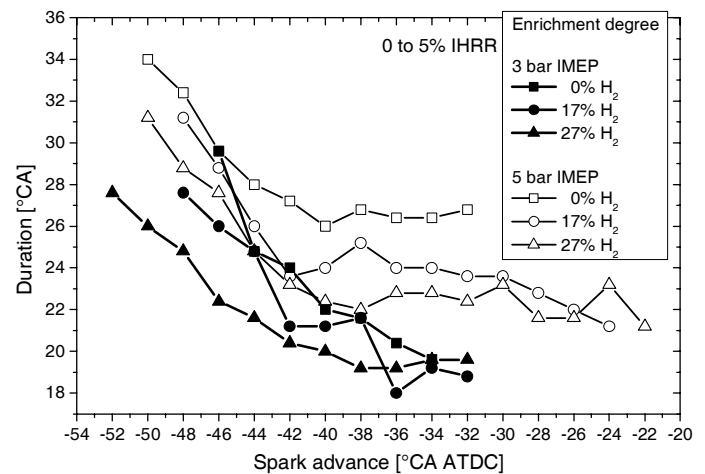

Figure 10: Duration of the 0-5% combustion phase at different degrees of hydrogen enrichment, without EGR.
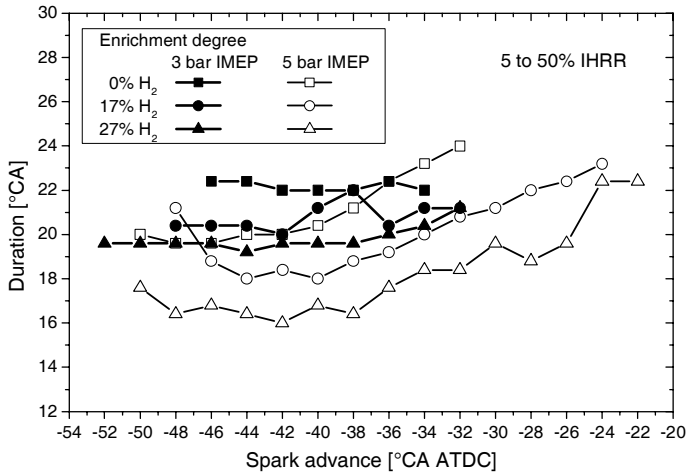
5

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Figure 11: Duration of the 5-50% combustion phase at different degrees of hydrogen enrichment, without EGR.
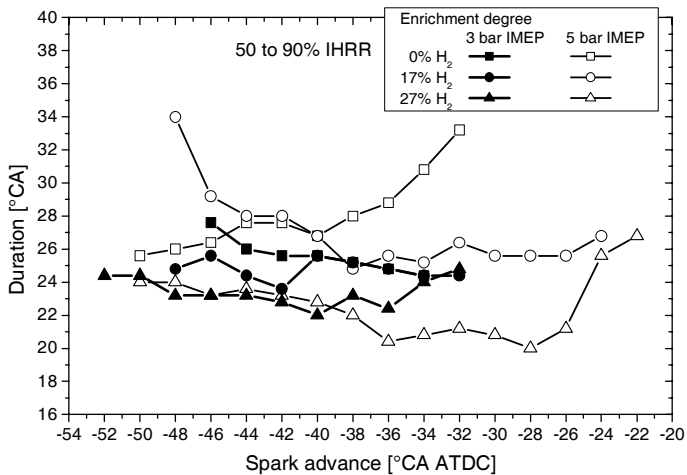

Figure 12: Duration of the 50-90% combustion phase at different degrees of hydrogen enrichment, without EGR.

When comparing the duration of the 0 to 5% energy release from pure gasoline to 27% hydrogen enrichment (Figure 10), an interesting behaviour appears: The higher the hydrogen enrichment degree, the less the combustion duration seems to be affected by the delaying of the ignition timing. For late ignition timing at 27% hydrogen, the curve tends to be flat. A possible explanation is that at higher hydrogen enrichment degrees, the quantity of injected gasoline is smaller and ignition can be delayed more. This means that the smaller amount of gasoline in the vicinity of the spark plug has more time to evaporate and mix up with air. The combustion process tends therefore to shift from less to more homogeneous. For early ignition the evaporation process is believed to be the controlling mechanism that determines the duration of combustion in the first 5% energy release, whereas for very late ignition the combustion speed is believed to become the main controlling factor, which is not influenced by the ignition timing.

5 bar IMEP, without EGR

Figure 13 illustrates how power output and engine stability vary with the spark advance at higher load (5 bar IMEP), without EGR. The measurement with pure gasoline was taken at a slightly higher load (~10% higher than the target, 5 bar IMEP). This explains why the

curve is slightly shifted to earlier ignition timings than the curves at 17 and 27% hydrogen. Taking this into account, it is possible to notice that, as already showed at lower load, hydrogen addition increases the engine stability at early ignition timings, but in this case the most striking effect is the stability improvement at late ignition timings, which was not observed at lower load. Figure 14 helps explaining the possible reason for this difference. At higher load the global fuel-to-air ratio ϕ is higher and this changes the profile of the stoichiometry at the spark plug location, in fact ϕ tends to a higher value for large t. The more hydrogen is used, the more the flammability range is extended, mainly on the rich side. For large amounts of hydrogen, the extension on the lean side though, tends to encompass completely the stoichiometry profile at the spark plug for large t, making possible to delay ignition to very late timings. In fact, when carrying out the experiments, it was clear that at higher load, as opposed to lower load, combustion of hydrogen was taking place also without injection of gasoline (although in this case no net power was delivered by the engine) because the stoichiometry was close or within the flammability limit of the hydrogen/air mixture.
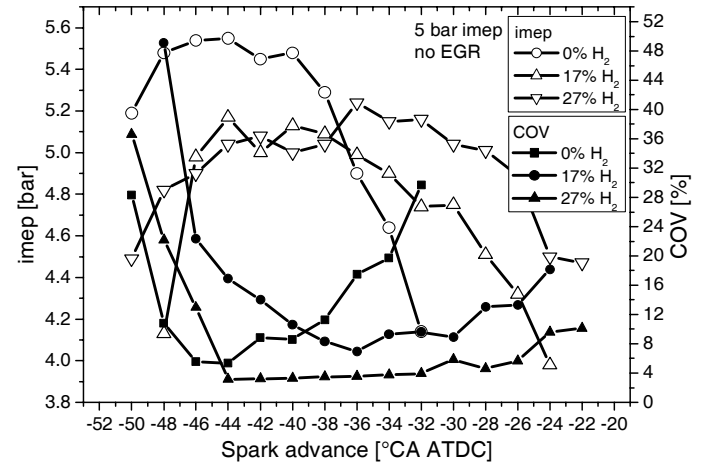

Figure 13: IMEP and COV versus ignition timing at gasoline operation, 17 and 27% hydrogen addition, without EGR, 5 bar IMEP.
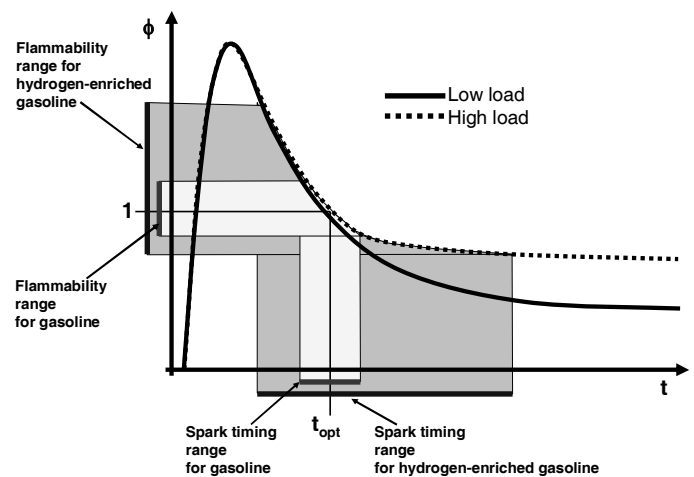

Figure 14: Qualitative evolution of the mixture stoichiometry with time at the ignition source, for gasoline and hydrogen addition, at high load.

6

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For what concerns emissions (Figure 15), at higher load as at lower load hydrogen enrichment significantly reduces HC emissions. In this case, when 27% hydrogen is used HC emissions remain very low even if the ignition is set to very late timings. In general, as at lower load, NOx emissions increase when hydrogen is used, but in this case the large ignition delay allowed by hydrogen addition makes possible to reduce NOx formation and keep it at same level as at pure gasoline operation. If one compares NOx emissions at same engine stability (Figure 16), it is clear that at high load a remarkable NOx reduction (~50% from the most stable point at pure gasoline operation) is possible even without using EGR. Use of EGR at high load can be reasonably expected to bring an even more striking reduction.
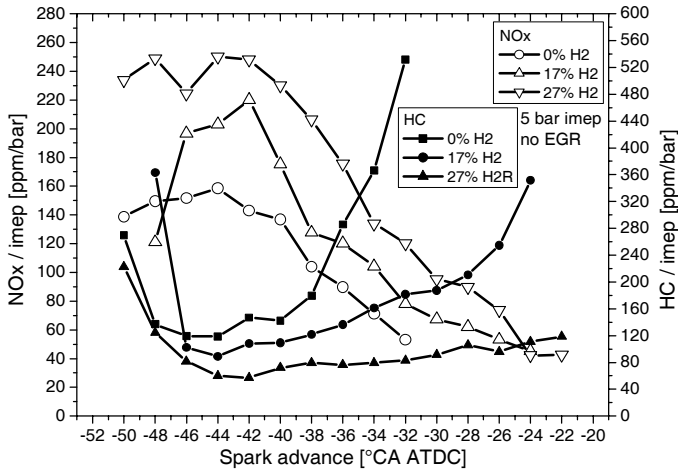

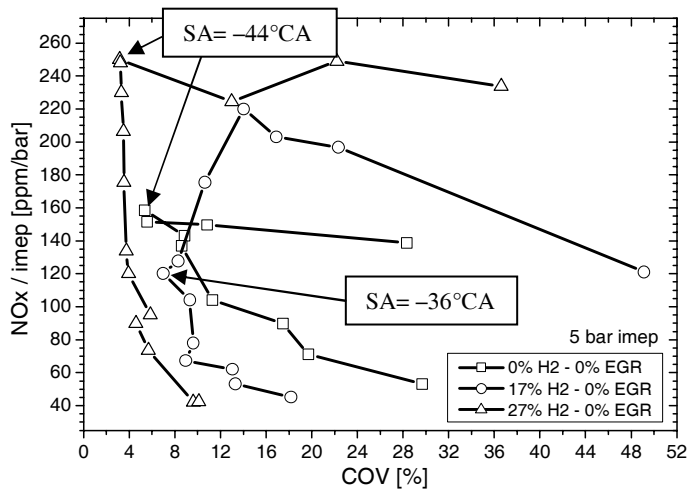**Figure 15: NOx and HC emissions. 5 bar IMEP, without EGR.**

Figure 16: NOx-COV trade-off at high load.

The efficiency increase given by hydrogen is evident in the whole range of ignition timings. For very early or very late ignition, it is due to better stability. Also for the best ignition timing, the increase of efficiency is remarkable. For instance, at -42°CA ignition, the indicated efficiency increases from 34% at pure gasoline operation to 36% at 17% hydrogen enrichment to 41% at 27% hydrogen enrichment. This is the result of better combustion stability, more complete combustion (as previously discussed regarding operation at low load) and shorter combustion, which in the considered point

decreases from 74°CA duration at pure gasoline operation to 62°CA at 27% hydrogen operation. Figure 17 illustrates the trade-off between indicated efficiency and NOx emissions. At same efficiency as in the best gasoline operation point NOx emissions can be curbed by a factor of 4 at 27% hydrogen operation from pure gasoline operation, just by delaying ignition, without using EGR.
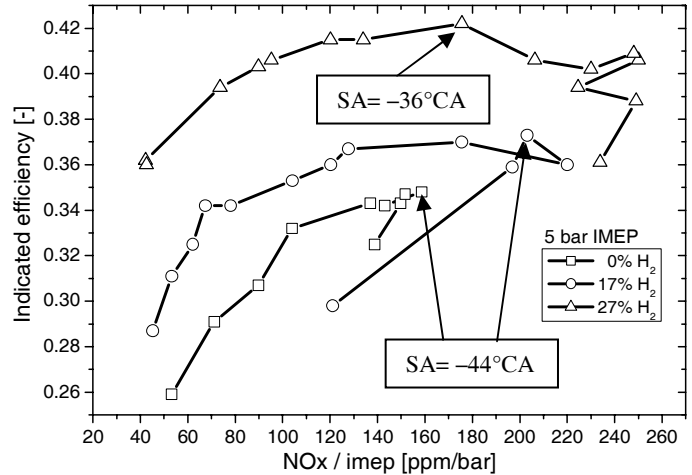

Figure 17: Indicated efficiency versus NOx emissions, for increasing hydrogen enrichment degree. 5 bar IMEP, without EGR.

5 bar IMEP - Combustion analysis

Figure 18 illustrates the rate of heat release. The curves are shifted to take into account the different ignition timings. A different behavior than at lower load can be observed in the way energy is released. In this case hydrogen addition produces a higher combustion rate since the beginning, because at higher load the global ϕ is higher and the hydrogen/air mixture is already ignitable or very close to it, therefore the evaporation of gasoline is less a limiting factor for the start of the combustion process. Figure 10, Figure 11 and Figure 12 illustrate the duration of the 0-5%, 5-50% and 50-90% phases. Similar trends as at lower load can be observed. In general hydrogen enrichment shortens the duration of the whole combustion process. In the 0-5% phase, as at lower load, delaying ignition makes the onset phase faster. As already explained, at high load it is possible to delay ignition significantly more than at low load. For very late ignition, the duration of the 0-5% phase seems to be less affected by the spark timing, if at all. This is due to the fact that the fuel has reached a higher degree of mixing with air; therefore the development of the combustion process is less limited by the spray evaporation. In the 5-50% phase, extreme ignition delays make the combustion duration increase, because the mixture has leaned out, causing slower flame propagation. In the final phase, 50-90%, where the mixture is anyhow very lean, the combustion duration for the three hydrogen enrichment degrees seems to be not much affected by the spark timing. The duration of the completion phase for early ignition at 17% hydrogen and for late ignition at 0% hydrogen are an exception, whose reason is not completely clear. The comparison of the combustion durations between 3 and 5 bar IMEP can be interpreted as follows: In general combustion in the 0-5% phase is rich. At high load the mixture is richer than at low load, giving slower flame speeds and longer duration. Combustion in the 5-50% phase is probably stoichiometric or lean, but at low load the mixture is leaner than at high load, giving slower flame speeds and longer duration. Finally, when the mixture is anyhow very lean, as in the 50-

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90% phase, there is no large difference in the flame speeds at low or high load, and the combustion durations lie in the same range.
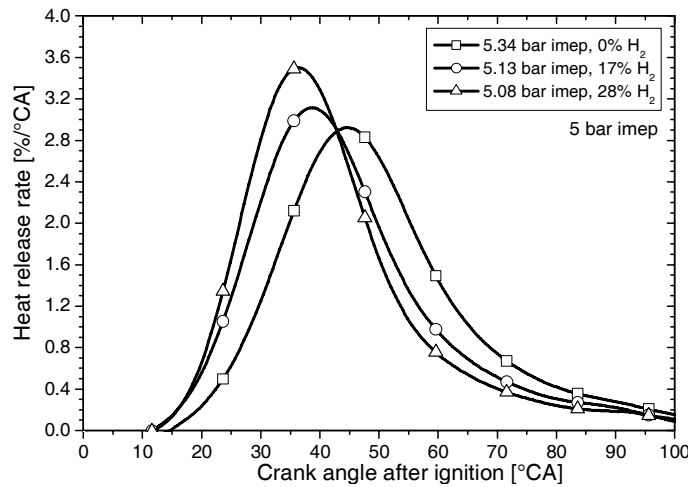

Figure 18: Heat release rate at 5 bar IMEP, without EGR.

CONCLUSIONS

The different combinations of global and local stoichiometry at the spark plug location give different results in terms of ignitability and NOx formation, depending on H_2 fraction, EGR rate and spark delay. Accounting for all the concurring effects is not trivial but can shed light into interesting phenomena.

At lower load (lower global ϕ) the small spark delay allowed by H_2 addition alone is not enough to compensate the higher NOx production. On the other hand, at lower load H_2 addition allows reaching EGR rates high enough to significantly curb NOx emissions.

At higher load (higher global ϕ) the large spark delay allowed by H_2 enrichment is able alone to limit substantially NOx production. Use of EGR at higher load is expected to give even better results.

The indicated efficiency increases with H_2 addition and EGR operation. This is due to the concurring effects of better engine stability, lower HC emissions and some faster combustion.

In all conditions, HC emissions are substantially lowered by hydrogen addition.

In general, H_2 addition gives extraordinary good results in terms of NOx, HC, efficiency and stability when considering the trade-offs among these quantities. At low load and with 30% EGR, NOx can be reduced by a factor of 3 at same engine stability as at pure gasoline operation.

Use of reformer gas instead of pure hydrogen can be expected to give lower NOx emissions, as the inert content would limit the temperature increase caused by hydrogen combustion. On the other hand, perhaps lower EGR rates could be achieved when using reformer gas instead of pure hydrogen. Also, use of reformer gas would produce a higher global ϕ, because of the inert gas displacing air, therefore an effect on ignition delay similar to what observed at higher load with pure hydrogen could be expected.

SYMBOLS AND ABBREVIATIONS

ATDC   After top dead center
B   Bore [mm]
BMEP   Brake mean effective pressure [bar]
°CA   Crank angle [°]
COV   Coefficient of variation of the IMEP [%]

CR   Compression ratio [-]
DI   Direct injection
EGR   Exhaust gas recirculation [%]
ϕ Fuel-to-air relative mass ratio, $\phi=1/\lambda$
GDI   Gasoline direct injection
HRR   Heat release rate [%/°CA]
ICE   Internal combustion engine
IHRR   Integral of the heat release rate [%]
λ   Air-to-fuel relative mass ratio:
        $(Air/fuel) / (air/fuel)_{stoichiometric} = 1 / \phi$ [−]
IMEP   Indicated mean effective pressure [bar]
LHV   Lower heating value [MJ/kg]
MBT   Maximum brake torque [Nm]
PMEP   Pumping mean effective pressure [bar]
RPM   Revolutions per minute
S   Stroke [mm]
SA   Spark advance [°CA]
TDC   Top dead center

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