Air Management of Heavy-Duty Hydrogen Engines

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Abstract: Hydrogen engines with supercharging and lean combustion are a highly suitable solution for sustainable mobility, specifically for high-duty applications. The properties of hydrogen as a fuel, and the characteristics of hydrogen combustion have an impact to the turbocharging system that needs to be considered.
The most important things to regard are the strong tendency of hydrogen towards combustion anomalies, and the specific dependency of the NO_x-formation on the air fuel ratio. Furthermore, the very low density of hydrogen needs to be taken into account.
All challenges are even aggravated during transient operation, where turbocharger lag becomes critical.

Key Words: Hydrogen; turbocharging; transient; combustion anomalies; NO_x

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1 Introduction

Hydrogen internal combustion engines may play an important role in the sustainable mobility of the future. Hydrogen as an energy carrier still possesses a higher storage density than batteries, and twice the mileage can be expected from hydrogen for most applications [1] [2] [3]. Hydrogen also has a significantly shorter refuelling time than batteries, being far more similar to fossil fuel refuelling which recommends it for high-duty (HD) applications.
Hydrogen (H_2) can be converted by a combustion engine or by a fuel cell. Internal combustion engines (ICE) are cheaper, more mature and evidentially resistant to fuel quality and harsh conditions in general. Fuel cells are more efficient and totally free of pollutant emissions.
However, the benefit of their efficiency is less pronounced than might be expected from the sheer sweet-spot figures. The gap narrows as the average load increases and eventually vanishes with high load applications [4].
We consider that the hydrogen combustion engine is a solution to be reckoned with. This assessment is underpinned by the evidence of a recent increase in the number of related activities in the international research community.
However, some problems still need to be faced, one of which is the turbocharging and transient operation.

2 Introduction to Hydrogen Engines

Hydrogen is an excellent fuel for internal combustion engines, but it has some characteristics that need to be considered (Table 1).

Table 1 Properties of Hydrogen, Methane and Diesel

Property	unit	**Hydrogen**	Methane	Diesel
Density @ ambient conditions[a]	kg/m³	***0,090***	*0,717*	*820 - 845*
Lower Calorific Value	MJ/kg	***120***	*50*	*42,6*
Stoichiometric Air Fuel Ratio	kg/kg	***34,3***	*17,2*	*14,5*
Mixture Calorific Value (PFI)[a,b]	MJ/m³	***3,19***	*3,39*	
Mixture Calorific Value (DI)[a,b]	MJ/m³	***4,52***	*3,75*	*3,86*
Minimum Ignition Energy[b,c]	mJ	***0,017***	*0,29*	*0,24*
Ignition Limits (λ [d])[a,c]	-	***0,13 - 10***	*0,6 - 2,1*	*0,48 - 1,35*

[a] 1.013 bar / 0°C
[b] at λ = 1
[c] in Air
[d] Air-fuel equivalence ratio

Hydrogen is gaseous under most conditions. In fact, hydrogen becomes liquid only below -253 °C, what might be used for cryogenic storage systems. Under ambient conditions the density is very low, only 90 g/m³.
Hydrogen has a very low ignition energy of 0.02 mJ, and wide flammability limits of 0.13 to 10, in terms of air-fuel equivalence ratio λ, both making hydrogen prone to combustion anomalies, i.e. unwanted ignition. In fact, a flammable hydrogen mixture can ignite virtually everywhere: in the intake manifold, in the exhaust manifold, in the cylinder before ignition (pre-ignition), in the crankcase. Avoiding of combustion anomalies is one of the most demanding aspect of hydrogen engine development. The effects and causes of combustion anomalies are not yet fully understood. It is known, that the spark ignition system, the residual gas in the cylinder, the air-fuel equivalence ratio, the design of the intake port (with port fuel injection) and oil droplets influence the occurrence of anomalies. The residual gas content, via the scavenging, needs to be considered when turbocharging is the subject.
Hydrogen has a very high heating value of 120 MJ/kg, and also a high stoichiometric air-fuel ratio of 34,3. This coincidence, and the low density under ambient temperature, leads to the effect, that the mixture calorific value differs greatly between external and internal mixture formation, with 3.2 MJ/m³ and 4.5 MJ/m³ respectively (@ $\lambda = 1$).
With hydrogen there are numerous possibilities for mixture formation and ignition. Similar to gasoline, hydrogen can be introduced directly (DI), or into the ports (port fuel injection PFI). When using direct injection it can be discriminated between early and late injection with homogeneous, stratified or even diesel-like mixture preparation. Hydrogen can be injected at ambient temperature, but also cryogenic injection has been investigated. The ignition of the hydrogen mixture can be realized with conventional spark ignition, but also with jet ignition (diesel), auto-ignition and glow-plug ignition.
However, latest research and literature reveals a converging of solutions. Almost all examples use spark ignition, due to its simplicity and robustness. Most examples use homogeneous charge, and injection is either in the ports (for retrofit systems and applications with less demand) or directly, with early timings. Early timings come along with good homogenisation and lower injection pressure. This paper, therefore, covers only these two injection variants.
The role of exhaust gas recirculation (EGR) on hydrogen engines is subject to ongoing discussion. It seems to be clear that the effect of EGR in terms of NO_x reduction cannot be compared to that of diesel combustion. This is mostly due to the homogeneous mixture. However, EGR might be used to decrease the pumping work in high power regions and to improve the combustion stability in high load operation [5]. Currently we are operating most of our engines (HD and LD) without EGR. This paper, therefore, covers only operation without recirculation.

3 Turbocharging under Steady-State Conditions

The kind of fuel injection influences the turbocharging (TC). If the hydrogen is injected into the intake ports (PFI), then it displaces the charge air and thus decreases the volumetric efficiency. While this is the case with all types of fuels, the effect is far more pronounced with hydrogen due to its very low density. Depending on λ, up to 45% of power density is lost by this effect compared to direct injection (Figure 1) [6].

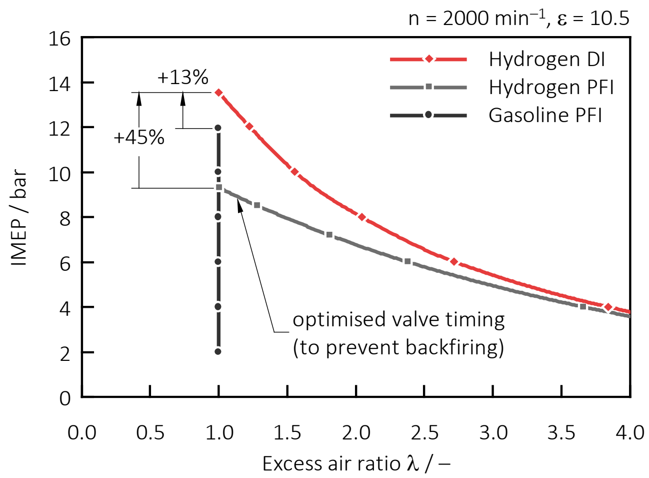

Figure 1 Indicated Mean Effective Pressure of hydrogen direct injection, hydrogen port fuel injection and gasoline PFI; naturally aspirated engine [6]

This effect is described in more detail in Figure 2. The diagram shows simulation results of a 13 lit. HD hydrogen engine. The full load curves are compared for direct hydrogen injection and port hydrogen injection. There are no other differences in the simulation models. The wastegate control of the turbocharger was used to meet a compressor pressure ratio of 3.8 for the entire speed range, the fuel amount was set to result in a λ of 1.7 for both the variants, and the TC efficiency is identical. We found that the BMEP of the engine with DI is 3 to 4 bar higher, while the boost pressure is equal. This can clearly be attributed to the air-displacing effect of the port fuel injection. The reduction of the air mass flow reflects the differences in BMEP. Quite interestingly, the BTE is very similar with slight advantages for PFI. Please note, that these 1D-simulations do not allow for combustion anomalies. The shown operation points may be subject to anomalies on a real engine.

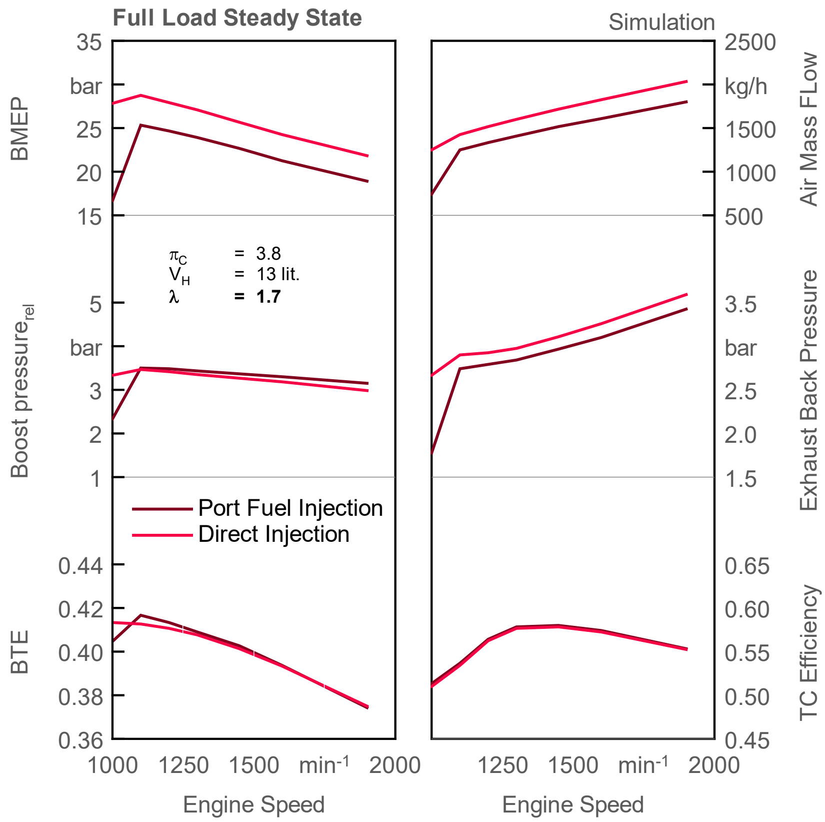

Figure 2 Full load curve of a H2 HD engine with DI and PFI; equal λ; 1D CFD simulation

Beside the effect of air-displacement, the origin of most turbocharging challenges is the NO_x formation of hydrogen engines. The NO_x formation depends on λ, as it is generally the case with any fuel. But this dependency is far more pronounced than that of diesel or gasoline engines. Figure 3 shows the engine out NO_x concentration over λ for a homogeneous hydrogen combustion. A maximum is reached around λ = 1.2. If leaner mixtures are used, the concentration drops, and it is remarkably low from λ = 2.4 and on, registering in the single-figure ppm range (see Figure 3). The origin of this characteristic correlation is the homogeneous mixture and the high flame velocity, compared to diesel and gasoline. Therefore, it is of utmost importance to maintain a high λ under any operating condition.

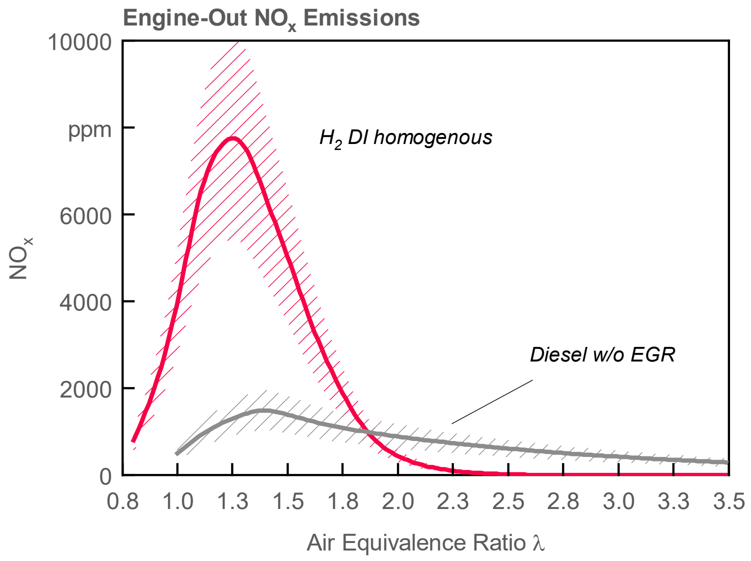

Figure 3 Typical NOx emissions of a hydrogen engine

The behaviour of NO_x emissions is slightly different between DI and PFI. For same air-fuel equivalence ratio λ the NO_x formation is less with PFI. This is accounted for by a less homogenized charge in the case of DI. If a certain NOx level is required, then a PFI engine can reach it with a richer mixture. An example is given in Figure 4. To reach a NO_x concentration of 1000 ppm, the PFI engine can use a mixture with λ = 1.7 while the DI engine has to be operated at λ = 1.9. This is a typical behaviour we observed on several engines, the exact figures clearly depend on the actual engine.

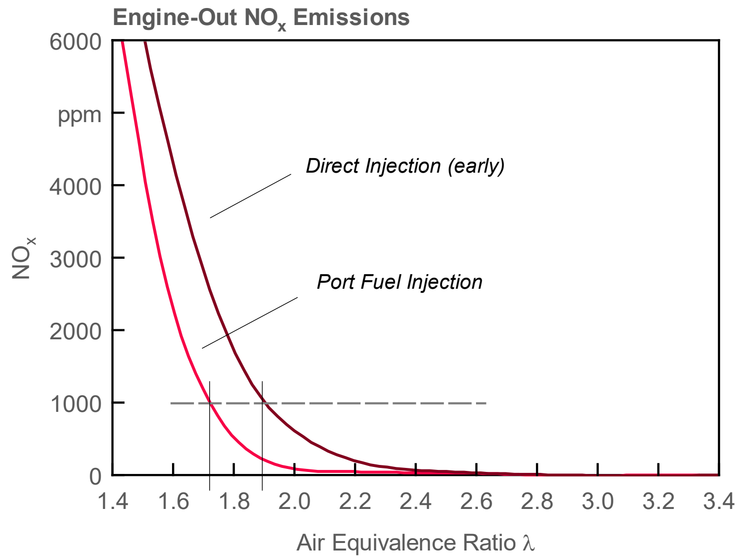

Figure 4 Typical NOx emissions of a hydrogen engine; PFI vs. DI

This effect of reduced NO_x emissions counteracts the effect of air-displacement, shown in Figure 2 to a certain extent. Figure 5 shows the same simulations again, but this time λ is set differently for both the variants. According to Figure 4, PFI is operated with λ = 1.7 while DI is operated with λ = 1.9. This should result in equal NO_x emissions.
Again, all other parameters are identical. The difference in the λ leads to an equalisation of the BMEP (that the BMEP is almost identical is a coincidence). The higher pumping work of the DI variant results in a slightly decreasing BTE.
Again, it has to be stated, that these 1D-simulations do not allow for combustion anomalies, because they originate from effects that are far beyond the used modelling resolution. A PFI engine suffers from combustion anomalies in the intake manifold (backfiring) with increasing load. This often limits the achievable full load curve. With DI engines there is, at least, no flammable mixture in the intake manifold. We estimate that the difference in BMEP can reach 2 - 3 bar at higher engine speeds, even with unlimited boost pressure.

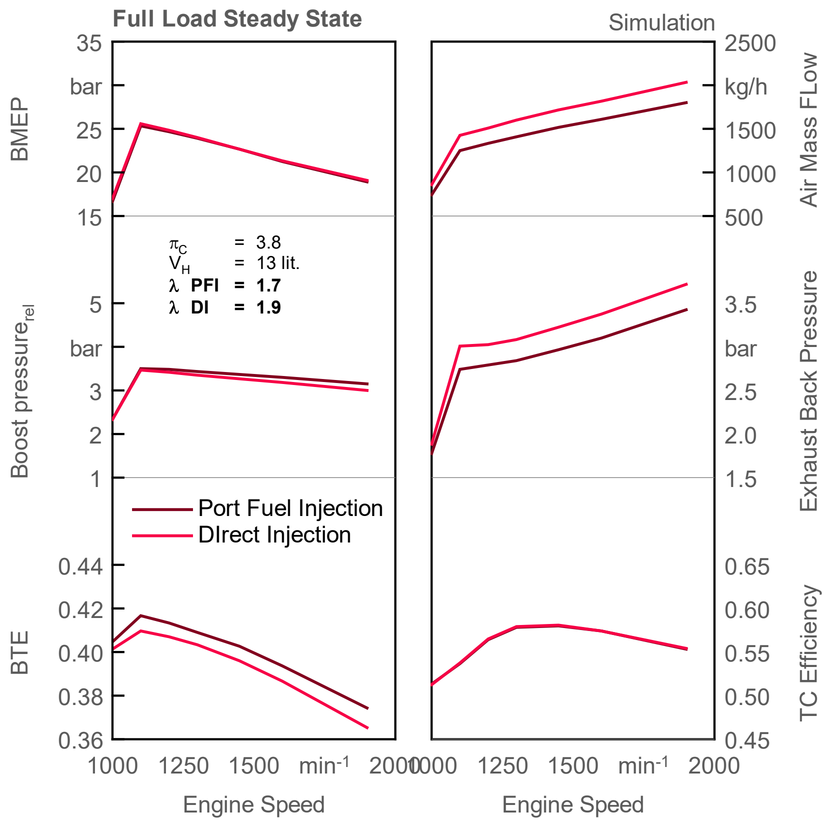

Figure 5 Full load curve of a H2 HD engine with DI and PFI; different λ; 1D CFD simulation

Hydrogen operation clearly has an impact on the turbocharger matching of an HD engine. The biggest difference compared to diesel engines is the exhaust gas recirculation, which is missing at the hydrogen engine.
Figure 6 shows a comparison between a hydrogen operated HD engine with direct injection and a diesel engine with a Euro VI calibration at 1000 min^{-1}

and 24 bar (full load). The diesel engine is operated with 26 % exhaust gas recirculation, while there is no EGR with the hydrogen engine. The air-fuel equivalence ratio λ of the hydrogen engine is set to 2.2, what let expect NO_x emissions of around 5 g/kWh. The λ of the diesel engine is 1.4 (with EGR). A λ of 2.2 requires an increase of the boost pressure on the hydrogen engine.
The higher boost pressure and the missing of EGR leads to a significantly increased mass flow across the compressor, in this load point by 40%. At the same time, the turbine swallowing capacity increases not as much, in this load point only by 25%. This is due to the higher boost pressure and the lower exhaust gas temperature. We observed this correlation on several engines of different sizes. The general conclusion is, that a hydrogen engine requires a larger turbocharger, but with a turbine not as large as the compressor (compared to diesel TCs), what is still challenging in terms of hardware availability for testbed work.

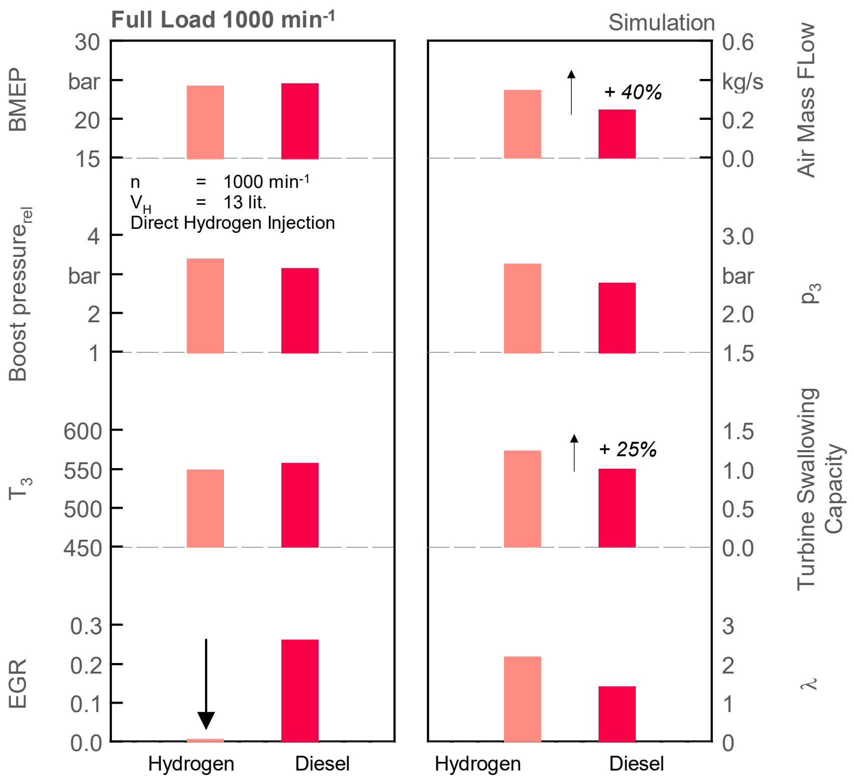

Figure 6 HD engine, Hydrogen vs. Diesel EuroVI operation; 1000 min^{-1} Full Load

However well the turbocharging system is matched, it has its limits, which will eventually define the NO_x formation of a hydrogen engine over load. As soon as the boost pressure cannot maintain the necessary λ with rising load the NO_x will greatly increase, due to the pronounced dependency on λ (Figure 7). This is typical of hydrogen engines, see Figure 7.

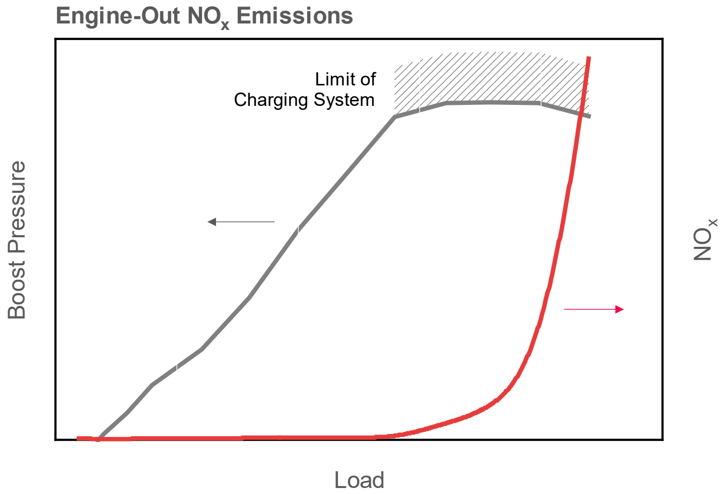

Figure 7 Typical NO_x Emissions of a turbocharged hydrogen engine over load; personal representation out of [7]

3.1 Excursus: Thermodynamic Simulation of Hydrogen Engines

Combustion anomalies, typical for hydrogen, are a serious problem for thermodynamic engine simulation (i.e. 1D-Simualtion). The occurrence of these anomalies can by no means be covered with the modelling resolution of 1D simulation. In a way, these anomalies are comparable to knocking, but they are far more complicated and less understood.

The approach in simulation is to deduce characteristic values from existing measurements on comparable engines (as comparable as possible) and to use these values as limits for the simulation. The definition of these values bases on the existing assumptions about the origin of anomalies and the important influencing factors. As an example, with PFI engines the air-fuel equivalence ratio in the intake port is the most important factor for backfiring.

We investigated a retrofit hydrogen combustion system with diesel jet ignition (i.e. dual fuel system) for a large bore truck engine. The aim was to use as much hydrogen but only as little diesel as possible (high substitution rate). The hydrogen is introduced into the ports (PFI) while the existing diesel injector was used to introduce the diesel directly into the cylinder. From earlier measurements on a HD single cylinder engine we deduced, that the minimal λ of hydrogen in the intake port ($\lambda_{H2,port}$) should not be less than 3 to prevent backfiring. The original map of the single cylinder measurements is shown in Figure 8. This map also shows the hydrogen share in terms of energy (substitution rate).

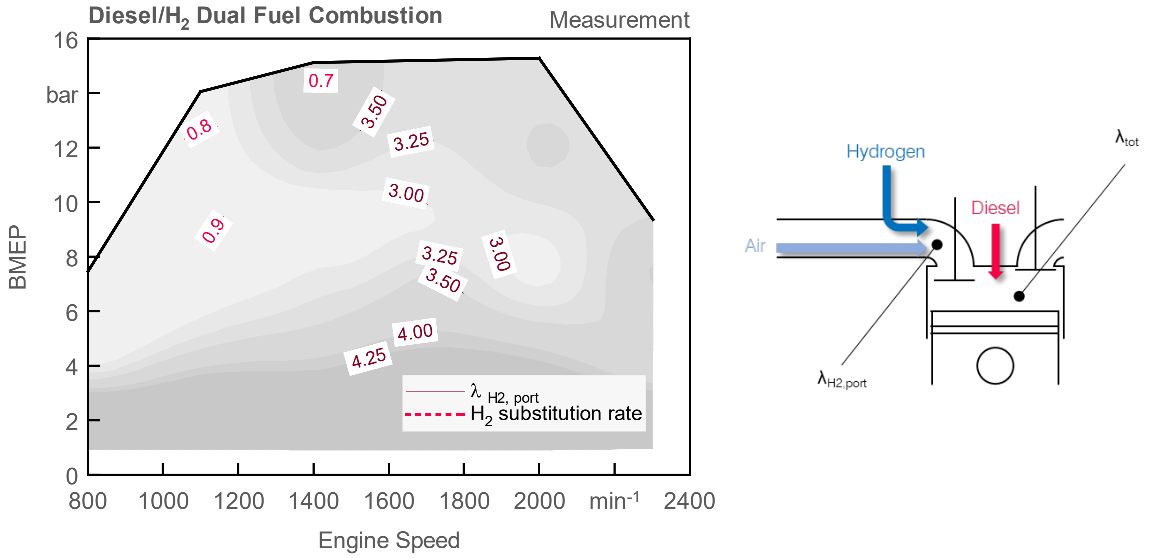

Figure 8 Lambda in intake manifold and hydrogen substitution rate of a H_2-diesel dual fuel SCRE; measurements

This lower limit for $\lambda_{H2,port}$ in the intake port was parametrised in the simulation model to define the hydrogen amount. The total required fuel energy was balanced with diesel injection. The hydrogen was injected into the intake ports, the engine was not modified otherwise (retrofit solution).
We found a substitution rate of 80% possible when assuming a minimum $\lambda_{H2,port}$ of 3 over the considered load range of 16 - 21 bar BMEP - see Figure 9. If a minimum $\lambda_{H2,port}$ of 2,5 were possible, than a substitution rate of 90 % could be reached. On the other hand, if only a $\lambda_{H2,port}$ of 3,5 or 4 were possible, then the substitution rate would drop to 70 % or 60 % respectively.

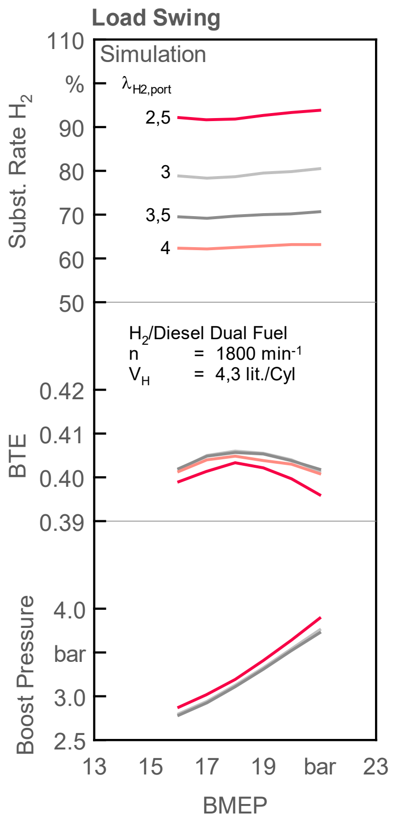

Figure 9 Hydrogen Substitution rate of a Dual Fuel HD engine depending on the minimal intake port $\lambda_{H2,port}$; 1D Simulation

The key message of this excursus is the following: In the meantime we are, by continuous advancement, able to operate our test engines with a $\lambda_{H2,port}$ of 2. The assumptions made in the simulation model are therefore no longer valid and the results are outdated. It has to be concluded, that a reliable thermodynamic simulation in advance is still barely possible due to the effects of combustion anomalies.

4 Turbocharging Under Transient Conditions

The problem of turbocharging is even aggravated during transient load steps. Equal to every other turbocharged engine, the charger requires a certain time to gain speed and to deliver the desired boost pressure for the new (higher) load point. During this time the engine might be operated with a λ below the desired target λ. With diesel engines, this can result in excessive emission of particulate matter, and that is why the injected fuel amount is limited to maintain a certain λ ('smoke limiter').
With hydrogen engines it is clearly the NO_x that is a problem in transient operation. A drop of the λ during the load step can result in exorbitant NO_x formation, due to the pronounced dependency of NO_x on the λ.
Figure 10 shows a load step from 1 to 18 bar BMEP at an engine speed of 1000 min^{-1} of a HD engine. This 13 lit. engine has a VNT single stage turbocharger and is operated without EGR. The lower limit for the air-fuel

equivalence ratio during λ the load step is varied between 1,3 and 2,1. The load response (BMEP) is clearly enhanced with richer mixtures. This has two causes. One the one hand, more hydrogen can be introduced for the same amount of charge air. On the other hand, higher exhaust temperature increases the available energy at the turbine.
The drawback of enrichment are, obviously, the NO_x emissions. In accordance to the correlation given Figure 3, the concentrations rise sharply with the enrichment. With a λ = 1,3 the NO_x concentration reaches 6000 ppm peak, what is hugely more than figures known from diesel engines. The small plot indicates that this follows the general NO_x-λ correlation.

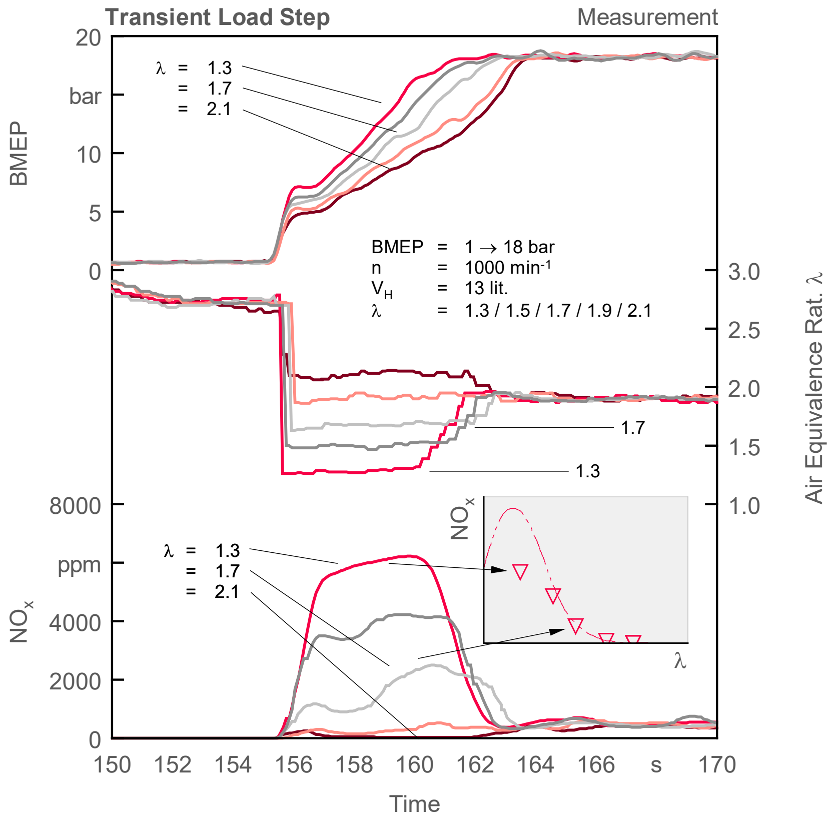

Figure 10 Load steps with varying lower limit of air-fuel equivalence ratio λ

During the load step, the VNT needs to be closed to gain energy for the turbine. This instantly increases the exhaust back pressure, while the boost pressure lags behind (the steep rise of the pressures in the very first moment of the load step is due to the opening of the throttle). The result is an increasing residual gas content in the cylinder. As already described in Chapter 2, the residual gas content is critical in terms of pre-ignition. During the load step with a minimum λ of 1,3 two pre-ignitions occurred in the later part of the load steps, recognizable by the very early position of the combustion (50% of fuel burned - $MFB_{50\%}$). Similar to knocking, Pre-ignitions result in higher cylinder pressures that harm the engine. Pre-ignitions must be avoided by all means.

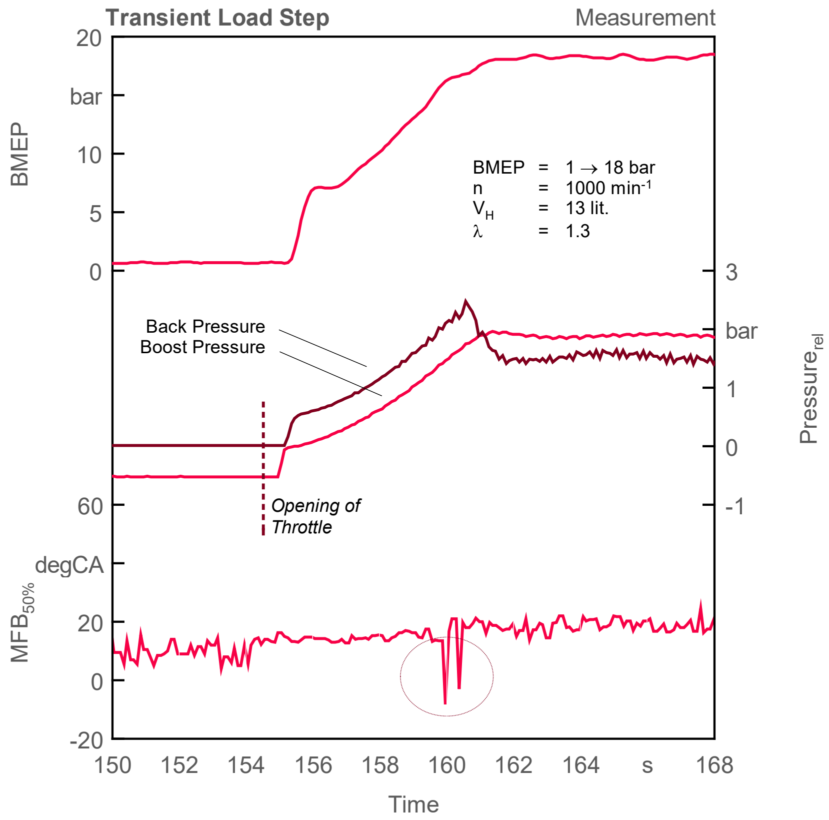

Figure 11 Boost control during load step and its effect on combustion anomalies

We investigated the correlation between response time and NO_x formation in more detail on a 2 lit. passenger car engine [8]. A representative load step was deduced from the WLTC cycle for a D-Segment car. It starts from 1000 min^{-1} and 2 bar and ends at 1500 min^{-1} and 14 bar, with a ramp-up time of 1 second. This was found to be the second most demanding load step in the WLTC, due to the almost totally missing exhaust enthalpy in the beginning (only a load step from idle operation was more demanding). We found a clear trade-off between cumulated NO_x mass and response time (98% of target torque), and a lower limit for both the parameters. For the final calibration a minimum λ of 1,8 was chosen.

In one aspect hydrogen combustion is easier to handle than diesel, and that is the retardation of the combustion (the ignition) during the load step. While this is clearly not possible with diesel due to the formation of particles, it helps to decrease the NO_x formation and, at the same time, increases the enthalpy for the turbine [8].

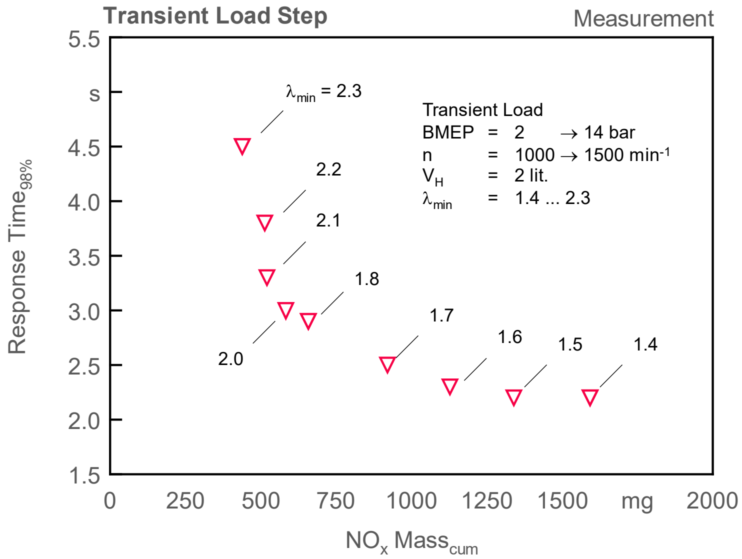

Figure 12 Response - NO_x Trade Off during transient load step on a Hydrogen LD engine; personal representation out of [8]

The NO_x formation during load steps and the pronounced dependency on the steady-state load (Figure 7) can lead to the characteristic, that during a transient cycle the NO_x occur virtually solely during accelerations. This is particularly true for highly transient cycles with low average load, as they are typical for lighter vehicles.

5 Emissions

Clearly, NO_x is the emission of highest concern on a hydrogen engine. It is possible to reach very low engine out emissions with a lean combustion, but we consider an aftertreatment system to be necessary nevertheless, due to questions of degradation over lifetime but also due to the acceptance on a future market strongly driven by environmental viewpoints. However, NO_x can be regarded as settled, also under assumption of very stringent future limits [7]. There is also unburned hydrogen in relevant concentrations, but it can be converted easily in a downstream oxidation catalyst.

There are other emissions that do not immediately cross the mind when talking about hydrogen engines. These are hydrocarbons (HC), carbon monoxide (CO) and particles. They originate only from the combustion of lubricant. The lubricant originates from the cylinder liner and the turbocharger [9] [10]. The turbocharger causes 5 to 40% [11] of the total oil consumption, strongly depending on the overall wear of the engine.
While HC and CO can be neglected under normal conditions (and would be easily converted in a downstream oxidation catalyst), particle number can exceed the current limits under certain conditions. We observed this behaviour on several engines of different sizes and primarily under transient cold

start conditions. The conclusion is, that the turbocharger is responsible for a significant share of the overall particulate number on a hydrogen engine.

6 Outlook

The NO_x formation of a homogeneous hydrogen combustion remains a key issue for R&D. The fact, that almost the entire NO_x is formed during transient and high load operation, suggest several possible solutions.
As far as the charging system is concerned, an electrical booster, or an electrical assisted TC promises a high potential. It can overcome the gap in boost pressure during run-up of the TC.
The injection of water might be a possibility to bring down the NOx during phases of rich combustion.
The phlegmatisation of the engine by using a hybrid powertrain will also help to decrease NO_x emissions. Hybridisation does not only help to decrease the NO_x during transients, but it also allows to avoid steady-state high load operation by smoothing the torque demand on a large time scale. With first measurements we were able to show a huge potential, also for demanding transient cycles of non-road mobile machinery - see Figure 13. Compared to the conventional powertrain the NOx are virtually zero and barely visible in the plot (in fact, they are ranging between 0,02 and 0,1 g/kWh).

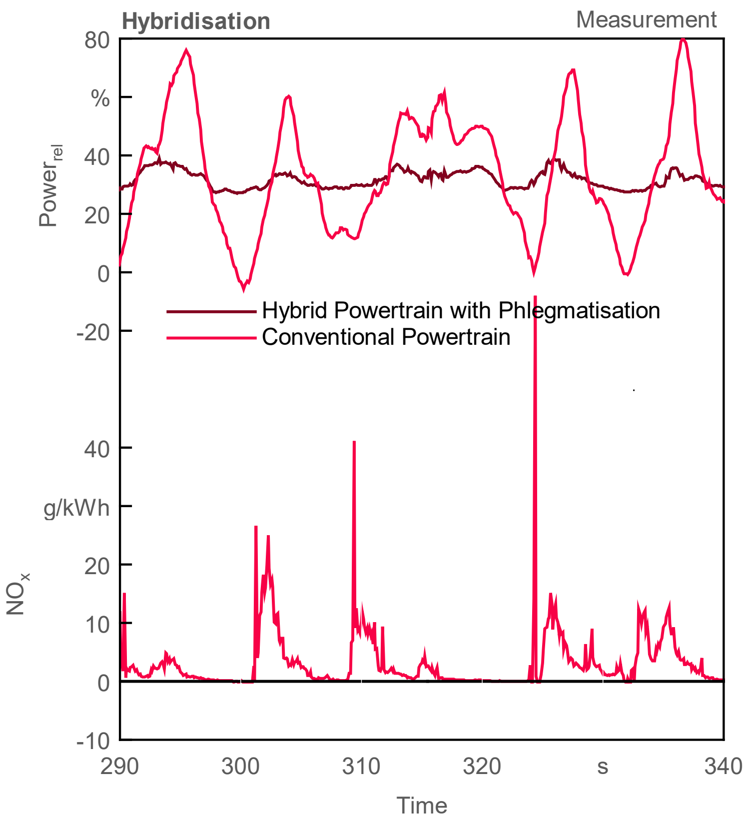

Figure 13 Power and NO_x Formation of a mobile machinery with and without hybridisation; Measurements

All these methods are, in principle, known for a long time from diesel and gasoline engine research. However, the characteristic NOx formation of hydrogen engines let expect a far higher potential of these methods. The investigations of these methods are ongoing or pending at our testbeds.

7 Acknowledgement

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