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Influence of blend ratio on turbocharging & combustion in HS gas eng. applications with CH4/H2 blend

New Engine Developments - Gas

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ABSTRACT

In the near future, hydrogen is expected to play a major role in the storage and distribution of energy converted from electricity generated by renewable sources. Different scenarios exist for large-scale hydrogen usage, including blending withnatural gas within the existing pipeline network or even in anew, dedicated network.

High-speed gas engines play an essential role in peak shaving and grid balancing in electric grids with a high share of renewables. Nowadays, these engines predominantly operate with natural gas and biogas of varying composition. An increased hydrogen content requires special attention to be paid to the combustion concept and essential engine components, such as the turbocharging system. Depending on the blend ratio, necessary measures range from minor adjustments to major modifications of the setup. An important parameter will be the range of blend ratios in which the engine is required to operate.

As part of the EvoLET research project conducted by the Large Engine Competence Center LEC of the Technical University of Graz, in cooperation with partner universities and industrial companies, such as INNIO and Accelleron, many technical questions concerning the blending of hydrogen in natural gas networks were studied. The research work comprised simulation-based theoretical studies as well as engine-based experimental investigations. For the tests,depending on the blend ratio, either a multi-cylinder engine in serial trim or a single-cylinder engine – both based on the Jenbacher Type 6 engine platform – was used. The single-cylinder engine allowed the comparison of port injection and direct injection systems.

The test results were processed to characterize the difference in the boundary conditions for the turbocharging system between pure natural gas, natural gas / hydrogen blends and pure hydrogen. With engine simulation models adapted accordingly, the effects on the turbocharging requirements were investigated and turbocharging concepts derived therefrom.

This paper gives some insight into the most relevant findings and an outlook on the technical measures needed to achieve a suitable setup for natural gas / hydrogen blends and pave the way for the future development and optimization of hydrogen engines.

1 INTRODUCTION

High-speed gas engines operated with natural gas are a vital element in the supply of electricity and heat to domestic and industrial consumers. Especially in power grids with a high share of electricity from alternative energies such as solar and wind high-speed engines serve the purpose to balance the supply and demand.

In the efforts towards a further defossilization of electricity and heat a gradual shift from natural gas to hydrogen produced via electrolysis with electricity from sustainable production plays an important role. The utilization of the hydrogen is feasible in its pure form in a dedicated infrastructure or alternatively by admixture of up to around 25 vol% to the unmodified existing natural gas network. The latter concept may play an important role for a transitional period until sufficient supply of hydrogen is secured and a distribution network is established.

Within the second phase of the COMET research program EvoLET, which was supported with funds by the federal and some provincial governments of Austria, different solutions for hydrogen engines were investigated. This assumed tests at various mixing ratios of hydrogen and natural gas on single- and multi-cylinder engines. The test results and acquired insights were used to deepen studies in the turbocharging of engines operating partly or fully with hydrogen as a fuel.

2 HYDROGEN ENGINES

Large gas engines are well suited to utilize a wide variety of gases, so using hydrogen and hydrogen-natural gas mixtures for power generation are viable options [1, 2, 3].

INNIO Jenbacher classifies three categories of hydrogen engine applications (Figure 1):

A) Engines which are operated with hydrogen that is a compound in the natural gas

B) Engines which are operated with hydrogen that is locally admixed to the gas engines' fuel

C) Pure hydrogen engine applications

It is important to understand the availability of hydrogen. In the natural gas network hydrogen can be blended in for a few hours per year, continuously but fluctuating, or continuously with constant amount. Admixing hydrogen locally to the gas engines' fuel allows to control the amount of hydrogen that is mixed with natural gas, but again the hydrogen may be available for a few hours per year, seasonally or continuously. The pure hydrogen gas engine can run only when hydrogen is available [4].

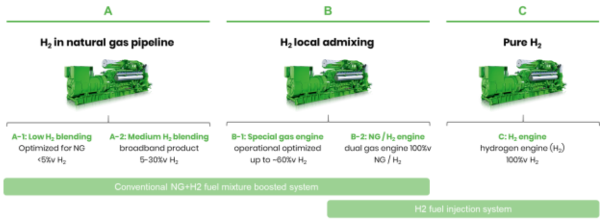


Figure 1. Gas engine solutions depending on H_2 availability [5]

2.1 Impact of hydrogen blending on the natural gas pipeline systems

Hydrogen admixing to natural gas in the transmission and distribution networks is anticipated in parallel to the development of an independent hydrogen infrastructure. Especially when hydrogen is mixed to natural gas in the distribution network system, many gas consumers are affected with fluctuating fuel properties, as the physical composition of the gas is changing.

The aim of adding hydrogen to the natural gas network is to decarbonize natural gas and use the existing gas infrastructure for transport and storage. Studies show that up to 20 or 30 vol% of hydrogen can be added to the existing natural gas transmission and distribution system. 20 resp. 30 vol% of hydrogen in the natural gas results in a decarbonization by only 7 resp. 11 %, because hydrogen has a volumetric heating value of less than 30 % relative to natural gas (Figure 2).

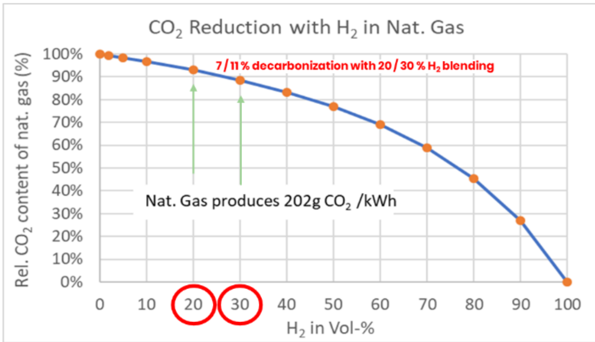
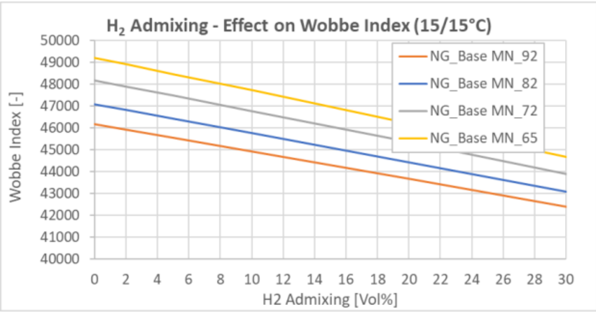


Figure 2. Hydrogen effect on decarbonization of natural gas [5]

The Wobbe index (WI) expresses the heating value of gases used as fuels, considering the proportionality of the calorific value to the specific gravity, or the density ratio between the given fuel gas and air. The methane number (MN) describes the resistance of fuel gases to engine knock. Both fuel property values are important metrics for gas engine power plants.

With 20 vol% hydrogen the Wobbe index (WI) of the gas is going down by 5 to 6 % (Figure 3) and the methane number (MN) is going down by around 13 points (Figure 3). If permanent admixing – a switch from zero to a certain constant percentage of hydrogen – could be done, all end users could adjust their appliances, turbines, gas engines, boilers, etc. accordingly. The real challenge is a very likely discontinuous hydrogen injection to the natural gas grid. Therefore, a stable and constant mixture of natural gas and hydrogen needs to be ensured by gas grid operators to avoid plug flows and inhomogeneous mixtures. A varying hydrogen content in the natural gas results in a “wide Wobbe” gas with wide ranges of heating value and MN.



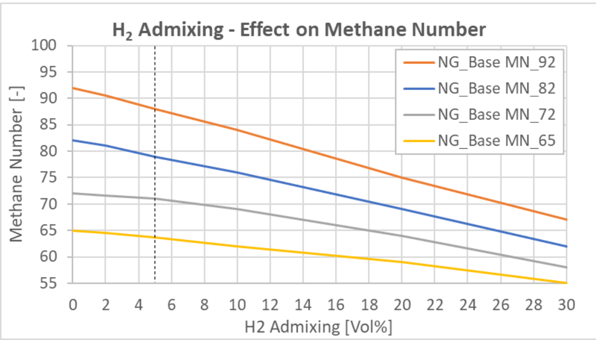


Figure 3. Wobbe Index (WI) and Methane Number (MN) change depending on H_2 content in pipeline gas [5]

2.2 Category A: gas engine solution for hydrogen in natural gas pipeline systems

Smaller amounts of hydrogen in the natural gas with less than 5 vol% change the MN only slightly and that is typically within the tolerance of the gas engine design. Such an engine can operate with a design optimized for the base natural gas composition. A higher amount of hydrogen in the natural gas typically requires a broadband product designed for a lower MN. Such an engine can tolerate both higher and lower MN gas and maintain constant output, but with lower electrical efficiency.

There is a significant difference, whether a reliable signal to the current hydrogen content in the natural gas is available for the gas engine control system or not. If so, the control of the engine can automatically adjust accordingly and the gas engine can run on full output at high hydrogen content, while meeting emission requirements and run reliably. If not, the control system will switch to a safe operation mode when entering the knocking limit with the consequence of a reduced power output.

2.3 Category B: local admixing of hydrogen to natural gas fuel

Admixing hydrogen locally at the gas engine allows controlling the amount of hydrogen mixed with natural gas, and the hydrogen content is always known to the engine control system. That makes this gas engine configuration easier to design and operate. INNIO Jenbacher distinguishes between two applications:

- **B-1**, premixed gas engines for local admixing of up to around 60 vol% hydrogen to pipeline natural gas. This allows using a conventional fuel system with a single pre-turbocharger fuel-air mixing system. The fuel supply system including the turbocharger matching is adjusted according to the maximum amount and the availability of admixing hydrogen. The engine can still run on 100 % natural gas achieving full output but will reduce output at higher hydrogen admixing depending on the base gas and engine version. The design of the engine can be optimized for 100 % natural gas operation or for the operation with high hydrogen admixing.

- **B-2**, a dual gas engine is required, if the engine should be capable of running on 100 % natural gas as well as 100 % hydrogen. In this case two fuel supply systems on the engine are required. A conventional fuel system with a pre-turbocharger fuel-air mixing system for natural gas operation and a separate fuel injection system for hydrogen operation. The engine performance can be optimized for operation on 100 % natural gas or 100 % hydrogen.

2.4 Category C: 100 % hydrogen applications

Operating a distributed power asset on 100 % hydrogen enables a further defossilization of power generation to significantly low CO_2e emissions.

If hydrogen is the fuel of choice and no other carbon neutral or carbon free fuel for backup power is necessary, the gas engine can be designed and optimized for operation at 100 % hydrogen with a single fuel system.

3 LABORATORY ENGINE TESTS WITHIN THE FRAME OF COMET CENTER LEC EVOLET

Two measurement campaigns have been carried out for the experimental evaluation of hydrogen combustion in a large bore engine. The first test targeted applications falling in the INNIO Jenbacher category A (hydrogen blending to the natural gas pipeline system). The second investigation targeted a 100 % hydrogen application (category C).

3.1 Hydrogen admixture on INNIO Jenbacher J612 engine

For the investigation of the impact of hydrogen blending tests were carried out at LEC on the INNIO Jenbacher J612 engine. The objective of this measurement was the assessment of the impact of blending up to 25 vol% hydrogen to the natural gas supply on a serial production engine without modifications to the engine hardware or the engine control. The range of test conditions included a variation of the natural gas methane number, the intake charge temperature, and the target NO_X emissions in natural gas operation for various hydrogen admixing ratios.

The test bed fuel supply was modified to allow the dosing and measurement of hydrogen and propane to the natural gas that was subsequently fed to the fuel dosing unit (TecJet [6]) of the engine and to the pre-chamber gas supply.

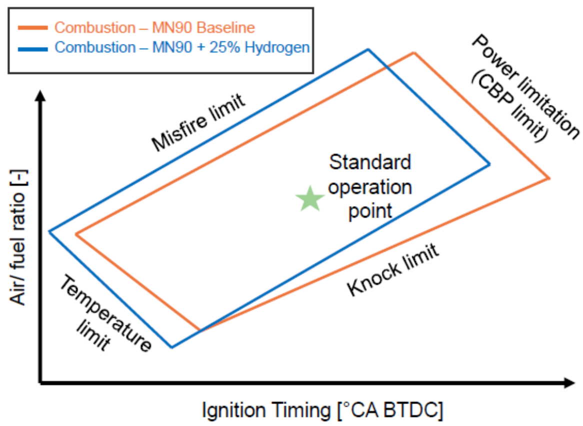


Figure 4. Operating window for natural gas operation and operation with 25 % hydrogen blending at full load

The experiments showed that full load operation was feasible without any adjustments, even with hydrogen blending ratios exceeding 25 vol%. Due to the higher flame speed of hydrogen in comparison to natural gas the combustion was accelerated (cf. Figure 5), resulting in earlier combustion phasing with constant ignition timing and in higher peak cylinder pressures.

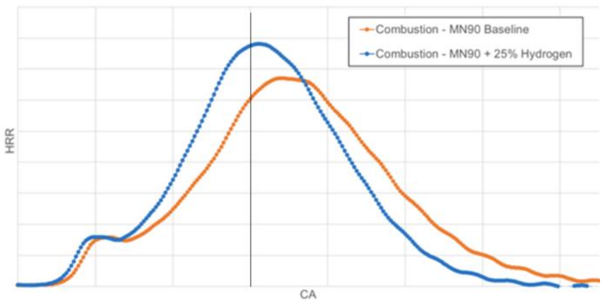


Figure 5. Heat release rate for natural gas operation (orange) and 25 % hydrogen blending to natural gas (blue)

The impact of the higher flame speed and the wider flammability limits of hydrogen resulted in a shift of the engine operating window (Figure 4). The misfire limit was extended to higher excess air ratios and the faster combustion resulted in a lower exhaust gas temperature that allowed an extension of the timing range to include later ignition timing. The lower exhaust gas enthalpy also resulted in a reduced turbine power and a lower compressor recirculation, limiting the feasible ignition timing range (CBP limit). The methane number reduction that results from the hydrogen addition (cf. Figure 3) led to a shift of the knock limit to higher excess air ratios. The shifting of the operating window did not impact the standard operating point that is still well within all the limits of the operating range. An additional benefit of the hydrogen blending that is not reflected in the operating range, is the significant reduction of the emissions of unburned hydrocarbons.

3.2 Pure hydrogen combustion on single cylinder research engine

To study combustion performance for the category C engines experimental investigations were carried out on a single cylinder research engine (SCE) derived from the INNIO Jenbacher Type 6 series.

The engine was configured with a gas scavenged prechamber and a camshaft with early intake valve closing (IVC) before bottom dead center. The compression ratio (CR) was chosen in the range of typical gas engine applications with low flashpoint fuels. Table 1 shows further information about the engine.

The hydrogen was admitted to the engine via port fuel injection. For this, a modified serial production gas dosage valve and an injection nozzle design – investigated with 3D CFD prior to the engine testing

– were used. Figure 6 shows a computer-aided design (CAD) model of the assembly of the nozzle and the intake ports.

Table 1: SCE Type 6 technical data

Rated speed	1500 rpm
Displacement	≈ 6 dm^3
Valve timing	Miller valve timing
Number of intake and exhaust valves	2/2
Swirl/tumble	≈ 0/0
Charge air	Provided by external compressors with up to 10 bar boost pressure
Hydrogen supply	Port fuel injection, up to 10 bar
Ignition system	Modified high-voltage capacitor discharge ignition system

All engine fluids including cooling water, lubricating oil, fuel gas and charge air are controlled to ensure well-defined and reproducible testing conditions. Instead of a turbocharger, an air compressor upstream of the engine and a flap in the engine exhaust system are used to adjust intake and exhaust manifold pressures. A flush mounted piezoelectric cylinder pressure transducer enables real-time calculation of the IMEP of each cycle.

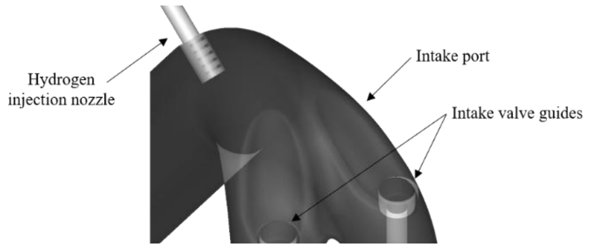


Figure 6. Position of the hydrogen injection nozzle in the intake port (Type 6)

The investigation of 100 % hydrogen fueled engine operation included a study of the impact of ignition timing variations for different excess air ratios and different indicated mean effective pressures (IMEP). While in contrast to the experiments on the J612 described in the previous section, no limits had to be considered for the compressor recirculation, and the exhaust gas temperature limit could be extended since no turbine inlet temperature limitation had to be taken into account.

One objective of the investigation was the assessment of the knocking and misfire limits and their dependencies on excess air ratios and indicated mean effective pressures. Selected results for the ignition timing operating range are depicted in Figure 7 for indicated mean effective pressures of 11.5 bar, 16.5 bar and 21.5 bar.

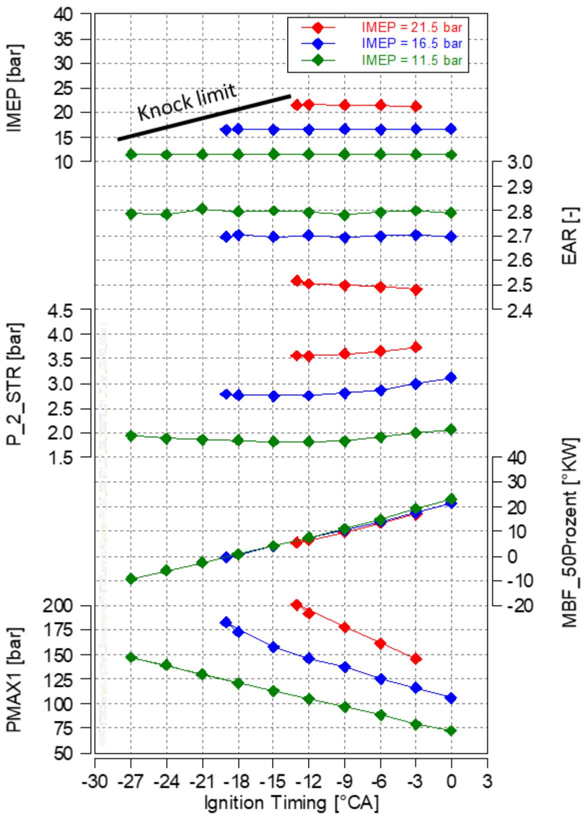


Figure 7. SCE performance parameters for hydrogen operation at various IMEP

For the two higher load investigations the feasible combustion phasing range was limited by knocking combustion for very early combustion phasing. This limit is extended to later timings with higher load. The operation at the knock limit was also the operation with the highest peak cylinder pressures with approximately 200 bar. The excess air ratio for all three timing variations was maintained in a range between 2.5 and 2.8 that was assumed to be achievable with single stage turbocharging. For each timing variation the excess air ratio was maintained at a fixed value such that the boost pressure demand (P_2_STR) was a result of this investigation.

Figure 8 illustrates typical heat release rates for pure hydrogen combustion for three different mean effective pressures with similar combustion phasing. In contrast to the natural gas operation and the operation with hydrogen blending (cf. Figure 5) the distinct initial peak in the heat release rate caused by the prechamber combustion is hardly visible with pure hydrogen combustion. Additionally, the rise of the peak heat release rate when the prechamber flame jets ignite the mixture in the main combustion chamber is steeper than for natural gas or natural gas/hydrogen blends.

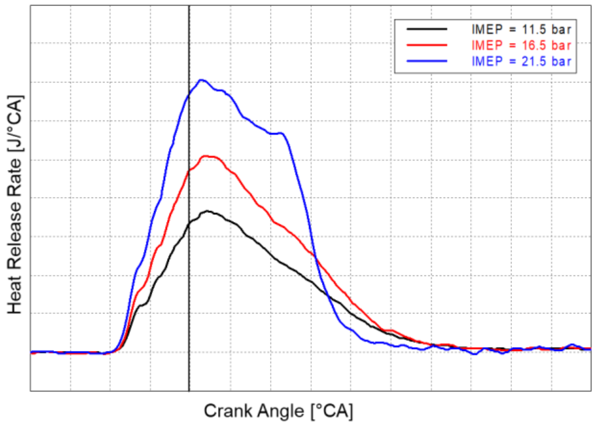


Figure 8. Heat release rates for pure hydrogen operation for different indicated mean effective pressures

4 HYDROGEN ADMIXTURE OF UP TO 25 % TO NATURAL GAS ENGINES

The test campaign at LEC with a variable share of hydrogen admixture to natural gas on a multi-cylinder engine brought valuable insights into its effect on engine performance and NO_X emissions. A variation of the excess air ratio and ignition timing demonstrated feasible configurations under preservation of emissions, combustion stability and control margin. A theoretical investigation of the potential benefits of an adaptation of the turbocharger specification was conducted for the same engine type, as such modifications were not in the scope of the test program.

4.1 Influence of hydrogen admixture on fuel properties and turbocharging

The change of the fuel composition draws with it an impact on the turbocharging requirements [7]. While the chemical fuel properties have a direct influence on the resulting mass flow rates for air and fuel, the ignition and combustion properties define the operating conditions for the turbocharger via the combustion concept.

Even though the lower heating values LHV and the stoichiometric air to fuel ratios L_{min} are very different for hydrogen and natural gas, the stoichiometric masses of air per energy content L_{min}/LHV are relatively close for both fuels. For 25 vol% of hydrogen in natural gas the difference to pure natural gas becomes very small, as the mass fraction of hydrogen in the fuel blend amounts to only around 4 % due to the much lower density of hydrogen.

For a more accurate estimate of the impact of hydrogen admixture to natural gas on the mass of mixture to be delivered to the engine by the turbocharger the excess air ratio needs to be included, see in eq. (1). As the concerned engine is equipped with a central gas mixer before the compressor of the turbocharger the fuel needs to be taken into account as well in this calculation. Table 2 specifies the air/fuel mixture mass flow relative to its combustion heat content for natural gas, hydrogen and a blend with 25 vol% of hydrogen, which is calculated as follows:

$$\frac{\dot{m}_{mixture}}{H_{fuel}} = \frac{\dot{m}_{mixture}}{\dot{m}_{fuel} \cdot LHV} = \frac{1 + L_{min} \cdot \lambda_{comb}}{LHV} \qquad (1)$$

The hydrogen content in the fuel blend influences the combustion duration, such that the excess air ratio λ_{comb} must be adapted to keep the NO_X emissions at the same level as for pure natural gas and to obtain a reliable combustion without knocking.

Table 2. Fuel properties

Fuel type	LHV [MJ/kg]	L_{min} [kg/kg]	$\frac{L_{min}}{LHV}$ [kg/MJ]	λ_{comb} [-]	$\frac{\dot{m}_{mixture}}{H_{fuel}}$ [kg/MJ]
Natural gas	49.09	16.9	0.344	1.98	0.70
Hydrogen	119.96	34.3	0.286		
25 % H_2 blend	51.84	17.57	0.339	2.09	0.73

All-in-all, the admixture of hydrogen up to 25 vol% has only a small impact on the mixture requirement. Therefore, the influence on the turbocharging system can be considered as quite moderate.

4.2 Simulation model set-up

The engine model for the simulation studies for the natural gas/hydrogen blends was calibrated to tests within the frame of the EvoLET research projects conducted on the INNIO Jenbacher J612 at the LEC.

Table 3. Engine main parameters

Parameter	Unit	Value
Bore	[mm]	190
Stroke	[mm]	220
$BMEP_{nom}$	[bar]	22.2
$Speed_{nom}$	[1/min]	1500

In accordance with the findings from various tests within the EvoLET project, the combustion duration was adjusted for the variation of the hydrogen blending rate and the excess air ratio, while the shape and the duration of combustion were kept constant over the entire engine load range (cf. Figure 9).

The test engine is equipped with a single-stage turbocharging system, which is controlled by a compressor bypass. The bypass rate is set to achieve a defined turbocharger control margin. As this engine corresponds to a serial engine for pure natural gas operation, the compressor and turbine specifications were matched accordingly. In the engine simulation model, the turbocharger components are represented with measured maps.

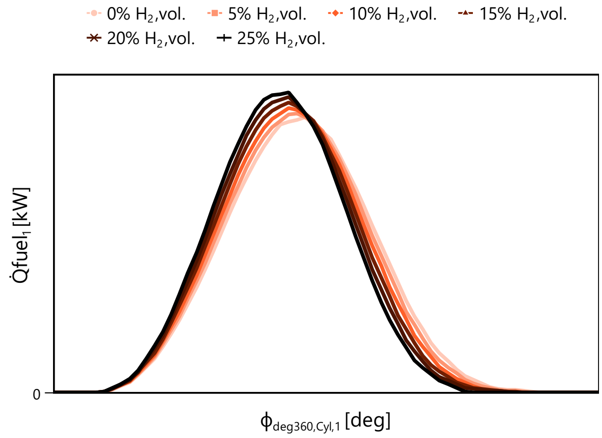


Figure 9. Heat release rates for various hydrogen admixture rates to natural gas

For the engine and turbocharging system simulations the in-house developed 0D/1D tool ACTUS [8] of Accelleron was applied. The combustion heat release rate for the natural gas and hydrogen blends was modelled with the empiric function of Wiebe [9].

4.3 Concept with constant NO_X, ignition timing and turbocharger matching

The engine simulations for the hydrogen admixture were conducted in steps of 5 vol% hydrogen from pure natural gas up to 25 vol%. Corresponding to the engine test results, a constant ignition timing, but adapted shape of combustion heat release and excess air ratio λ_{comb} were applied.

The operation point in the compressor map moves to a pressure ratio higher by about 3 % and a reference volume flow lower by 2 % (cf. Figure 10). This shifts the operation line into an area at lower isentropic efficiency closer to the surge line in the compressor map. The exhaust temperature is reduced by around 25 K due to the higher excess air ratio. Together, the higher compressor pressure ratio and the lower exhaust temperature cause a reduction of the compressor recirculation ratio to only 2 % of the engine mass flow, which is not sufficient for the engine operation at high ambient temperatures and high altitude and in transient operation. Engine efficiency is increased by about 0.4 % due to the shorter combustion duration.

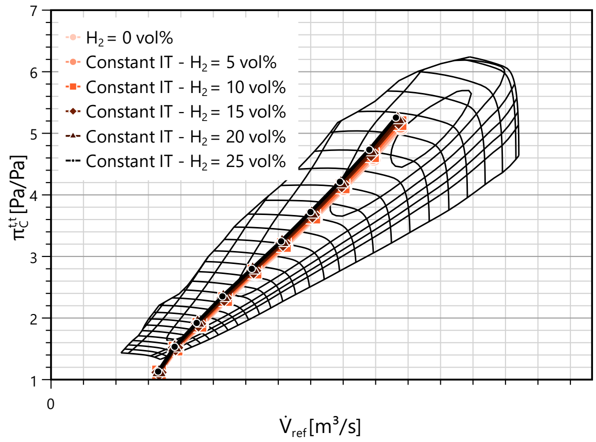


Figure 10. Positions of engine operating lines on the compressor map for various hydrogen admixture rates to natural gas

To adjust the control margin with increased hydrogen content back to the level of pure natural gas operation, two concepts are feasible:

- The first method builds on a delayed start of combustion to increase the exhaust enthalpy to the required extent, but without an adaptation of the turbocharger specification.
- The second method focuses on a compensation by a smaller effective area of the turbine, but no modification on engine tuning parameters.

The effective turbine area is defined as the area of an equivalent nozzle with isentropic flow (Ψ denotes the nozzle flow function) and can be related to the widely applied definition of the reduced mass flow:

$$S_{effT} = \dot{m}_T \cdot \frac{\sqrt{R_{exh} \cdot T_{T,in}}}{p_{T,in} \cdot \Psi(\Pi_T)} = \dot{m}_{red,T} \cdot \frac{\Psi(\Pi_T)}{\sqrt{R_{exh}}} \qquad (2)$$

4.4 Concept with constant NO_X and turbocharger matching and delayed ignition timing

A later start of combustion leads to a higher control margin for the turbocharging system. The center of combustion is delayed, hence the turbine inlet temperature and exergy increased. At the same time, maximum cylinder pressure and temperature are reduced. At equal NO_X emissions, this allows for a lower excess air ratio λ_{comb}. Thereby, the compressor pressure ratio can be reduced, which additionally improves the control margin. In pure natural gas operation, the performance remains unaffected, unlike for the method with a modified turbocharger specification.

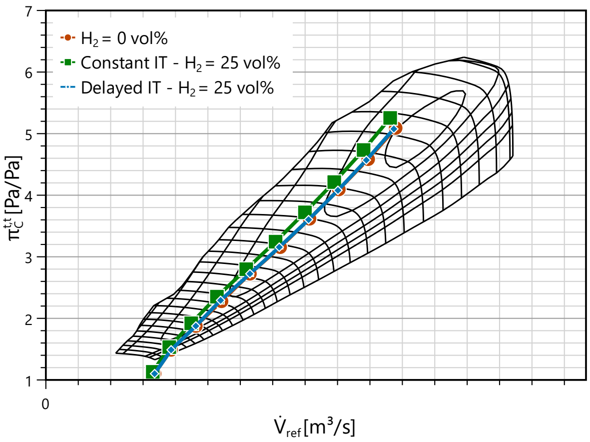


Figure 11. Influence of a delayed ignition timing on the position of the engine operating line on the compressor map

To obtain the same turbocharger control margin for 25 vol% hydrogen as for the pure natural gas engine the start of combustion in the simulation needs to be delayed by approximately 4°CA. The operating line in the compressor map remains very similar to the operation with pure natural gas. At full load the compressor reference volume flow is reduced by only 1 % at a preserved compressor pressure ratio (cf. Figure 11). The exhaust temperature is increased by slightly less than 10K. The turbocharger efficiency is kept constant. Engine efficiency is only slightly reduced by about 0.1 % from pure natural gas to 25 vol% of hydrogen due to the shorter combustion with hydrogen admixture.

4.5 Concept with constant NO_X and ignition timing and modified turbocharger matching

The most straightforward method to increase the turbocharging control margin is to use a turbine specification with a smaller effective area. This leads to a higher turbine pressure ratio and consequently to a higher turbine power output. The resulting excess power on compressor side converts into an additional control margin in the compressor recirculation. As a drawback, the exhaust backpressure is increased and at pure natural gas operation engine efficiency is reduced, as the supplementary power recovered from the turbine is dissipated via the compressor recirculation.

In the simulated case with 25 vol% hydrogen the turbine specification needed to be rematched with an about 11 % smaller effective area to achieve the targeted control margin.

The compressor reference volume flow is increased by 7 % and the compressor pressure ratio by 8 % (cf. Figure 12). Due to this, the margin to the maximum speed of the turbocharger is reduced.

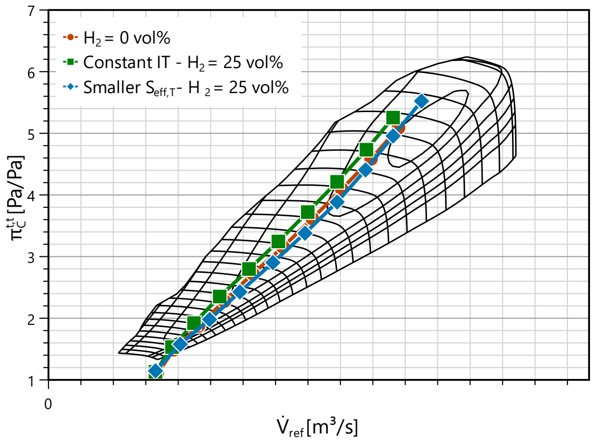


Figure 12. Influence of a modified turbocharger matching on the position of the engine operating line on the compressor map

With the applied turbine specification, the compressor recirculation ratio results 1 % higher than with the original setup. The exhaust temperature is reduced by close to 15K. With the reduced effective turbine area, the exhaust back pressure of the engine exceeds the inlet pressure. In practical operation, such conditions may reduce the combustion stability. Despite of a similar level of turbocharger efficiency, the engine efficiency is reduced by 0.6 %, as the formerly positive contribution of the gas exchange loop to the engine power is turned into a loss.

4.6 Conclusions

An increased hydrogen content of up to 25 vol% leads to a reduced control margin due to the shorter combustion duration. With a delay of the ignition timing the control margin can be re-established via a higher exhaust enthalpy and additionally by converting the gained NO_X margin into a lower excess air ratio. Not only the same turbocharger as for the pure natural gas application can be applied, but also a similar level of engine efficiency can be achieved and the margin to the speed limit of the turbocharger is preserved. Alternatively, with a reduced effective turbine area, the control margin can be compensated as well, but at reduced engine efficiency and margin to the turbocharger speed limit.

For the application of the adapted ignition timing the simulation results show only a minor shift of the operating points on the compressor map for the range of considered fuel blends. Thus, not even a modification of the compressor specification is required to operate the engine at preserved control margin with the full fuel flexibility from 0 to 25 vol% of hydrogen.

Table 4. Comparison of control methods

Method	Effect on TC	Advantages	Drawbacks
IT const.	$\dot{V}_{ref} \downarrow$ $T_{T,in} \downarrow$ $\Pi_C \uparrow$	$\eta_{engine} \uparrow$	ctrl. margin $\downarrow$ surge margin $\downarrow$ $p_{cyl,max} \uparrow$
IT adapted	$\dot{V}_{ref} \rightarrow$ $\Pi_C \rightarrow$ $T_{T,in} \uparrow$	$\eta_{engine} \rightarrow$ ctrl. margin $\rightarrow$ $p_{cyl,max} \downarrow$	none
TC adapted (IT const.)	$\dot{V}_{ref} \uparrow$ $\Pi_C \uparrow$ $T_{T,in} \downarrow$	ctrl. margin $\rightarrow$	$\eta_{engine} \downarrow$ n_{TC} margin $\downarrow$ $p_{cyl,max} \uparrow$

The current Accelleron turbocharger portfolio for natural gas engines is well suited for high-speed gas engines running on natural gas / hydrogen blends up to at least 25 vol% hydrogen.

5 PURE HYDROGEN ENGINES

While hydrogen admixture is an interesting concept for the use of hydrogen in existing natural gas distribution networks, infrastructures operating with hydrogen only are under planning. For such applications dedicated hydrogen engines are required.

5.1 Fuel properties and combustion concepts

Compared to natural gas the properties of hydrogen are much more difficult to handle in internal engine combustion. Due to the very small ignition energy of hydrogen the risk of premature ignition of the fresh charge in the cylinder is much higher than for natural gas. The laminar flame speed greater by an order of magnitude over natural gas leads to a much faster conversion of the fuel energy.

Table 5. Combustion properties

Fuel type	Minimum ignition energy [mJ]	Autoignition temperature [°C]	Laminar flame speed [cm/s]
Methane	0.28	535	≈ 37
Hydrogen	0.016	500	≈ 290

The conditions in the combustion chamber therefore need to be adapted such that the ignition can be kept under control and the rate of release of the combustion heat can be reduced to a reasonable level.

As outlined by different sources [10, 11, 12] following concepts are feasible to keep hydrogen combustion under control:

- Highly lean charge in Otto cycle with port fuel injection (PFI) or low-pressure direct injection (LPDI)
- Lean charge in Otto cycle in combination with exhaust gas recirculation (EGR)
- Combustion concepts based on high-pressure direct injection (HPDI), e.g. partially homogeneous charge with spark ignition or diffusion combustion with compression ignition

With the application of exhaust gas recirculation or high-pressure direct injection in a Diesel cycle an operation of the engine at lower excess air ratios than for a spark-ignited homogeneous charge is feasible. However, both are higher in costs due to the additional components for the EGR system or the high-pressure fuel supply system and the higher technological effort for HPDI compared to the gas admission valves of a PFI system. Combustion concepts with highly lean charge at excess air ratios of up to around 3.5 are therefore the most affordable technology, which could only be offset in costs by the alternative concepts, if a significantly higher power density could be achieved.

5.2 Turbocharging of hydrogen engines with highly lean combustion

Due to the high excess air ratio, for a given engine power rating, the required charge air pressure becomes much higher than for state-of-the-art engines operated with natural gas. As charge air pressure is limited by the compressor map of the turbocharger the maximum achievable engine power density must be reduced with very high excess air ratios.

For the turbocharger a high excess air ratio does not only mean that the pressure ratio of the compressor needs to be maximized to yield a high power density at the engine. As the fuel consumption does not greatly alter with an increased excess air ratio, a comparable enthalpy of combustion is distributed over an enlarged amount of working gas. Hence, the turbine inlet temperature is reduced, while the inlet pressure is increased. The density at turbine inlet becomes higher, manifesting in a smaller effective turbine area. As described in detail in [7], the ratio of the specific volume flows through the compressor and the turbine is dependent on the density at turbine inlet. If the components of a given turbocharger generation are not specifically designed for such applications, the portfolio of available specifications may not necessarily fit to the resulting turbocharger requirements.

At Accelleron a study was conducted to evaluate the matching capabilities and limits on engine side for the A200-H turbocharger type [13] – the product with currently the highest available single-stage pressure ratio in Accelleron's portfolio – on the INNIO Jenbacher J612 engine in lean-burn hydrogen operation.

The turbocharging system was simulated with constant isentropic efficiencies of 80 % for the compressor and 83 % for the turbine. This allowed to avoid distortion of the simulation results by matching part specific efficiency charts. The selected values for the component efficiencies orient on typically achievable levels at the layout point of the chosen turbocharger type. They may not be reached at the highest feasible pressure ratios. In the turbocharger matching a constant control reserve via compressor recirculation at the level of natural gas engines was included.

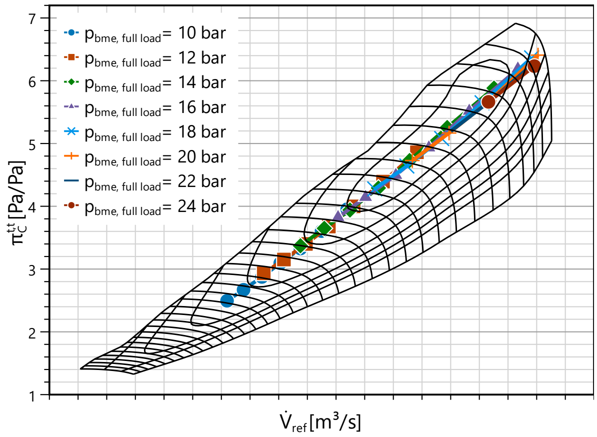


Figure 13. Operating points for various mean effective pressures and excess air ratios compared to typical compressor map of A200-H [13]

As boundary conditions for the simulation an altitude of 500 m.a.s.l. and a temperature of 298 K were set, corresponding to the altitude of the R&D site of INNIO and the ISO standard temperature. For different ambient conditions the feasible range of excess air ratios must therefore be adapted.

Brake mean effective pressure p_{bme} was varied in a range from 10 to 24 bar in steps of 2 bar. Excess air ratio λ_{comb} was set to levels starting from 2.4 in notches of 0.2 up to the highest feasible value below the technical limit of a typical compressor map of A200-H and not higher than 3.6 (cf. Figures 13 and 14).

As for the investigation of hydrogen admixture the shape of the combustion heat release was adapted to the excess air ratio.

5.3 Results with port fuel injection

A first series of layout points was calculated with a port fuel injection system with gas admission valves for the hydrogen admission. The advantage of gas admission valves over injectors is their simplicity and low costs. If hydrogen is injected into the intake ports, it fills a high share of the space due to its more than 14 times lower density than of air; at an excess air ratio λ_{comb} of 3.0 the proportion of the fuel to the mixture volume is at around 12 %.

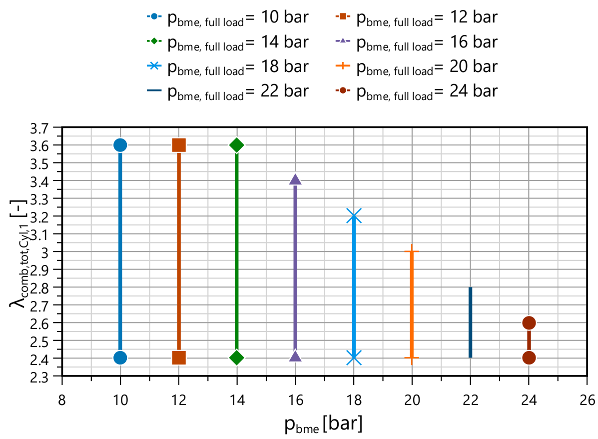


Figure 14. Feasible excess air ratios at various mean effective pressures

Under the specified conditions the maximum excess air ratio λ_{comb} of 3.6 can only be reached at a p_{bme} of 14 bar. Beyond that level the maximum excess ratio needs to be gradually reduced. At a p_{bme} of 24 bar an excess air ratio of more than 2.6 is not feasible.

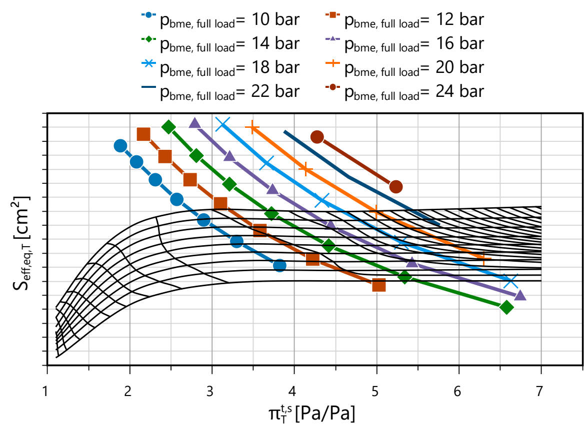


Figure 15. Operating points for various mean effective pressures and excess air ratios compared to nozzle rings of smallest A200-H [13] turbine trim

On turbine side the resulting effective areas were compared with the maps of the stator specifications for the smallest rotor trim of the chosen turbocharger frame size. The chart in Figure 15 shows that for the range of p_{bme} from 12 to 18 bar the smallest stator nozzle rings are not sufficient to achieve the target compressor pressure ratio. In addition to this, the turbine pressure ratios required for the highest excess air ratios at p_{bme} levels between 14 and 20 bar lead to engine back pressures in the region of the charge air pressure. This may impose issues to the combustion stability via a negative effect on the gas exchange process.

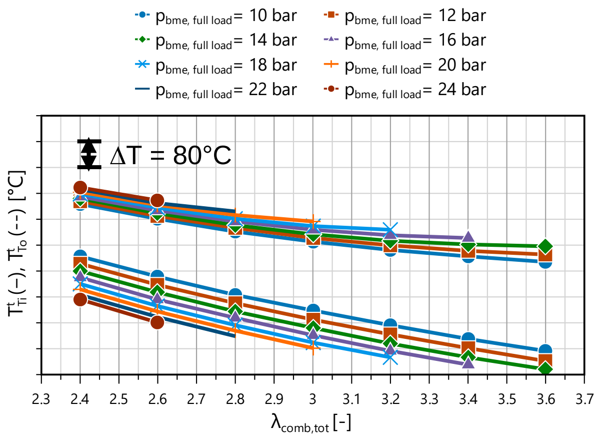


Figure 16. Temperatures at turbine inlet and outlet for various mean effective pressures and excess air ratios

The temperature at turbine inlet is heavily influenced by the excess air ratio and the brake mean effective pressure (cf. Figure 16). Between the lowest and the highest excess air ratio the temperature difference is about 150 to 200°C, being a strong contributor to the wide range of effective areas.

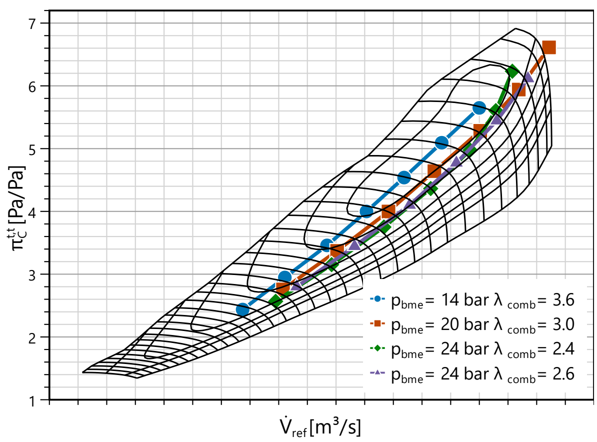


Figure 17. Engine operating lines on compressor map for selected cases

For a selected number of the shown cases the simulations were repeated with the actual performance maps for the components. Instead of only the full load points the operating lines were calculated down to 40 % of the rated engine power. The excess air ratio is kept constant along the operating line by adjustment of the compressor recirculation rate. As the turbocharger efficiency is not constant along the operating line and is not equal for the turbocharger specifications chosen for the different cases, the operating lines do not behave identically (cf. Figure 17).

For the case with 24 bar at an excess air ratio λ_{comb} of 2.6 the full load point ended up outside of the compressor map. Therefore, only the 90 % load point is displayed. At 20 bar with an excess air ratio λ_{comb} of 3.0 the full load point is located slightly above the highest feasible turbocharger speed as well. These cases demonstrate well that for applications at the limit of a specific turbocharger generation sufficient matching reserves need to be included such that for different cylinder numbers and engine tuning specifications of a certain engine type the same rated power can be achieved.

5.4 Results with low-pressure direct injection

Low-pressure direct injection (sometimes also called medium-pressure direct injection) avoids the constraint with port fuel injection of the space requirement of hydrogen as a fuel in the intake port at the expense of higher component costs. Due to this, a slightly lower charge air pressure is required to deliver the same amount of air/fuel mixture to the engine or, alternatively, the gained margin can be converted into a higher power output.

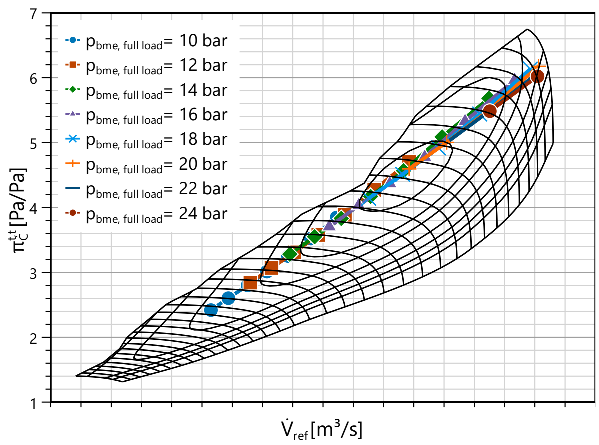


Figure 18. Operating points for various mean effective pressures and excess air ratios compared to typical compressor map of A200-H [13]

A repetition of the variation of brake mean effective pressure p_{bme} and excess air ratio λ_{comb} with low-pressure direct injection instead of port fuel injection demonstrates a slightly lower compressor pressure requirement (cf. Figure 18). At the layout points with the highest pressure ratios the difference is around 0.2. This is a noticeable step, but by far not a massive leap towards eased turbocharging requirements. From the motivation of improving the conditions for turbocharging the hydrogen engine the application of low-pressure direct injection is not worthwhile.

5.5 Conclusions

Hydrogen engines require combustion concepts adapted to the high reactivity and flame speed of hydrogen. A simple solution based on cost-efficient components can be established by the combination of a high excess air ratio with port fuel injection.

Studies to single-stage turbocharged hydrogen engines confirm that even with the latest generation of turbochargers optimized for highest pressure ratios combustion concepts based on highly lean air/fuel mixtures are limited to brake mean effective pressure ratings much lower than for natural gas engines. Not only the application limits of the compressor map put constraints to the achievable output, but also the high exhaust back pressure at high compressor pressure ratios. Moreover, high excess air ratios necessitate small turbine areas due to the high exhaust density.

Towards higher power densities in Otto cycle with hydrogen as an engine fuel different options are feasible. The application of a two-stage instead of a single-stage turbocharging system would allow for higher pressure ratios on compressor side, while through the higher turbocharging efficiency the exhaust back pressure could still be preserved. Another alternative would be the application of exhaust gas recirculation to reduce the reactivity of the cylinder charge and the flame speed.

6 CONCLUSIONS AND OUTLOOK

Hydrogen engines are expected to play an important role in the defossilization of power and heat generation. In the transition phase from natural gas to hydrogen as engine fuel admixing of hydrogen to natural gas will be a possible solution as well as the introduction of dedicated hydrogen networks. In case of admixture the hydrogen can be added in any location within the pipeline network or at the engine. Moreover, the volume fraction of hydrogen may fluctuate. Based on these operating concepts, INNIO Jenbacher distinguishes three different categories of technical solutions to serve the market with optimum products. Within the COMET research program EvoLET various solutions for hydrogen admixture and pure hydrogen operation were investigated. For the admixture of up to 25 vol% of hydrogen to natural gas engine tuning based solutions could be demonstrated, which allow for a stable engine operation while still fulfilling NO_X emission limits and keeping the required control reserves. Tests with pure hydrogen concepts on a single-cylinder engine allowed for an exploration of the technical limits of hydrogen combustion. As the turbocharging system was simulated by an externally powered supercharger and an adjustable orifice at the engine outlet, the tests were not constrained by the characteristics of a turbocharger. Building on the gained insights from the different test series Accelleron investigated turbocharging solutions for hydrogen admixture as well as pure hydrogen engines by simulation. An alternative concept based on a modified turbocharger specification for hydrogen admixture could not match the performance of the adjustment of the ignition timing. This confirms on the other hand that with a turbocharger matching dedicated to pure natural gas operation an admixture of hydrogen up to 25 vol% is feasible without a compromise in engine operation and performance. For the pure hydrogen operation, the limits of single-stage turbocharging with highly lean cylinder charge were investigated and assessed for the A200-H turbocharger series. The achievable compressor pressure ratio and turbocharger efficiency set clear limits to the potentially feasible excess air ratio and power density of the engine. The applications of port fuel injection and direct injection do not show any significant difference in these findings. Concepts with application of exhaust gas recirculation may offer an extended margin towards a lower excess air ratio, such that higher power density may be feasible.

7 DEFINITIONS, ACRONYMS, ABBREVIATIONS

°CA: Degree crank angle

BMEP: Brake mean effective pressure

BTDC: Before top dead center

CAD: Computer-aided design

CBP: Compressor bypass

CFD: Computational fluid dynamics

CO_2e: Carbon dioxide equivalent

CR: Compression ratio

EAR: Excess air ratio

EGR: Exhaust gas recirculation

H_2: Hydrogen

HPDI: High-pressure direct injection

HRR: Heat release rate

IMEP: Indicated mean effective pressure

IT: Ignition timing

IVC: Intake valve closing

LHV: Lower heating value

L_{min}: Stoichiometric amount of air

LPDI: Low-pressure direct injection

m.a.s.l.: Meter above sea level

MBF: Mass burnt fraction

MN: Methane number

$\dot{m}_{red,T}$: Reduced turbine mass flow

$\dot{m}_T$: Turbine mass flow

NO_X: Nitrogen oxides

p_{bme}: Brake mean effective pressure

$p_{cyl,max}$: Maximum cylinder pressure

PFI: Port fuel injection

PMAX: Maximum cylinder pressure

$p_{T,in}$: Turbine inlet pressure

R&D: Research and development

R_{exh}: Specific gas constant for exhaust gas

SCE: Single-cylinder engine

$S_{eff,T}$: Effective turbine area

TC: Turbocharger

$T_{T,in}$: Turbine inlet temperature

vol%: Volume percent

V_{ref}: Compressor reference volume flow

WI: Wobbe index

η_{engine}: Engine efficiency

η_{TC}: Turbocharger efficiency

λ_{comb}: Combustion excess air ratio

Π_C: Compressor pressure ratio

Π_T: Turbine pressure ratio

φ: Crank angle

Ψ: Flow function for isentropic nozzle

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