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Proceedings

Ein stetig steigender Fundus an Informationen ist heute notwendig, um die immer komplexer werdende Technik heutiger Kraftfahrzeuge zu verstehen. Funktionen, Arbeitsweise, Komponenten und Systeme entwickeln sich rasant. In immer schnelleren Zyklen verbreitet sich aktuelles Wissen gerade aus Konferenzen, Tagungen und Symposien in die Fachwelt. Den raschen Zugriff auf diese Informationen bietet diese Reihe Proceedings, die sich zur Aufgabe gestellt hat, das zum Verständnis topaktueller Technik rund um das Automobil erforderliche spezielle Wissen in der Systematik aus Konferenzen und Tagungen zusammen zu stellen und als Buch in Springer.com wie auch elektronisch in Springer Link und Springer Professional bereit zu stellen. Die Reihe wendet sich an Fahrzeug- und Motoreningenieure sowie Studierende, die aktuelles Fachwissen im Zusammenhang mit Fragestellungen ihres Arbeitsfeldes suchen. Professoren und Dozenten an Universitäten und Hochschulen mit Schwerpunkt Kraftfahrzeug- und Motorentechnik finden hier die Zusammenstellung von Veranstaltungen, die sie selber nicht besuchen konnten. Gutachtern, Forschern und Entwicklungsingenieuren in der Automobil- und Zulieferindustrie sowie Dienstleistern können die Proceedings wertvolle Antworten auf topaktuelle Fragen geben.

Today, a steadily growing store of information is called for in order to understand the increasingly complex technologies used in modern automobiles. Functions, modes of operation, components and systems are rapidly evolving, while at the same time the latest expertise is disseminated directly from conferences, congresses and symposia to the professional world in ever-faster cycles. This series of proceedings offers rapid access to this information, gathering the specific knowledge needed to keep up with cutting-edge advances in automotive technologies, employing the same systematic approach used at conferences and congresses and presenting it in print (available at Springer.com) and electronic (at Springer Link and Springer Professional) formats. The series addresses the needs of automotive engineers, motor design engineers and students looking for the latest expertise in connection with key questions in their field, while professors and instructors working in the areas of automotive and motor design engineering will also find summaries of industry events they weren't able to attend. The proceedings also offer valuable answers to the topical questions that concern assessors, researchers and developmental engineers in the automotive and supplier industry, as well as service providers.

Alexander Heintzel
(Hrsg.)

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Geleitwort

HERZLICH WILLKOMMEN LIEBE KONGRESSTEILNEHMER

Die politische Vorgabe in Europa zielt auf ein Verbot von verbrennungsmotorisch angetriebenen Neufahrzeugen ab 2035, um in Stufen über die Agenda „Fit for 55“ im Verkehrsbereich zu Nullemissionen zu kommen. Offen bleibt in der aktuellen Diskussion der beschlossene und notwendige Austausch zwischen den politischen, wissenschaftlichen und sozialen Stakeholdern zur Integration von re-fuels und damit einer sektorgekoppelten Bewertung in der finalen Gesetzgebung.

Isoliert sich Europa mit dieser Vorgehensweise von den technologie-neutralen Lösungen der führenden Volkswirtschaften im Wettbewerb von Wissenschaft, Industrie und Wertschöpfungsketten? Effektiver Klimaschutz erfordert neben Geschwindigkeit und Innovations- Wettbewerb von Technologien auch gleichrangige ökonomische Betrachtungen.

Im 10. Internationalen Motorenkongress führen wir den aktuellen, umfassenden wissenschaftlichen und technologischen Wissensstand im Gesamtsystem nichtfossiler Kraftstoffe und Potenziale des Verbrennungsmotors zusammen.

Sichern Sie sich Ihren Wissensvorsprung und profitieren Sie!

- Es erwarten Sie internationale Referenten, hochkarätige Vorträge und Diskussionsrunden.
- Nutzen Sie den Kongress zum Netzwerken – der Abend der Motoren-Community bietet interessante Gespräche in ungezwungener Atmosphäre.
- Eine begleitende Fachausstellung informiert über innovative Produkte und Dienstleistungen.

Wir freuen uns auf Ihre Teilnahme!

Im Namen der Programmbeiräte

Prof. Dr. Peter Gutzmer
Wissenschaftlicher Leiter des Kongresses,
Herausgeber ATZ | MTZ-Gruppe

Foreword

A WARM WELCOME TO ALL PARTICIPANTS

The political target in Europe is to ban vehicles powered by internal combustion engines from 2035 in order to achieve zero emissions in the transportation sector stage by stage via the “Fit for 55” package of measures. What remains unresolved in the ongoing debate is the necessary discussion between the political, scientific, and social stakeholders on the integration of electrofuels and therefore on a sector-linked assessment in the final legislation.

But with this approach, is Europe now isolating itself from the technology-neutral solutions of the leading economies in the competition between science, industry, and value chains? Effective climate change not only needs speed and competition between innovative technologies, but also requires economic considerations that are equally important.

At the 10th International Engine Congress, we bring together the current state of comprehensive scientific and technological know-ledge in the overall system of non-fossil fuels and the potential of the combustion engine.

Stay abreast of current trends and benefit from a lead in knowledge!

- You can expect international speakers as well as top-level presentations and panel discussions.
- The congress is a great opportunity to “network“ – the evening event for the engine community offers stimulating discussions in an informal atmosphere.
- The trade exhibition, held in parallel, provides ample information about innovative products and services in the field of combustion engine development.

We look forward to your participation.

On behalf of the program advisory boards

Prof. Dr. Peter Gutzmer
Scientific Director of the Congress,
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H_2 Engine Hybrid Powertrain for Future Light Commercial Vehicles

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Abstract. Within the joint project "H_2 ICE Democar", funded by the Federal Ministry for Economic Affairs and Climate Action, Ford, Bosch, MAHLE, Umicore, Institute of Automotive Engineering (IFS, University of Stuttgart), Research Institute for Automotive Engineering and Powertrain Systems (FKFS), Chair of Thermodynamics of Mobile Energy Conversion Systems (tme, RWTH Aachen University), DHL and Shell are identifying key requirements for the powertrain of a light commercial vehicle with H_2 engine and hybrid powertrain. The potential of the technical solution will be demonstrated for a logistics application by a demo car. Besides engine and vehicle testing, simulations and experiments are carried out to gain a deeper understanding about the specifics of H_2 engine operation in a hybrid powertrain.
This paper focuses on selected aspects of the project such as modification of the well-known 1.0 L Ford EcoBoost three-cylinder gasoline engine for H_2 operation. The analysis and optimization of the mixture preparation, verified by high-speed Schlieren measurements, 3D-simulation and thermodynamic analysis performed on multicylinder engines is discussed in detail as well as the selection of a suitable turbo charging system to enable lean operation ($\lambda > 2$) in the complete engine map.

Keywords: Hydrogen, Hydrogen Engine, 3D-CFD, Boosting System, Democar.

1 Introduction

Hydrogen (H_2) is on its way to become an important energy carrier of the future. It will contribute to well-to-wheel and life cycle CO_2 neutral on-road mobility solutions including EU targets for CO_2 neutrality. The European Union as well as important Asian markets (Japan, Korea, China) plan to establish a H_2 based energy infrastructure as part of a sustainable future energy mix. Complementary to the already initiated electrification of propulsion systems on large scale the prospect of an emerging hydrogen infrastructure suggests the usage of H_2 in sustainable powertrain systems. H_2 can be employed in fuel cells but the advantageous fuel properties may also offer the opportunity

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to step up the potential of H_2 engines into the era of sustainable mobility and transportation.
Within the joint project “H_2 ICE Democar”, funded by the Federal Ministry for Economic Affairs and Climate Action, a consortium of industry companies and scientific institutions is identifying key requirements for a hybrid light duty commercial vehicle powertrain with a lean H_2 engine and hybrid powertrain. This innovative technology enables CO_2-free operation with long driving range, fast refueling, zero-impact exhaust emissions in combination with a high thermal efficiency H_2 engine. The potential of the technical solution will be demonstrated by a demonstrator vehicle for a logistics application (see **Fig. 1**). Besides engine and vehicle tests, simulations and experiments are carried out to gain a deeper understanding about the specifics of H_2 engine operation in a hybrid powertrain. Optimizing the combustion system in line with the development of a specific exhaust gas aftertreatment, a suitable hybrid operating strategy and engine control unit software functions allow for the holistic evaluation of the potential of a H_2 engine.

To maximize engine efficiency for high driving range in combination with low nitrogen oxide (NO_x) emissions, a lean combustion concept is implemented over the entire engine map using a low pressure H_2 direct injection system. Paired with a sophisticated boosting system, both key targets “high efficiency” and “lowest emissions” are feasible.

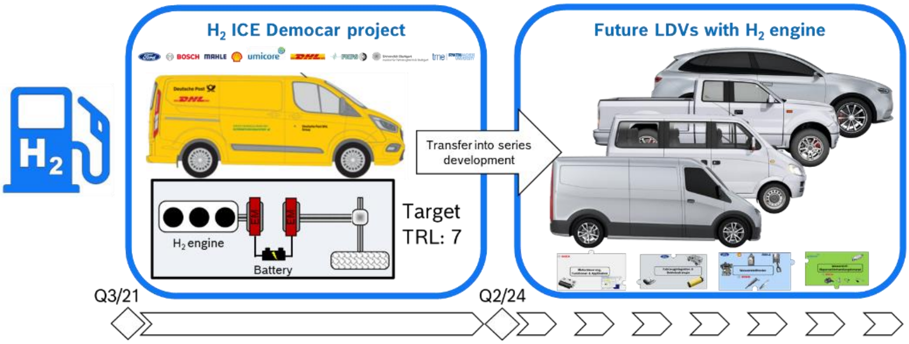

Fig. 1. Description of the project idea behind “H_2 ICE Democar”.

2 Engine Modification for hydrogen operation

The H_2 engine is derived from the well-known 1.0 L Ford EcoBoost three-cylinder gasoline engine [1]. It provides gaseous H_2 direct injection and is operated under lean operation conditions ($\lambda > 2$) over the entire engine map to minimize NO_x emission. **Table 1** summarizes the technical data.

For the dedicated engine operation with compressed H_2 the compression ratio has been increased to CR = 11.5. The cylinder head has been adapted for H_2 direct injection and the modifications allow for an optimized engine operation without compromising the robustness of the engine.

Special attention has been paid to high combustion peak pressure and potential irregular combustion events. The engine crank train has been reinforced utilizing a steel crank shaft and steel conrods. Also, the bearings have been enhanced for increased forces caused by the combustion process. The valves and the valve seats have been updated by MAHLE with material that is resistive for H_2 embrittlement and “dry” operation. An optimized piston with ring carrier for high wear resistance in the first ring groove and a salt core cooling gallery for efficient piston cooling has been introduced. For an absolute minimization of the oil consumption the cylinder honing has been optimized as well as the tribological system with matching piston rings.

For H_2 engines, a major challenge that must be accounted for is the fast combustion and the resulting engine-out NO_x emissions. Hence, a lean combustion concept with a minimum relative air/fuel ratio $\lambda > 2.0$ over the engine map has been selected. Lean operation with low exhaust gas temperatures demands for an adapted boosting system. Amongst various concept alternatives a single stage VGT turbocharger has been designed (see section 3) which enables lean operation even at full load. The specific engine power of 70 kW/l at rated power is achieved at n = 5000 1/min. Since the engine is used in a serial hybrid vehicle powertrain, the requirements for the “knee point” are less important because power demand can be compensated with a slight increase of engine speed.

Table 1. Technical data of target 1.0 L EcoBoost H_2 engine concept

Displacement / cc	999
Cylinder	3
Fuel	compressed H_2
Bore x Stroke / mm x mm	71.9 x 82.0
Compression Ratio	11.5:1
Fuel System	H_2 direct injection, centrally mounted
Ignition System	Cold spark plug, modified ignition coil
Cylinder Head	Aluminum
Cylinder Block	cast iron
Crankshaft	Steel
Connecting Rod	Steel
Piston	with cooling channel
Turbocharger	single stage VGT
Rated Power / kW	70

The initial H_2 engine tests were carried out with a Bosch series production injector for gasoline direct injection as a tool to enable H_2 direct injection (as done before in previous studies [2]). The series turbocharger has also been used as interim solution. Since the injector opening cross section is too small for H_2 operation, a high injection pressure of 150 bar was necessary to ensure adequate injection duration. The injector is in a centrally mounted position with a retraction of z = -4 mm to prevent jet/wall interaction (see section 4.1).

Selected test results with the initial engine set up are shown in **Fig. 2**. The vehicle target operation line for best powertrain efficiency in gasoline series operation (marked in orange) could be realized. This target operation line is just a reference since the optimal operation strategy for H_2 operation will be a result of the project.

Despite the filling losses, it was necessary to apply very early start of injection (SOI) timings during the opening phase of the inlet valves. Due to poor fuel/mixture preparation, later SOI timings resulted in cycle-to-cycle variation and combustion anomalies. The initial test results indicate that countermeasures have to be taken to improve mixture preparation (e.g. jet guiding caps, see section 4.5). This offers the opportunity for closed valve injection to optimize the trapped air mass and to further reduce the already low engine-out NO_x emissions. The emissions of unburnt hydrocarbons, carbon monoxide and carbon dioxide are already on a very low level.

Even with the series wastegate (WG) turbocharger, a lean engine operation (λ >2) can be realized in large parts of the engine map. Since NO_x-emissions are predominantly a function of λ (affecting the cylinder temperature), this enables very low emissions. Only at low-end torque and rated power, the standard turbo charger is limited to supply the boosting pressure for leaner mixtures, which leads to higher engine-out NO_x-emissions in the initial test. The results highlight the importance of the boosting system and furthermore of the serial hybrid operating strategy. The significantly reduced dynamic requirements of a serial hybrid system offer degrees of freedom for the boosting system design and for shifting operating points to more favorable engine map areas.

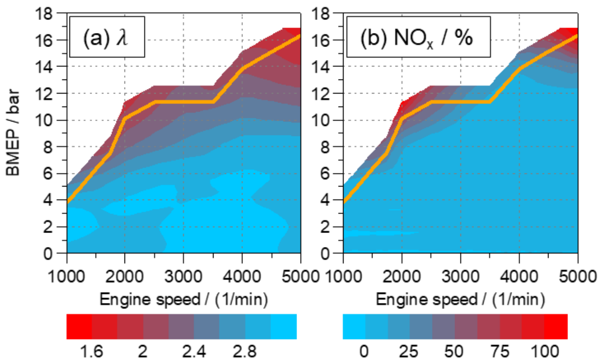

Fig. 2. Selected results of the first engine tests: (a) exhaust lambda and (b) engine-out NO_x emissions (normalized to the maximum value).

Even though the results shown so far indicate the high potential of the H_2 engine, they also highlight the key challenges that are addressed in this project:

- **Low-pressure H_2 direct injection**: Bosch provides dedicated designed H_2 functional injector samples. With these injectors, the required H_2 mass can be injected in short injection windows at low injection pressures. Closed valve injection can be applied to enhance trapping efficiency. Limiting injection pressures below 30 bar increases the fuel tank utilization and, hence driving range.
- **Improving fuel/mixture preparation:** Schlieren experiments under stationary conditions in combination with 3D-CFD simulations (see section 4) and engine testing create a deep understanding of the cause-effect relationships. This knowledge is

prerequisite to identify optimization potential for mixture formation and subsequently to achieve lowest possible emissions.

- **Boosting System**: The very wide ignition limits of H_2 in combination with a very high laminar flame speed enable an efficient homogeneous lean engine operation with low engine-out emissions. To fully employ this potential, as shown in **Fig. 2**, a powerful boosting system is required (see section 3).

3 Boosting system matching study

As discussed in section 2, increasing the relative air/fuel ratio towards lean engine operation conditions can be applied as a NO_x emission reduction measure. Increasing air/fuel ratio requires a higher boost pressure. At the same time, the exhaust gas temperature and thus the available turbine power are reduced. This leads to challenges in terms of the used boosting concept. Finally, the enleanment potential of the engine can be considered as a trade-off between available exhaust enthalpy and boost pressure generation. Therefore, the focus is set to efficiently convert the exhaust enthalpy into compressor power. Further enleanment can be realized, for example by an electrically supported turbocharger or an electrical compressor.

As a fundamental part of the project, different boosting concepts are investigated and compared. The evaluation focuses on achieving the performance targets while enabling lean engine operation, balance the control strategy complexity, support the vehicle package and manage system cost. The focus of this publication is set on the selected boosting concept (VGT turbocharger) and its comparison to the series boosting concept of the baseline engine (WG turbocharger).

3.1 Engine process simulation model

The boosting system study is performed with GT-Power. Based on a calibrated simulation model of the gasoline engine, a H_2 engine model is established. The in-cylinder wall heat transfer is modelled using the model by Woschni [3] and calibrated as a function of the air/fuel ratio. For a H_2 combustion process, an increasing wall heat loss can be expected from a smaller flame quenching distance compared to a gasoline combustion. The flame quenching distance increases with a higher air/fuel ratio. A Wiebe function is applied to model the heat release. The burn duration is derived from experimental data as a map depending on the air/fuel ratio and the engine speed. The increasing air/fuel ratio reduces the flame speed, and the burn duration increases. Especially for air/fuel ratios $\lambda > 2.0$, pre-studies of the series engine operated with H_2 showed no significant knocking tendency. Therefore, the center of combustion is controlled to the maximum allowed in-cylinder peak pressure.

The H_2 operated engine process model is calibrated and validated using available experimental data of the engine running with a series wastegate turbocharger. A model validation for λ-variations at different engine speeds, $n_{eng} = 2000\ldots4000$ 1/min and loads, BMEP = 4…16 bar, ensures a high accuracy in a wide engine operating range.

With the 0/1D hydrogen engine model a boosting system optimization has been carried out for H_2 operation. The optimization is based on a large turbocharger test data bench that allows to determine flow capacities and efficiencies utilizing empirically determined multipliers. An extended Zeldovich model [4] for NO_x emission evaluation is calibrated to engine-out NO_x measurements of the H_2 engine.

Preliminary to the boosting study, a valve timing optimization is performed at different engine speeds to maximize the trapping efficiency of the engine. As a result, the intake valve event length is reduced to 175 °CA referenced to 1 mm valve lift. The exhaust valve event length increased to 220 °CA (ref. 1 mm) to reduce pumping work.

3.2 Engine process simulation results

The compressor and turbine size are optimized at rated power conditions using an initially selected VGT turbocharger from the database of the tme, RWTH Aachen University. Therefore, the air/fuel ratio is defined to keep the engine-out NO_x emissions low. Including a defined safety margin on the maximum turbocharger speed, the compressor size is selected as small as possible. The turbine size is defined as small as possible with respect to the maximum allowed turbine inlet pressure and temperature. Especially for the enleanment potential at low engine speeds, a small turbocharger is required. Using the turbocharger optimization results, a well-matching part is provided by a turbocharger supplier.

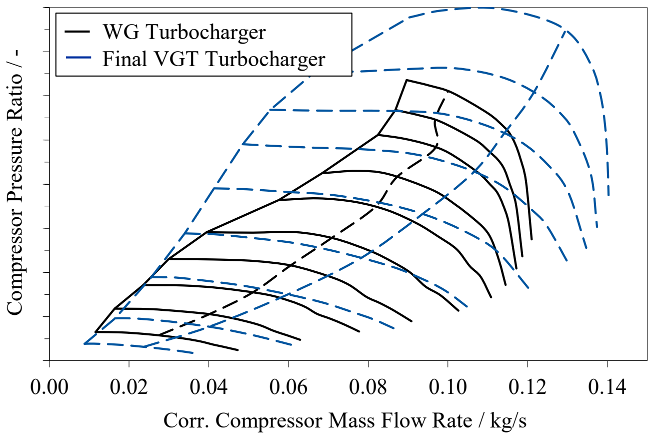

Fig. 3. Compressor map comparison of the series compressor with WG-turbine (black lines) against optimized compressor with VGT-turbine (blue lines).

Compared to the compressor map of the WG turbocharger, the new compressor wheel size was increased to $D_2 = 43$ mm providing a 16 % higher mass flow capacity. Especially for higher compressor pressure ratios, the compressor map width is increased which offers a higher flexibility in terms of the trade-off between low-end torque and rated power conditions. Furthermore, the maximum compressor efficiency has been increased by about 7 %.

The turbine wheel size is relatively small. Due to the stronger enleanment, the exhaust enthalpy reduces because of the lower exhaust gas temperatures. To provide the required turbine power, an increasing turbine pressure ratio is considered. This results in a larger wheel diameter ratio between compressor and turbine of $D_2/D_3 = 1.3$. Using the final turbocharger hardware, the air/fuel ratio is maximized in each operating point and enables engine operation leaner than lambda 2.

Especially with the final VGT turbocharger hardware, a further enleanment potential is achieved due to higher turbine efficiencies. Due to the low exhaust gas temperatures, the required turbine power is mainly generated by high turbine inlet pressure of up to 4.5 bar. At rated power the engine operation is limited by the maximum allowed cylinder pressure. Compared to the series WG turbocharger, with the VGT charger and lean combustion a significant NO_x reduction can be achieved at high engine speeds.

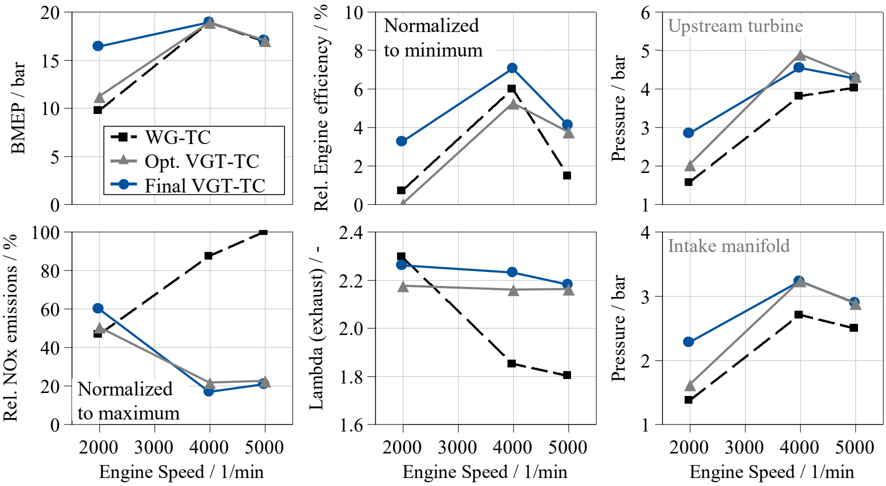

Fig. 4. Simulation results of the turbocharger matching study.

To improve the operating strategy of the engine, several design of experiments (DoE) approaches were performed to vary the engine load and the air/fuel ratio. The results shown in **Fig. 5** are based on the optimized VGT-turbocharger.

The optimal stationary operating strategy for the engine can be considered as a trade-off between the achievable engine torque and the engine-out NO_x emission reduction. On one side, a high air/fuel ratio is required to minimize the engine-out NO_x emissions to a minimum. Further, the effective engine efficiency can be increased as result of reducing wall heat losses. On the other side, the extractable turbine power is limited by the lower exhaust enthalpy. The VGT rack position closes and the pressure upstream turbine increases. The required pumping work increases and thus the required boost pressure. The optimal operating strategy of the engine is set by the limits for the optimal engine efficiency ($BSFC_{min}$) and the minimum engine-out NO_x emissions ($NO_{x,min}$). Under the limitation of a maximum engine-out NO_x emission threshold a maximum low-end torque of 71 Nm can be achieved. Considering five times higher NO_x engine-out emissions, the engine torque can be increased up to 100 Nm.

Overall, the operating strategy is mainly limited by the available exhaust enthalpy. Additional electrical support e.g. using an electrically supported VGT turbocharger or an electrical compressor, offers a significant improvement potential to increase λ for all operating conditions. The electrical support influences the enleanment potential especially at low engine speeds. In this part of the engine map, the required electrical compressor power is small compared to boost pressure generation at rated power.

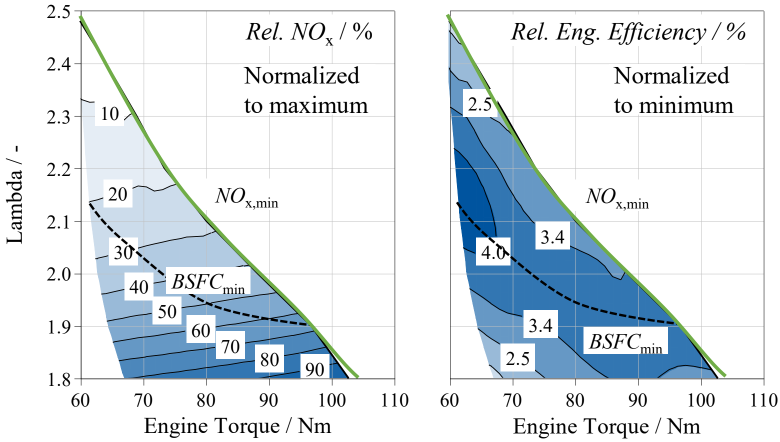

Fig. 5. DoE simulation results to determine the optimal operating strategy at n = 2000 1/min.

4 Optimizing mixture preparation

As discussed in section 2, the mixture preparation must be improved to realize a high engine efficiency and lowest engine-out NO_x emissions. To gain a deep understanding of the cause-effect relationship and to identify suitable measures, the H_2 gas jet is analyzed by high-speed Schlieren experiments in a constant volume chamber at Bosch in line with 3D-CFD simulations at tme.

4.1 Schlieren Experiments

The Schlieren measurement technique enables visualization of gas jets with a high temporal resolution of 50 kfps at a spatial resolution of 200 µm / Pixel. For safety reasons and for improving image quality, helium was used as a surrogate fuel for H_2. More detailed information to the experimental setup can be found in [5].

Previous publications have shown that the geometry of the near injector tip region influences the jet shape significantly. Therefore, a 3D printed cylinder head model was integrated into the chamber (see left hand-side of **Fig. 6**). This inlay enables the reconstruction of the later engine experiment at constant conditions. The model features all details such as spark plug, valves, and valve seats. It provides optical access even into the combustion chamber roof to visualize the jet already after the exit of the bore.

A simple solution to form the jet shape is retracting the injector in its bore. The right-hand side of **Fig. 6** shows the gas jet 1.3 ms after electrical actuation for three different

injector mounting positions at constant backpressure of 3 bar. Starting from a flush mounted position (retraction z = 0 mm), increasing retractions up to -6 mm have been measured. In case of the flush-mounted position, the jet directly follows the walls, increasing the risk of unwanted preignitions at hot parts, e.g. spark plug, exhaust valves, or the risk of backfiring by H_2 backflow into the intake manifold. Knocking pre-ignition can lead to very high peak pressures and temperatures and needs to be avoided. As further research on H_2 pre-ignition is required, the intuitive approach is to be safe and avoid jet contact with those surfaces. As **Fig. 6** illustrates, this can be achieved by a recess of z = -3 mm. The jet shows no trend to follow the walls, which results in a much deeper penetration. Setting higher retraction levels does not influence the resulting penetration length and the jet shape anymore. It can be concluded that a recess of z = -3 mm is optimal, since higher recess leads to an increase of dead volume.
Another way to form the jet is the use of jet guiding caps (JGC), which are mounted on the injector tip and allow to form other jet shapes with enhanced surface (multi-hole caps) and support the tumble charge motion (one hole in tumble direction).

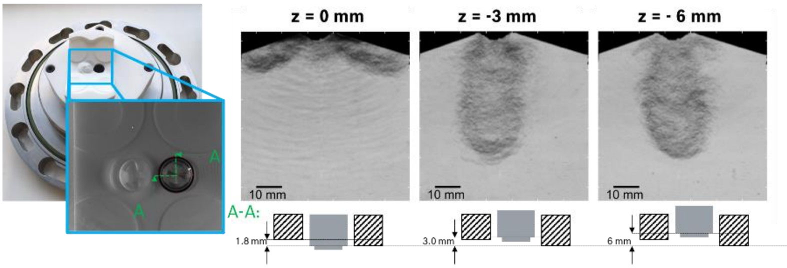

Fig. 6. Cylinder head model for the constant volume chamber and resulting jet shape for different injector recess positions (ensemble averaged, injection pressure 25 bar, back pressure 3 bar, 1.3 ms after trigger).

4.2 3D-CFD model validation of hydrogen injection

For the 3D-CFD modeling, CONVERGE CFD has been utilized. The model was validated by Schlieren measurements as discussed in section 4.1. Therefore, a reduced representation of the constant volume chamber has been derived from CAD, including detailed geometry of the injector internals. Time-resolved boundary conditions have been provided by Bosch for needle lift, pressure levels and H_2 mass flows. The chamber base mesh size was set to $\Delta = 2$ mm with fixed refinements along the injector axis and the needle gap was resolved with $\Delta = 0.125$ mm cells. Additionally, adaptive mesh refinement was utilized to refine the grid based on velocity or H_2 concentration gradients to $\Delta = 0.5$ mm, resulting in a varying cell count between 2.3 and 2.6 Mio. For most simulations, a two-equation RANS k-epsilon RNG turbulence model was chosen. Local mixture averaged coefficients account for the more pronounced diffusivity of H_2 (low Le-number effect, [6]). **Fig. 7** shows the agreement with the measurement data for a configuration without cylinder head inlay and with an injector recess of z = -2 mm. The left image shows the Schlieren image of the jet. The graphic in the middle depicts the result of the corresponding 3D-CFD, illustrating a H_2 iso-surface with a gravimetric concentration of $y_{H2} = 3\,\%$. As well the overall jet shape is in good agreement. Even the residue of the initial jet roll-up close to the injector tip (at about z = 15 mm) is visible in both representations. Furthermore, the graph on the right demonstrates the excellent agreement of the vertical, gaseous penetration length of the H_2 jet.

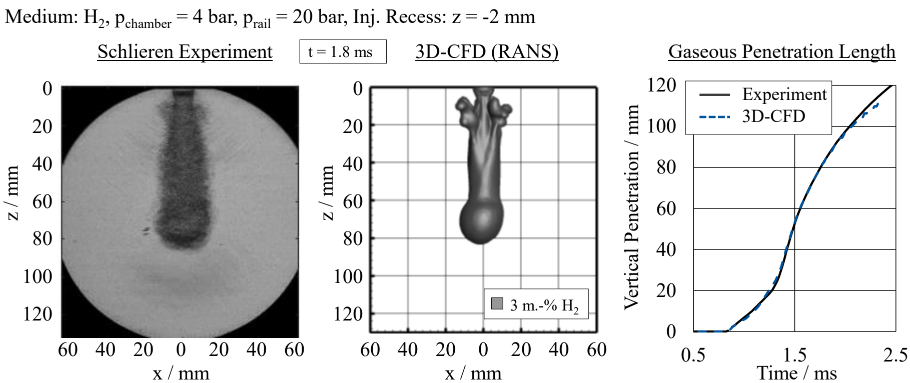

Fig. 7. Qualitative and quantitative validation of the 3D-CFD H_2 injection model.

4.3 Investigated load point and methodology

As seen from the investigation in section 2, especially in **Fig. 2**, high engine load is most challenging with respect to mixture formation, indicated by high engine-out NO_x emissions. This stems from long injection windows with the not dedicated H_2 injectors and less time for the mixture homogenization process. Therefore, the 3D-CFD study focuses on the rated power operating point, which was chosen as of **Fig. 8**. On the right, the valve-lift profiles along with the H_2 injection window are illustrated. The start of injection was chosen at SOI = 530° CA after firing top dead center ($aTDC_F$) to realize a remaining intake valve lift of $h_v = 2\,mm$ as first studies had shown a minor impact on cylinder filling with this injection timing, thus maximizing time for mixture homogenization with minimal additional boost pressure required.

Data	Unit	Value
Engine speed	1/min	5000
p_{me}	bar	17
λ	1	2.2
Injected Fuel Mass	mg	12.3
Start of H_2 Injection	CA $aTDC_F$	530
Residual gas	%	3.8
External EGR	%	0
Compression Ratio	1	12.0

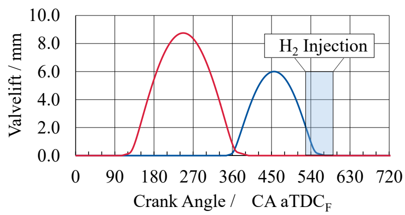

Fig. 8. Overview of the chosen operating point for mixture formation analysis.

The 3D-CFD simulation approach is carried over from the chamber validation, see section 4.2. Time-resolved boundary conditions at intake and outlet as well as initial values were carried over from the 1D gas exchange simulation, see section 3. Simulations were carried out between 270° CA $aTDC_F$ to 750° CA $aTDC_F$.

4.4 Injector recess variation in engine configuration

As shown in **Fig. 6**, injector recess can have a drastic influence on jet penetration and shape as there can be a strong interaction with the cylinder head. This effect was also observed in the 3D-CFD gas exchange simulations with the validated injector model. The upper row of **Fig. 9** illustrates the H_2 iso-surface at a value of $y_{H2} = 2\,\%$, indicating the H_2 distribution at 40° CA after actuation for three injector recess positions.

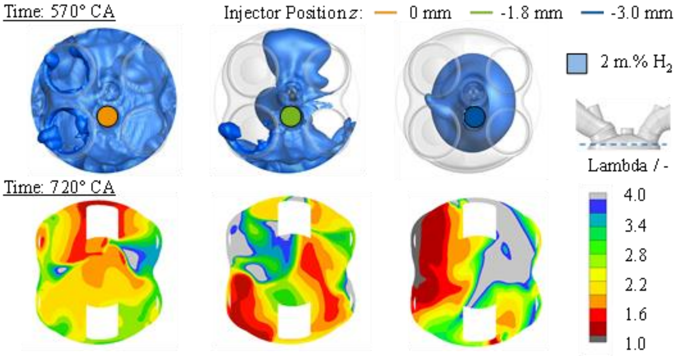

Fig. 9. Evaluation of H_2 jet formation 40° CA after SOI and mixture homogeneity depicted by a vertical cut section at TDC_F for an injector recess variation.

On the most left, no recess is realized, in the middle the injector is retracted by z = -1.8 mm and the right images show the model response to z = -3 mm recess. For z = 0 mm recess, most of the injected H_2 is sticking to the combustion chamber walls. Due to the asymmetric situation at the injector bore, the medium retraction of z = -1.8 mm also leads to asymmetric wall effects with the tendency of flow along the roof line toward the spark plug. The retraction of z = -3 mm eliminates, as observed in the chamber experiments, these wall effects and a clean jet protrusion can be observed. With respect to the discussed risks, an injector recess of z = -3 mm is clearly advantageous over the other two configurations. As engine-out NO_x emissions need to be reduced as far as possible, lowering the demand on the exhaust aftertreatment system, the homogeneity of the mixture is of key importance and should always be considered when comparing design layouts and operating strategies. As in this stage of the project no combustion simulations have been carried out, the tendency of NO_x formation needs to be correlated with the occurrence of fuel rich zones in the combustion chamber.
The lower row of section cuts in **Fig. 9** illustrates the distribution of the relative air/fuel ratio (λ) at TDC_F in a vertical section 3 mm above the head gasket plane. Dark grey and dark red areas indicate fuel rich zones, dark grey even below $\lambda = 1$ while blue and light grey areas depict lean areas. It can be observed that the mixture is much more stratified with increasing injector recess. This can be explained by a better circumferential H_2 distribution due to the wall interaction. While the penetration for the recess of z = -3 mm is much higher and there is a certain interaction with the piston, the overall level of the charge motion ("Tumble") is not strong enough to distribute the H_2 throughout the combustion chamber. While the cross sections shown in **Fig. 9** shows a valid trend for the

vertical distribution, a histogram analysis is always helpful to quantify the results (see **Fig. 10** for λ at TDC_F). In addition to the raw histogram, the values of the coefficient of variation ($COV(\lambda) = \sigma(\lambda)/\bar{\lambda}$) and the mass fraction of the cells below $\lambda = 1.6$ are given. The pronounced stratification with z = -3 mm is also reflected by the increased amount of mass in the richer bins. The overall λ distribution of the z = 0 mm variation looks fairly symmetric, which is also reflected by a low COV (λ).

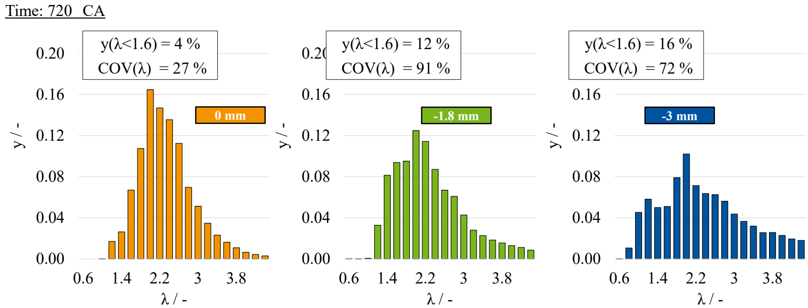

Fig. 10. Stratification analysis for injector recess variation.

As discussed, the strong stratification leads to higher engine-out NO_x emissions. With a strong spread between λ = 1 and λ > 4 inside the combustion chamber (as shown for z = -3 mm), also significant cyclic variations are to be expected due to the strong impact on H_2 flame speed. Since the other two variants impose challenges with respect to back-fire and surface pre-ignition risk, no robust candidate has been identified. It can be noted, that an earlier SOI will help with mixture homogenization, but at the cost of significant boost pressure requirement. The study was carried out under the objective to only utilize early SOI as fallback solution.

4.5 Evaluation of the potential of Jet Guiding Caps

The investigated functional sample of the H_2 injector can be equipped with jet guiding gaps. Especially for systems with low or swirl-type charge motion, JGC can significantly help improving the mixture homogeneity. Multiple layouts have been investigated, the working principle of three of them is shown in **Fig. 11**.
The baseline with z = -3 mm recess has been depicted, followed by a 6-hole design and two single hole caps which are oriented in and against the Tumble direction (intake is depicted on the left). The multi-hole design V1A tried to realize the good circumferential distribution of the variant without injector recess. Cap V2A (Tumble) supports the charge motion as desired, also increasing the turbulent kinetic energy at spark timing. Utilization of the Tumble-supporting cap has resulted in a better homogeneity with respect to COV and significantly reduced the richest zones in the chamber. Another significant improvement was achieved by the multi-hole Cap V1A with COV(λ) = 34 % and especially COV(λ<1.6) = 5 %, already improving the best result of the injector recess study. Cap V2B destroyed the already low charge motion inside the combustion chamber and has demonstrated the worst homogeneity with respect to fuel rich zones of all investigated variants (please refer to the summarizing **Table 2**).

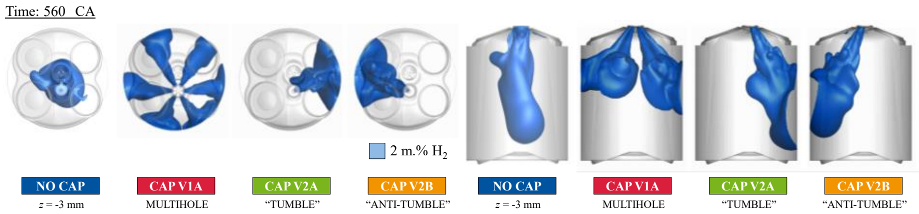

Fig. 11. Overview of investigated spray caps.

As a next step, the hole layout of the multi-hole cap was optimized. Maintaining a certain ratio of open area through the holes compared to the area at the needle gap, 7 hole layouts were investigated. Selected out of this investigation, the best candidate will be discussed: Cap V1B features a smaller centered, axial hole compared to the six outer holes, the diameter was reduced to realize comparable flow through all holes as the center hole tends to show lower flow losses due to separation effects. A visualization of the injected H_2 as well as the resulting stratification investigation is shown in **Fig. 12**. The center hole features a slightly lower flowrate, realizing less absolute penetration compared to the outer holes. Due to the angle of the outer holes, the vertical penetration is comparable for all holes at 560° CA. As the amount of charge motion is fairly low, having a lot of H_2 close to the (vertical) center of the combustion chamber is beneficial.

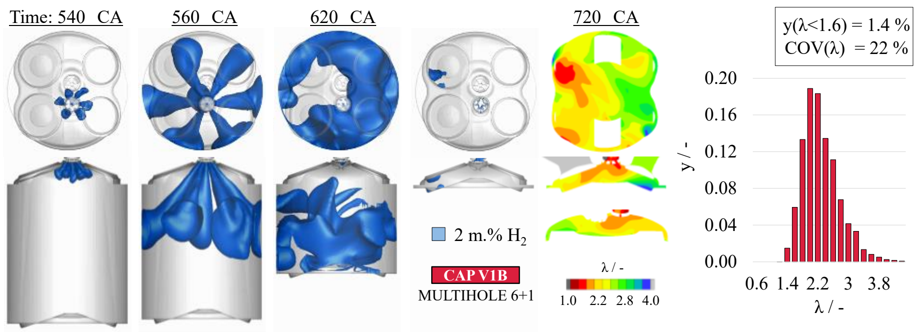

Fig. 12. Analysis of JGC V1B “Multi-Hole 6+1”.

Compared to the baseline of **Fig. 10**, the section cuts of the relative air/fuel distribution illustrate the better homogeneity, however there are still some slightly richer areas observed. Especially the cap itself cannot be fully scavenged, leading to a local λ in the order of unity. Therefore, it must be noted, that the JGC installation could introduce a potential source of pre-ignition in case they are not well-designed thermally.
A summarizing overview on the 3D-CFD studies is given in **Table 2**. It can be concluded that by introducing a multi-hole JGC design several risks have been eliminated while significantly improving mixture homogeneity, even without optimizing the intake port design or advancing the start of injection. Evaluation of $\phi = \lambda^{-1}$ was added as it can deliver additional information with respect to the shape of the distribution and avoidance of fuel rich zones. For equal values of COV(λ) and COV(ϕ), a somewhat symmetric shape of the distribution can be expected.

Following a baseline investigation without JGC (at z = -3 mm), the JGC V1B "6+1 Hole" and V2A "Tumble" have been chosen by the consortium to be applied in thermodynamic testing. The results will be mirrored again by updated 3D-CFD.

Table 2. Overview of key performance indicators for 3D-CFD mixture formation analysis.

	COV(λ)	COV(ϕ)	λ < 1.6	λ < 1.8	Wall Interaction	Back-Fire Risk	Early H_2 at Spark
No Cap, z = 0.0 mm	27%	25%	4%	11%	--	--	--
No Cap, z =-1.8 mm	91%	37%	12%	21%	-	-	--
No Cap, z =-3.0 mm	72%	49%	16%	22%	+	++	++
V1A "6 Hole"	34%	24%	1%	5%	++	++	++
V2A "Tumble"	40%	28%	6%	19%	++	++	++
V2B "Anti-Tumble"	46%	39%	16%	25%	++	++	++
V1B "6+1 Hole"	22%	20%	1.4%	7%	++	++	++

5 Summary and Outlook

5.1 Summary

This paper focuses on important aspects of the public funded "H_2 ICE Democar" project. Early engine test results discussed in section 2 confirm the potential of a very lean H_2 engine operation, which enables low engine-out NO_x emissions and a reduction of fuel consumption. All engine operation reference points derived from a vehicle with gasoline engine hybrid powertrain could be realized in H_2 operation with – if operation with $\lambda > 2$ was possible - low engine-out NO_x emissions. However, the initial results also highlight the importance of the turbocharging system and the optimization of mixture formation.

Section 3 outlines the model-based selection process of the turbo charger at tme. A single stage VGT turbo charger was selected for package reasons in the demonstrator vehicle. This turbo charger shows the potential to achieve a very lean mixture ($\lambda > 2$) even at rated power and hence to further reduce engine-out NO_x emissions.

By detailed 3D-simulations at tme, supported by Schlieren measurements at Bosch, in section 4, a deeper understanding of mixture formation is created. Finally, the potential of jet guiding caps is demonstrated with regards to mixture formation. In a next step, the effect of the most promising JGC variants on combustion stability (but also preignition) and engine-out emissions will be evaluated during multicylinder engine testing.

5.2 Outlook

Further steps of the project include the definition of a dedicated exhaust aftertreatment system for the H_2 engine and optimization of the operating strategy of the serial hybrid powertrain, using all degrees of freedom provided by H_2 operation. An H_2 pressure tank (installed on the vehicle roof) and fuel system for low pressure H_2 direct injection was already designed. All components are currently integrated into two demonstrator vehicles (transporter for the target logistic application and a people mover) with the outlined serial hybrid powertrain (see **Fig. 13**). With the finalized vehicle calibration, the performance of the "H_2 ICE Democars" will be evaluated on the chassis roller dyno and test track.

Fig. 13. H_2 ICE Democar (people mover version), already equipped with the fuel storage system on the vehicle roof.

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