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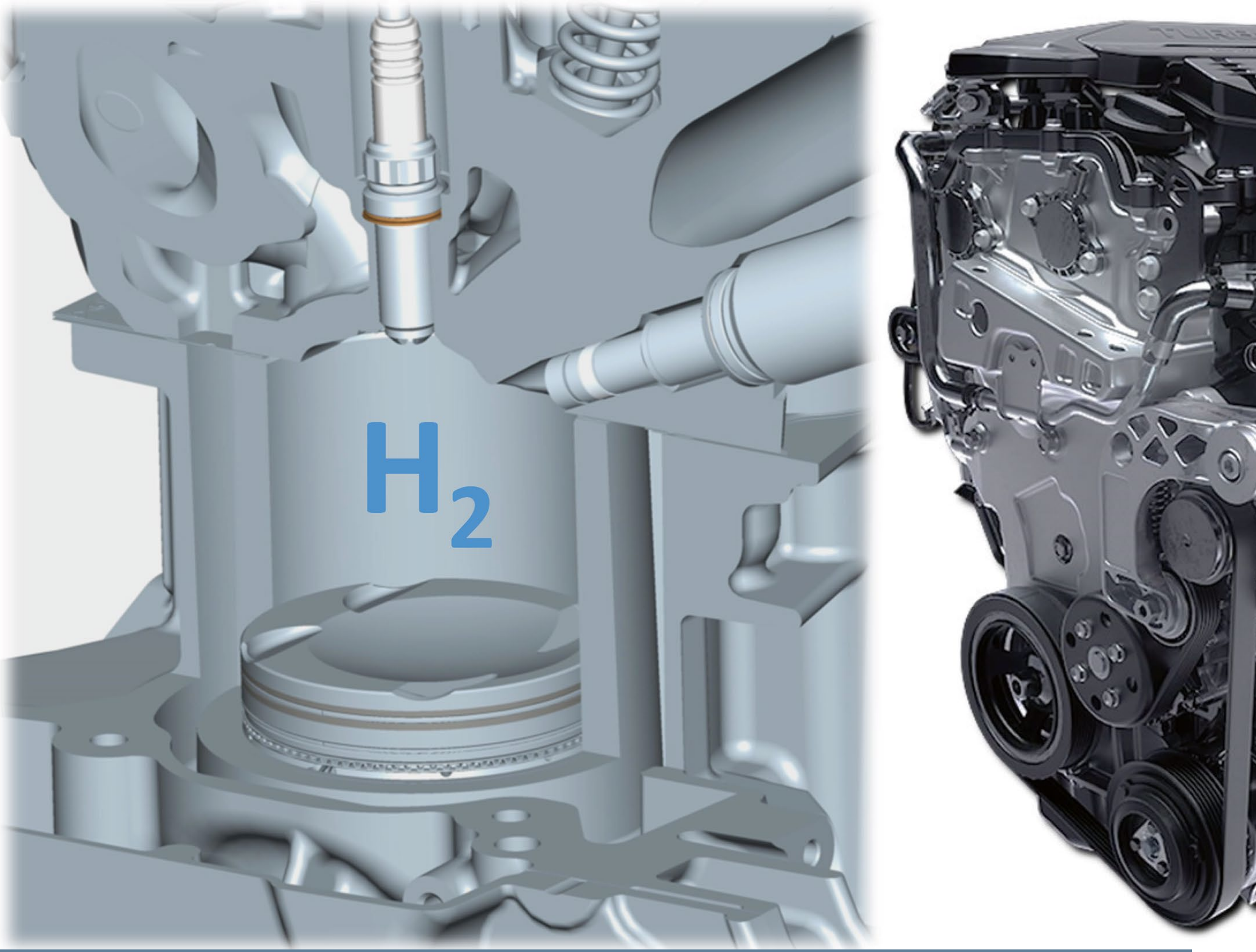
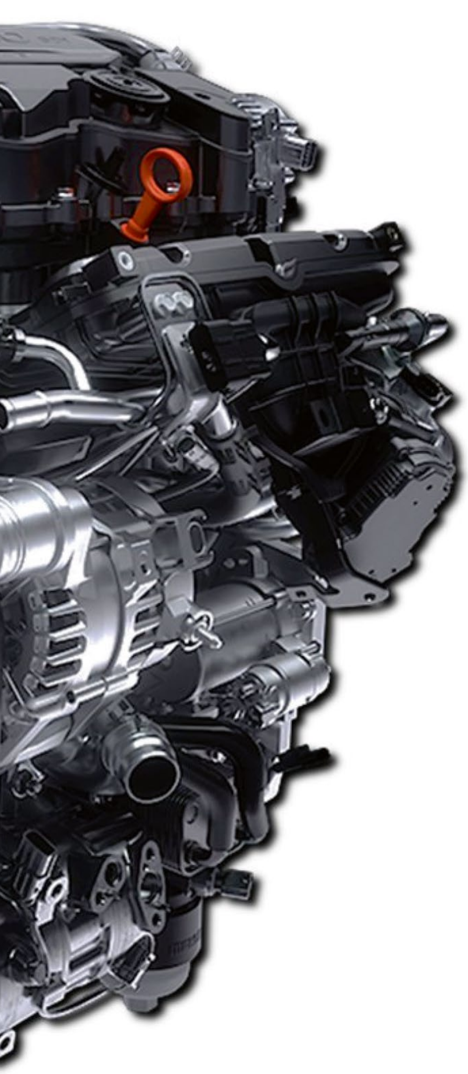


Hydrogen Engine for a Passenger Car Hybrid Powertrain

Complementary to fuel cells, hydrogen engine powertrains are considered as a potentially attractive solution for hydrogen mobility. Aramco, Bosch and Hyundai Motor Group investigated the potential of a passenger car hybrid powertrain with an H_2 engine for a D-segment SUV. A Hyundai series production four-cylinder 1.6-l T-GDI engine was converted to operate with hydrogen in a lean burn engine concept.



For the substitution of fossil fuels with CO_2 neutral alternatives, hydrogen will become an important energy carrier of the future. For on-road mobility applications, hydrogen will contribute to well-to-wheel and life cycle CO_2 neutral solutions. Important automotive markets like China, the United States, EU, Korea and Japan have already started and will accelerate the build-up of a hydrogen refueling infrastructure.

Following a technology open approach, and complementary to fuel cells, hydrogen engine powertrains are considered as a potentially attractive solution. For passenger car applications, the integration of hydrogen engines into hybrid powertrains is of particular interest because it creates significant functional benefits, especially in terms of maximum driving range. A joint research between the partners Aramco, Bosch and Hyundai Motor Group is established to investigate within a demonstration project the potential of such a system.

This article presents the hydrogen adaptation of the combustion system for a 1.6-l turbocharged mass production gasoline engine. It includes the simulation-supported alignment of hydrogen specific components, the integration of hydrogen subsystems as well as an empowered charging concept to enable the lean burn combustion. The results demonstrate feasibility and attractive characteristics of hydrogen engine powertrains for future passenger cars.

TABLE 1 Hyundai base engine (Hyundai)

Engine type	–	In-line four-cylinder
Displacement volume	cm³	1598
Bore × Stroke	mm	75.6 × 89.0
Valve train	–	Intake CVVD, intake and exhaust CVVT
Power	kW	132
Torque	Nm	265
Compression ratio	–	10.5

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BASE ENGINE AND DEVELOPMENT TARGETS

The series production four-cylinder, 1.6-l gasoline engine, which is equipped with 350 bar direct injection system and wastegate turbocharger, is converted to operate with hydrogen as fuel. Main technical features and specifications of this engine are summarized in **TABLE 1**. It comes with a Continuously Variable Valve Timing (CVVT) system on intake and exhaust side as well as with a Continuously Variable Valve Duration (CVVD) system on intake side, which is capable to vary the valve opening duration. This combination allows a high control flexibility in valve events to enhance fuel economy, low end torque and catalyst heating performance.

The hydrogen variant of the engine is developed for a hybrid dedicated power-

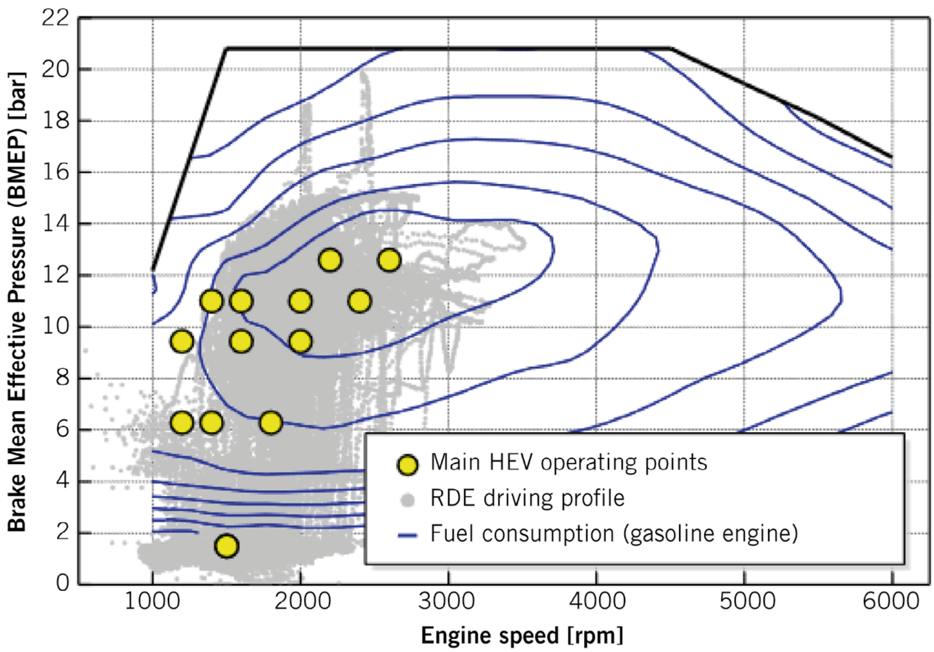

FIGURE 1 Exemplary RDE profile and twelve main test points of D-segment SUV (HEV) (© Hyundai)

train in D-segment SUV class to enhance the competitiveness in fleet CO_2 emission by achieving zero-CO_2 in this highly challenging vehicle segment.

Beside the zero-CO_2 capability, an elaborated design and calibration is focused on minimizing NO_x emissions. An exhaust aftertreatment system using SCR as core technology is introduced to provide sufficient potential for complying with expected Euro 7 emission regulations. The hydrogen engine is also aiming to achieve the same full load performance as base gasoline T-GDI engine, with 132 kW maximum power and 265 Nm maximum torque.

For the evaluation, twelve representative test points from the main hybrid operating area of the engine map are selected from various test cycles with the D-segment SUV and shown in **FIGURE 1** for an exemplary RDE drive cycle. The optimization of the control parameters is especially focused on these points to assess the thermal efficiency and emission potential.

CONVERSION TO HYDROGEN ENGINE

A Bosch prototype injector for low pressure H_2 direct injection is integrated into the prototype engine. This injector is driven by a solenoid actuator and has an outward opening nozzle to achieve a high flow rate, which is required for a low density gaseous hydrogen injection. The injector flow rate Q_{stat} of 7.9 g/s is chosen, considering an injection window of ~90 °CA for the required injection quantity at rated power. As visible from **FIGURE 2**, this Q_{stat} selection allows to keep the maximum H_2 injection pressure on a relatively low level of ~25 bar in order to improve the vehicle driving range.

ADAPTATION OF BOOSTING SYSTEM

The key control strategy of the hydrogen engine is to realize a highly lean combustion throughout the entire operating range for enhanced thermal efficiency, extremely low NO_x emissions and to achieve competitive full load performance by mitigating abnormal combustion, such as pre-ignition and knocking.

Therefore, one main development focus is on the optimization of the boosting system coming from Garrett

Features	
Static flow rate	~6-10.5 g H_2/s at 25 bar(a)
Minimal dynamic flow	1.1 mg H_2/stroke
Supply voltage	12 V (~40 V in boost phase)
Pressure range *Target*	~8-35 bar(a) *up to 40 bar(a)*
Jet preparation	Outward opening nozzle (optional jet guiding caps)

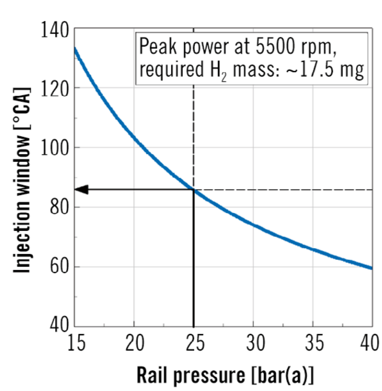

FIGURE 2 Bosch H_2 DI injector prototype and selection of the static flow rate Q_{stat} (© Bosch)

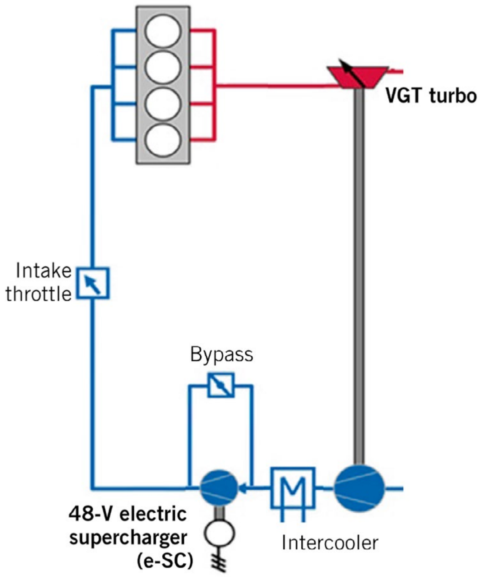

FIGURE 3 Layout of two-stage boosting system (© Hyundai)

Motion to address the much higher air mass requirement compared to a regular turbocharged gasoline engine. The serial wastegate turbocharger is replaced with a Variable Geometry Turbocharger (VGT), which is combined with a 48-V driven electric supercharger (e-SC). In this setup, the VGT is used as primary boosting device and is located downstream of the intake air filter. An intercooler is placed between the VGT and the e-SC to cool down the compressed charge air before entering the e-SC, which is mainly activated in the low speed area. The schematic of the boosting system layout is illustrated in **FIGURE 3**.

ADDITIONAL ENGINE MODIFICATIONS

In addition to the H_2 injection system and the enhanced boosting system, an active crankcase ventilation system from Mahle is integrated to ensure a hydrogen concentration below the explosion limit in the engine crank case.

Due to the wide ignition limits of hydrogen and to prevent undesired combustion effects like pre-ignition, a dedicated cold spark plug and a modified ignition coil are introduced. In order to adapt the hydrogen injection components and to improve the cylinder filling, the tumble charge motion and the hydrogen jet homogenization, the cylinder head needs to be modified. In the following, the simulation supported optimization steps for the intake ports, the injector pocket and the tip position are described in detail.

SIMULATION-BASED CYLINDER HEAD DESIGN

At an early development stage, the intake port has to be designed, with respect to the demand of a high excess air ratio for the lean combustion on the one hand and for the best possible hydrogen jet homogenization on the other hand. Therefore, two intake port geometries are chosen and evaluated using CFD simulations, **FIGURE 4**. One port is focused on a high cylinder filling and the other one on reaching a high tumble level.

The difference for the trapped air in the cylinder is less than 1 %, **FIGURE 4** (left) for both intake port variants, but significant for the achieved tumble levels. So, the high tumble intake port creates an around 25 % higher tumble ratio, **FIGURE 4** (middle) and is resulting in higher mixture uniformity, **FIGURE 4** (right). The uniformity index of 1 indicates a perfectly homogenized mixture and the relation between this index and the mixture quality is strongly nonlinear. Thus, also small differences indicate a significant impact on mixture formation. Considering the improved mixture formation and the negligible disadvantage in cylinder filling, the high tumble intake port is selected for further investigations.

After fixing the intake port design, the integration of the hydrogen injector and especially the injector pocket design as well as the injector recess is developed. Two different shapes for the injector pocket are chosen, the Very wide sides and the Very narrow variant. The Very wide sides pocket is designed to direct the hydrogen jet to the sides of the cylinder laterally. The Very narrow pocket is basically a cone and designed for a more focused jet guiding passing potentially hot parts as spark plug and exhaust valves. Each pocket variant is combined with two different injector tip

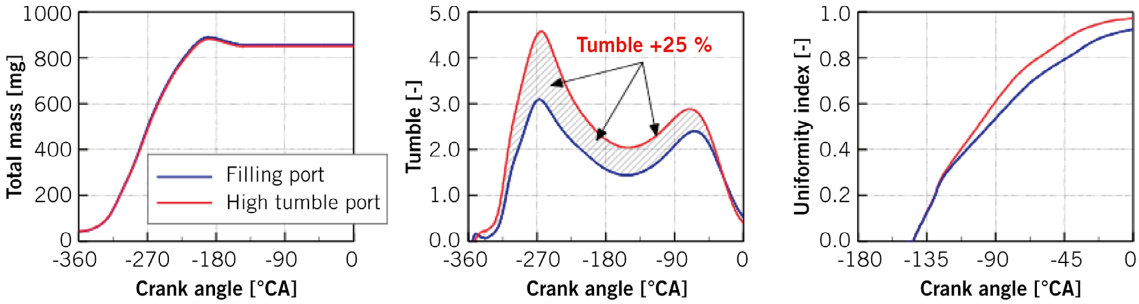

FIGURE 4 Evaluation of trapped air mass, tumble levels and mixture uniformity (© Hyundai | Bosch)

positions, flush mounted and with a recess of -2 mm.

For evaluation of the CFD-setup, the gas jet propagation is characterized by a high-speed Schlieren measurement with fixed geometry, which includes the whole pent roof of the combustion chamber to detect a possible jet attachment due to the Coandă effect.

The Schlieren results, **FIGURE 5** (upper row) show that the Very wide sides pocket has a lower jet penetration, specifically with the flush mounted injector. In opposite, the Very narrow pocket has a higher penetration, which is reinforced by the injector recess. For the flush mounted cases, the missing rotational symmetry of the injector pocket cavity becomes visible also in the jet topology. A recess of -2 mm is sufficient to redirect and focus the jet into the injector axis direction as well as strongly re-establish the rotational symmetry. In summary, it can be concluded that a narrow injector pocket as well as the recess are focusing the jet momentum in the injector axis direction most strongly.

In the first two rows of **FIGURE 5**, the high conformity of the Schlieren measurements and the CFD simulation can be seen. After extensive approximations especially for the hydrogen jet initialization, the jet shape and direction as well as the differences in the jet propagation are well represented by the CFD. The bottom row of **FIGURE 5** shows the lateral propagation of the hydrogen jet between the injector pocket variants. Also, from this perspective, the Very narrow pocket seems to be advantageous for avoiding jet impingement on the intake valves in case of open valve injection.

In addition, a dynamic CFD simulation is conducted at a representative operation point to evaluate the injector pocket and recess in terms of mixture formation. For the simulation, the injection is started after intake valve closing to avoid a backflow into the intake ports. The CFD simulation shows, that due to the high mass flow rates of the dedicated

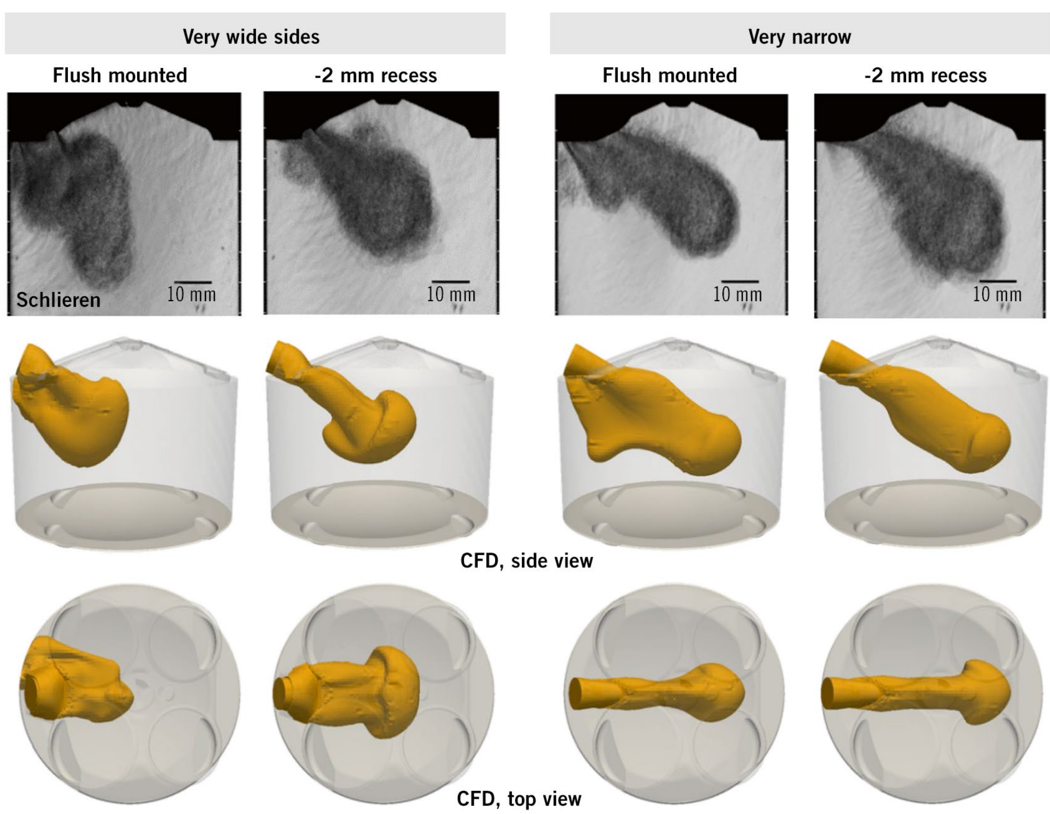

FIGURE 5 H_2 jet evolution in a stationary geometry: upper row: Schlieren pictures; lower two rows: iso-surface for H_2 mass fraction of 0.2 (CFD) (© Hyundai | Bosch)

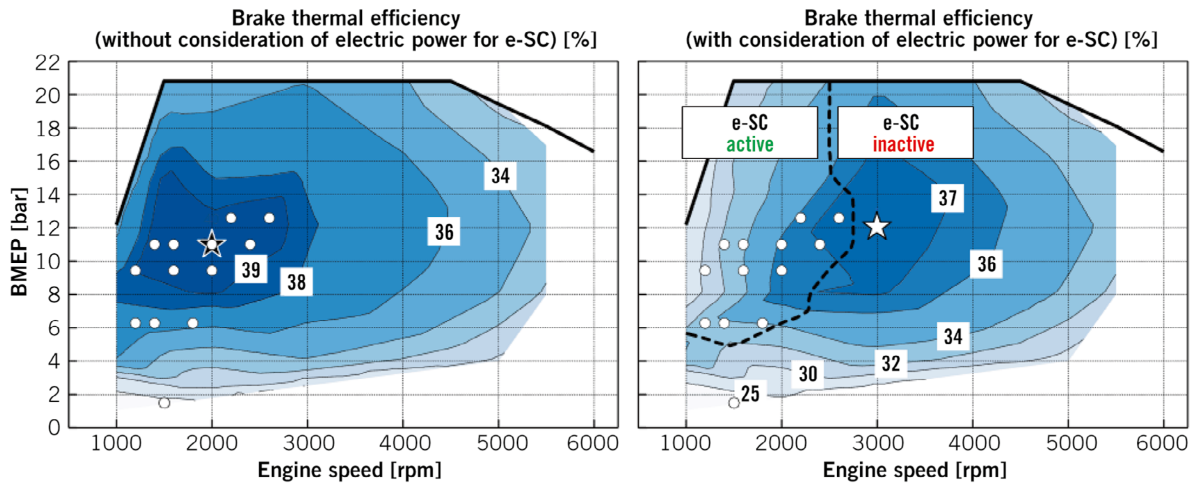

FIGURE 6 Brake thermal efficiency without (left) and with consideration of the electrical power demand for the e-SC (right) (© Hyundai | Bosch)

H_2 injector it is possible to provide the hydrogen mass with sufficient time for mixture formation before the ignition.

Overall, both Schlieren and CFD results indicate advantages for the Very narrow injector pocket in combination with the recess of -2 mm in terms robustness due to the most compact jet shape.

ENGINE CALIBRATION PROCESS

In order to evaluate the selected engine hardware configuration, both parameters for the air and fuel paths need to be optimized to demonstrate the full potential of the H_2 engine.

In the first step, the air path is calibrated for the target air-fuel ratio (λ) of ~2.5. Therefore, the share of the boosting devices is optimized with respect to the best overall system efficiency by finding the trade-off between increasing gas exchange losses by closing the VGT and the electric power consumption by the e-SC.

As a second step, the cam timings of the CVVT on exhaust and intake side as well as the intake valve opening duration by CVVD is calibrated for an optimized cylinder filling and therefore a high excess air ratio. For the intake valve openings, generally the early timings seem to be beneficial in terms of high excess air ratio and therefore high thermal efficiency and low NO_x emission. In conjunction to this, late exhaust valve closing lead to a large valve overlap and high air excess ratio, but also significantly increase in pumping losses. The best compromise between low NO_x emissions and high thermal efficiency is calibrated.

The variations of the intake valve event length (by CVVD) show an optimum with respect to cylinder filling, which is very close to the shortest possible event length for most operating points.

In the last step, after calibration of the air path parameters, the fuel path is calibrated. The hydrogen fuel pressure is set to 25 bar and only lowered, when a limitation by the minimum injection time is given. The start of injection (SOI) is varied from 220 °CA, which is around the intake valve closing, to ~90 °CA before top dead center. Very early SOIs before complete closing of the intake valves lead to an increased demand in boost pressure, as the injected hydrogen mass uses volume from the fresh air mass. Moving to later SOIs yield higher thermal efficiency at constant low NO_x

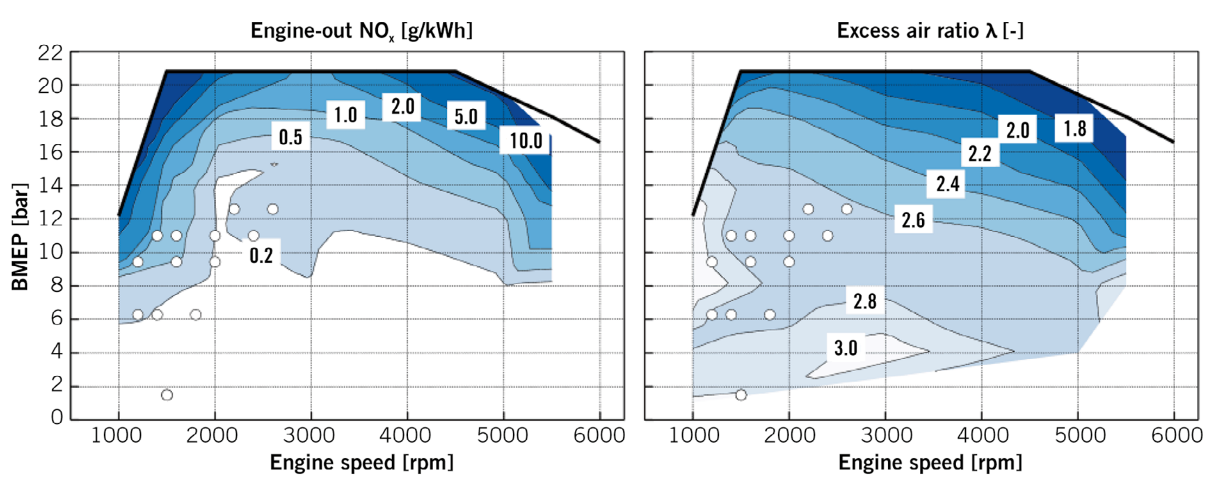

FIGURE 7 Engine-out NO_x (left) and excess air ratio (λ) (right) (© Hyundai | Bosch)

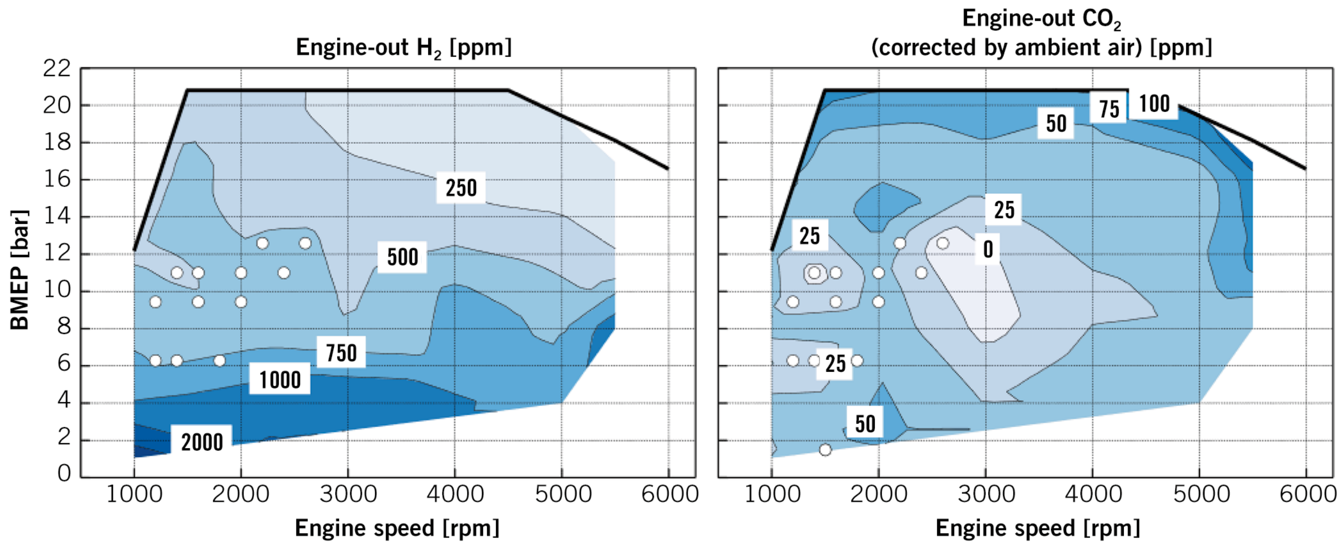

FIGURE 8 Engine-out H_2 (left) and CO_2 emissions (right) (© Hyundai | Bosch)

and H_2 emissions. This constant low emission level over a wide range of injection timing is an indicator for the good mixture homogenization in consequence of the high tumble level. At a certain point, after the best point at ~100 °CA, the NO_x emissions are increasing dramatically. Here, in addition to the lack of time for mixture formation, the injected hydrogen yet does not hit the tumble impulse any longer, which lead to this tear-off in homogenization behavior.

The ignition angle is adjusted to the optimal combustion efficiency at a MFB50 of ~8 °CA as long as there is no limitation in maximum peak pressure of the engine and the NO_x emissions are on a suitable level.

ENGINE MAP RESULTS

After the calibration optimization for a wide set of engine operation points, an overall engine map can be derived. The achieved brake thermal efficiency with and without the consideration of e-SC electrical power demand is shown in **FIGURE 6**.

If the electrical power demand of the e-SC can be neglected, for example due to provision of regenerative braking energy by the hybrid powertrain, a maximum brake thermal efficiency of up to 40 % and higher than 39 % for the majority of the key operation points can be achieved, **FIGURE 6** (left). Under consideration of the electrical power demand for the e-SC a system efficiency of 37.8 % is reached. The shift of the area with highest efficiency can be explained by the operational area of the e-SC, which is mainly at lower engine speeds and shown in the map of the overall system efficiency, **FIGURE 6** (right).

In terms of emissions, the engineering target for engine out NO_x of 0.2 g/kWh is met for a wide range of part load operating points, **FIGURE 7**, since a sufficiently high excess air ratio is applied with the selected boosting system.

FIGURE 7 also shows elevated engine out NO_x emissions despite apparently high excess air ratio in some Low-End Torque (LET) operating points of the engine map. Here, the excess air in the combustion chamber most likely differs from the excess air ratio measured in the exhaust manifold due to filling- and NO_x-optimized valve overlap calibration.

Similar to the engine out NO_x, the unburnt H_2 in the exhaust is on a low level over a wide range of the engine map and are illustrated in **FIGURE 8** (left). On the contrary to NO_x, the H_2 emissions have the highest potential for improvement at very low engine speeds and loads.

Under consideration that the concentration in the ambient air of the test cell is subtracted from the engine out CO_2, the CO_2 concentration is close to zero in large areas of the engine map and lower than 115 ppm, which corresponds to a value of less than 1 g/kWh at the highest point, **FIGURE 8** (right)

The HC and CO emissions (not shown here) are overall very close to the detection limit, considering the same correction of the concentration by the ambient air. It can be concluded that only a very low amount of engine oil is burned. The low engine out emissions HC and CO as well as H_2 in a lean exhaust gas can be converted very effectively by the well-known oxidation catalyst technology that will be part of the hydrogen exhaust aftertreatment system.

SUMMARY AND OUTLOOK

Complementary to fuel cells, hydrogen engine powertrains are considered as a potentially attractive solution for hydrogen mobility. Aramco, Bosch and Hyundai Motor Group are investigating the potential of a passenger car hybrid powertrain with a H_2 engine for a D-segment SUV. A Hyundai series production four-cylinder 1.6-l T-GDI engine was converted to operate with hydrogen in a lean burn engine, including a modified ignition system and an active crank case ventilation. In order to provide high air mass flow, the air path was equipped with a two-stage boosting system based on a VGT turbocharger combined with a 48-V e-SC. A dedicated prototype low-pressure H_2 DI injection system from Bosch was integrated.

CFD simulation and high-speed Schlieren measurements have been used to optimize the in-cylinder charge motion and mixture formation for H_2 combustion. A high tumble intake port has been selected to improve air fuel mixing. Injector pocket design and injector recess have been optimized to provide a focused H_2 jet that supports tumble and mixture homogenization by utilizing interaction with cylinder wall and piston surface. Additionally, a focused jet avoided early contact of hydrogen with hot surfaces of spark plug and exhaust valves, mitigating the risk for pre-ignition.

After a first calibration loop, engine out NO_x emission below 0.2 g/kWh and a brake thermal efficiency of up to 40 % (without e-SC electric energy demand) have been achieved in the main hybrid operating area of the engine map. Furthermore, the performance targets for maximum engine torque and rated power could be achieved and an excellent combustion stability in complete engine map was confirmed.

The next steps will include an optimization of the engine crank train including a variation of the compression ratio and an increase of peak pressure capability to improve the engine's high load performance and for further mitigation of pre-ignition risk amongst others. In addition, a full implementation of the active crankcase ventilation will be done. Finally, the transient operation in a hybrid powertrain as well as a potential design and calibration of a lean burn hydrogen exhaust aftertreatment system will be evaluated.

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