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System Optimization in a State-of-the-Art V8 6.6l Hydrogen Engine

Systemoptimierung in einem hochmodernen V8 6,6l Wasserstoffmotor

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Abstract

The simultaneous reduction of GHG gases and criteria pollutants is the main target of current engine development. Hydrogen (H_2) fueling is one of the best options available in both regards, as it does not emit tailpipe CO_2 as well as significantly abates all criteria pollutants already at engine-out level. However, optimizing H_2 combustion to exploit the fuel properties at best is a challenging task due to the very particular features of the H_2 once employed in an engine.

In fact, H_2 combustion promotes a clean, fast and complete combustion process. The usage of lean-boosted and EGR-diluted combustion strategies can significantly contribute to raise the brake thermal efficiency and lower the NO_x emission to extremely low levels.

To this end, the Authors developed a novel combustion system featuring optimized injection and ignition systems and their corresponding layout, as well as a dedicated piston bowl for an engine derived from Diesel operating as H_2 monofuel, in SI mode. In this paper, they discuss the effects on combustion system performances of the main parameters, investigated by means of CFD analysis and experimental DOE techniques.

Kurzfassung

Die gleichzeitige Reduzierung von THG-Gasen und Schadstoffen ist das Hauptziel der aktuellen Motorenentwicklung. Die Betankung mit Wasserstoff (H_2) ist in beiden Hinsichten eine der besten verfügbaren Optionen, da sie kein Auspuff-CO_2 emittiert und alle kriterialen Schadstoffe bereits auf der Stufe ohne Motor erheblich reduziert. Die Optimierung der H_2-Verbrennung, um die Kraftstoffeigenschaften optimal auszunutzen, ist jedoch aufgrund der sehr besonderen Eigenschaften des einmal in einem Motor verwendeten H_2 eine herausfordernde Aufgabe.

Tatsächlich fördert die H_2-Verbrennung einen sauberen, schnellen und vollständigen Verbrennungsprozess. Die Verwendung von Mager-aufgeladenen und EGR-verdünnten Verbrennungsstrategien kann erheblich dazu beitragen, den thermischen Wirkungsgrad der Bremse zu erhöhen und die NO_x-Emission auf extrem niedrige Niveaus zu senken.

Zu diesem Zweck entwickelten die Autoren ein neuartiges Verbrennungssystem mit optimierten Einspritz- und Zündsystemen und deren entsprechendem Layout sowie einer speziellen Kolbenmulde für einen von Diesel abgeleiteten Motor, der als H_2-Monokraftstoff im SI-Modus betrieben wird. In diesem Papier diskutieren sie die Auswirkungen der Hauptparameter auf die Leistung des Verbrennungssystems, die mittels CFD-Analyse und experimenteller DOE-Techniken untersucht wurden.

1. Introduction

In recent years, the research and development on Hydrogen-fueled Internal Combustion Engines (H_2-ICE) has picked-up in a significant way worldwide [1-8], thanks to the potential of hydrogen for sustainable and clean combustion, as well as to the proven qualities of ICE technology in terms of performance, efficiency, reliability, and favorable Total Cost of Ownership that are expected to be inherited by H_2-ICE (**Figure1**).

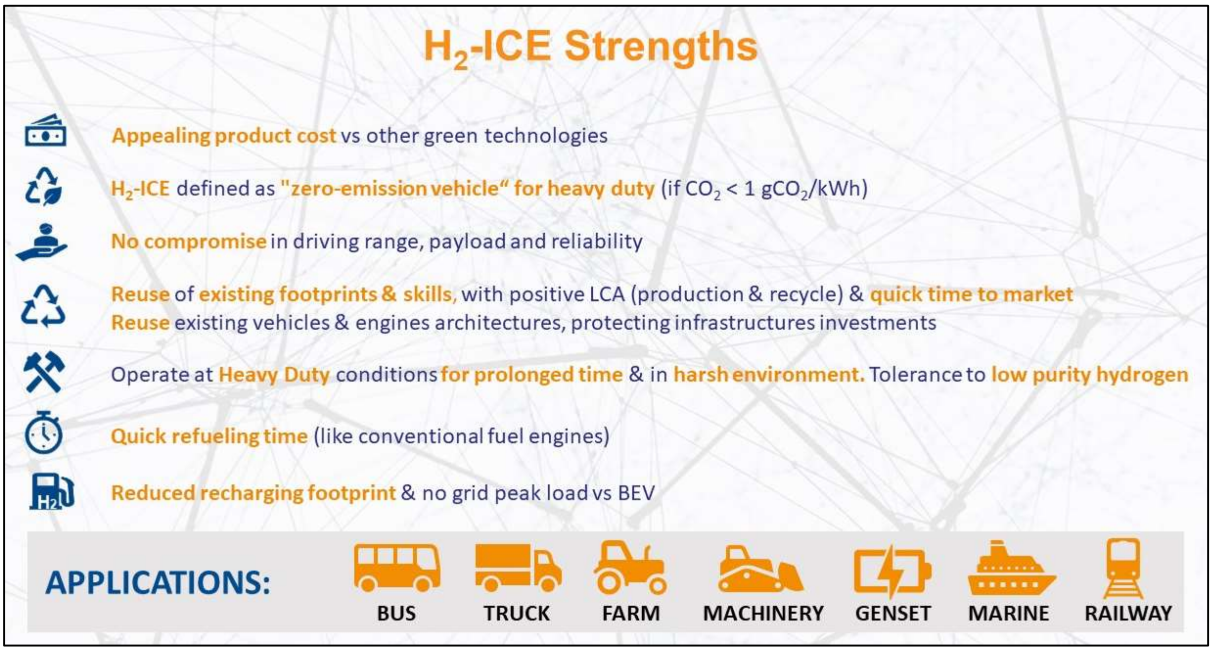


Figure 1 – H_2-ICE strengths.

Within this framework, PUNCH Group has invested considerable resources in developing a portfolio of technologies in the field of dedicated monofuel and dual-fuel hydrogen engines, with the scope to provide tailored products to its Customers for each specific application. The fast progress of PUNCH H_2-ICE development has been documented by several papers edited in prestigious congressed within the last 2 years [9-12], and encompassing analysis, single-cylinder engine testing and, more recently, its first multi-cylinder engine (**Figure 2**).

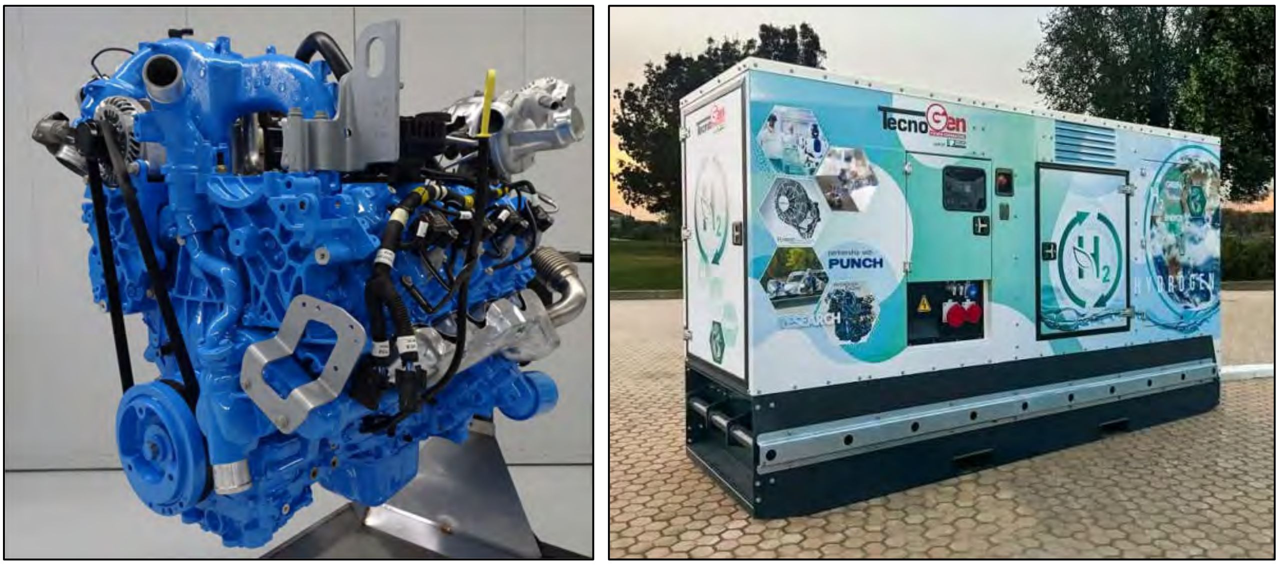


Figure 2 – H_2 V8 6.6ℓ PFI for GenSet application (LEFT) and whole GenSet by PUNCH (RIGHT).

In fact, Figure 2 (left) illustrates the first commercial application resulting from this effort, i.e. the V8 6.6ℓ Hydrogen Port Fuel Injection (H_2-PFI) engine for GenSet. In addition, Figure 2 (right) portrays the GenSet developed by PUNCH in close cooperation with its partner Tecnogen, recently presented at the Munich Bauma Show (Oct 24-30th, 2022).
This first application witnesses that H_2-ICE finds its most appropriate target where a relatively high specific power is required, in combination with high reliability, easy serviceability and quick refueling time. To meet all these targets, the choice of donor engine naturally fell back on a Diesel engine and specifically on the heavy-duty version of General Motors V8 6.6ℓ Duramax engine. This engine, already well-known at PUNCH (which – as a matter of fact – was formerly part of GM Global Propulsion Systems), develops 260kW as HD Diesel. Overall dimensions are just short of 1 meter in the three directions, with a weight of 450kg according to DIN70020. The ambitious target of PUNCH has been to develop an H_2-PFI V8 6.6ℓ engine variant with the similar rated power as the Diesel donor engine.

2. Engine application and systems architecture

2.1 H_2-PFI Engine Architecture

As mentioned in the Introduction, some of the strengths of H_2-ICE technology are the reuse of existing footprints and skills in the whole ICE supply chain, quick time to market and the relatively easy application to standard vehicle architectures. In this specific case, in fact 20% of the engine components have been re-designed or modified vs the Diesel version.

Engine Architecture	V8 – Firing order: 1-2-7-8-4-5-6-3
Displacement	6600 cm³
Bore / Stroke	103.0 mm / 99.0 mm
Rated Power	121 kW @ 1800 rpm @ 0.8 g/kWh NOx NTE 250 kW @ 3000 rpm @ 0.8 g/kWh NOx NTE
Rated Torque	850 Nm @ 2000-2500 RPM
Cylinder Block	Cast Iron with Lower Crankcase Extension
Cylinder Head	Aluminum, 4 valves per cylinder, Cam-in-Block
Piston	Aluminum
Timing System	Geared Cam-in-Block
Cooling	Mechanical water pump with 2 wax thermostats
Fuel Injection	Sequentially-phased Port-Fuel Injection, 2 injectors per cylinder
Ignition System	Ultra-cold spark plugs, centrally-mounted, with single coils
Charging	Single-stage VGT
EGR	High – Pressure EGR (cooled)
Charge Cooling	Low-temperature Charge Air Cooler
Engine Weight	450kg (DIN70020)

Table 1 – Main engine specifications of H_2 V8 6.6ℓ PFI.

The H_2-ICE key changes from Diesel baseline engine are shortly described below:

- **Injection system & intake manifold**: for the PFI solution the intake manifold has been designed to accommodate two H_2 injectors per cylinder and the related pipes to inject as close as possible to the cylinder intake ports. H_2 rail is integrated in the intake manifold as well. Several loops have been done, analysis and testing, to select the optimal solution in terms of mixing. This highly integrated design has been made possible using additive manufacturing technology.
- **Ignition**: spark plugs and coils are located where originally the Diesel injectors were. The cylinder head has been modified for optimal spark protrusion in the combustion chamber as well as for most effective spark cooling. The ignition system has been optimized to reach the required combustion stability and to avoid any combustion anomaly. Spark plugs and coils screenings have been completed to identify the final hardware configuration.
- **Air delivery and charging**: the turbocharger and its interface have been selected and designed to achieve the required Lambda values for lean combustion to control NO_x emission and to have the best BTE. Intake and exhaust valves phasing have been part of the optimization. The turbo has been repositioned to free-up space for H_2 injectors, so a specific turbo-bracket has been required
- **Piston and piston rings**: compression ratio, combustion bowl and piston design are key elements to achieve the required power level without combustion anomalies. Piston rings has been optimized to reduce the blow by for the same reason.
- **Engine oil and lubrication**: a wide test campaign has been conducted to select the proper oil specification to prevent combustion anomalies and to achieve the required compatibility with H_2 in terms of resistance to water. The blow-by system has been refined to avoid excessive oil in the combustion chamber
- **Valves and valve seats**: new materials were selected for dry combustion and H_2 compatibility
- **Material compatibility** has been verified for all components that get in contact with H_2
- **Specific EMS, composed by PEP (PUNCH Electronic Platform) and software**: proprietary Control Unit (HW) and Software has been developed to manage H_2 EMS architecture (H_2 sensors, Knock, sensors, etc.) and combustion.

Figure 3 shows the H_2 V8 6.6ℓ PFI engine for GenSet application from the two isometric CAD viewpoints, highlighting (in blue) the specific components modified or added for the hydrogen conversion, as described in the list above list.

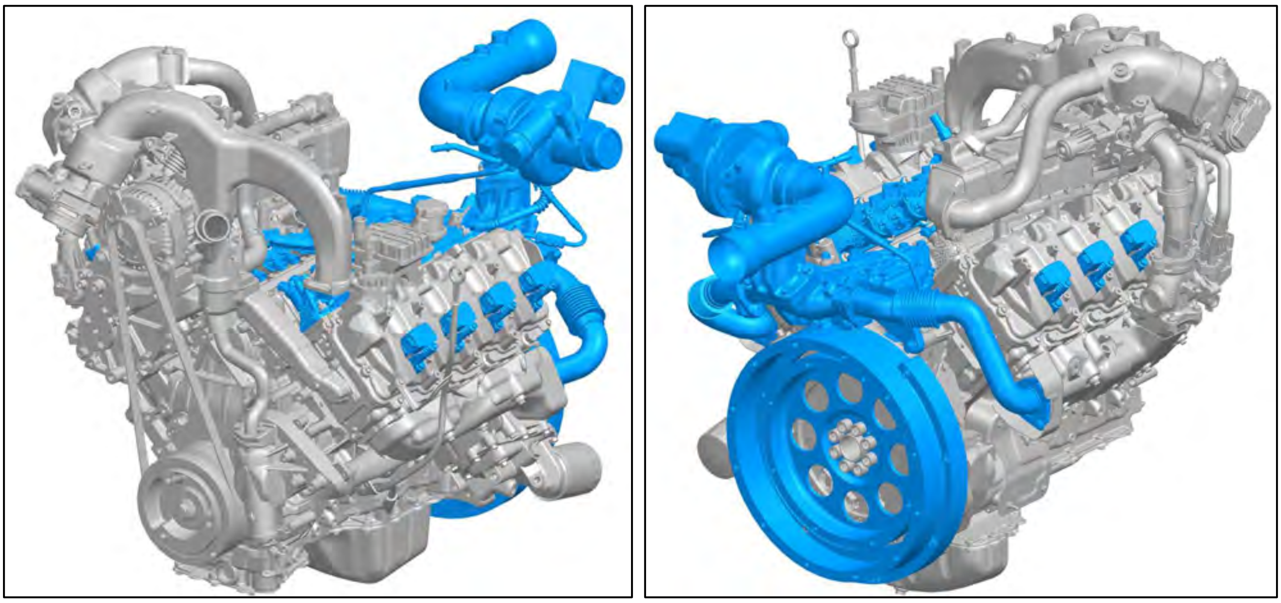
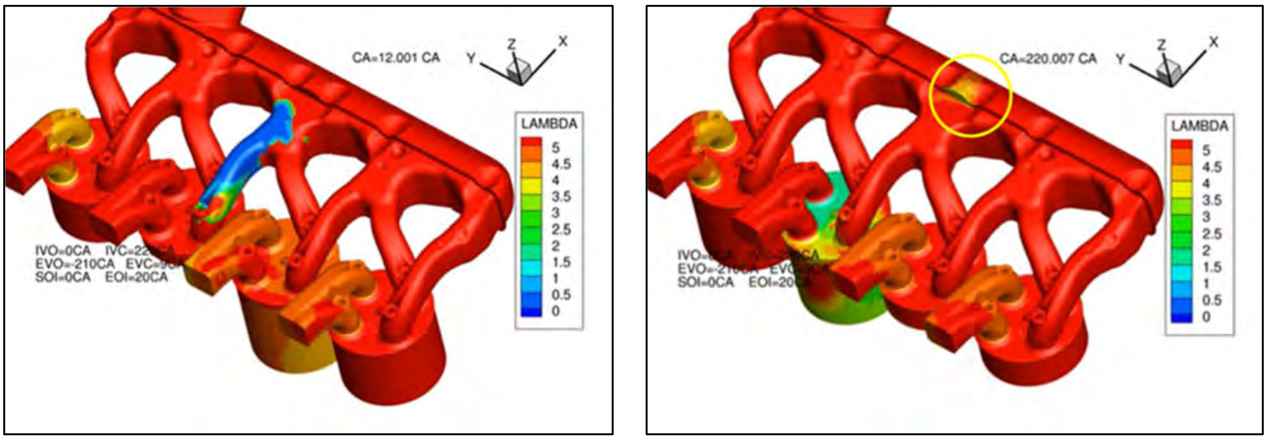


Figure 3 – H_2 V8 6.6ℓ PFI for GenSet application: isometric CAD views highlighting (in blue) specific components for Hydrogen conversion

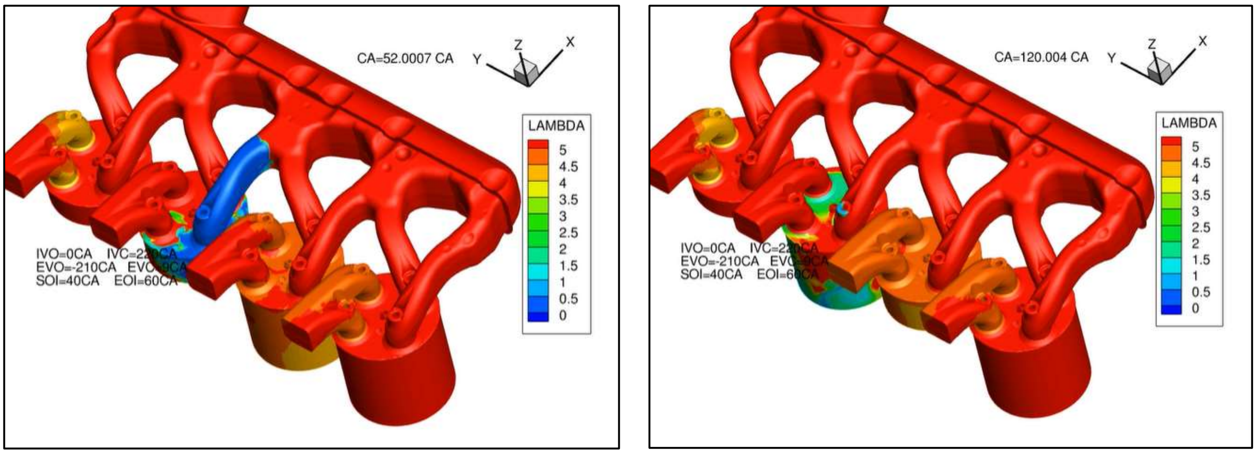
2.2 H_2-PFI combustion & charging system analysis and design

A CFD 3D analysis (performed within Converge environment) has been done to optimally place the Start of Injection (SOI) for the PFI fuel system with respect to Intake Valve Opening (IVO), a critical feature to make H_2 PFI safe from backfire standpoint as well as to minimize the volumetric efficiency penalty. Particularly, the SOI should be placed as early as possible to ensure enough timing to inject the desired quantity before IVC. At the same time, cylinder-to-cylinder fuel imbalance may appear if the SOI is over anticipated.
Thus, starting with SOI placed @ IVO, different simulations have been run, by progressively reducing the SOI advance. For simplicity, the analyses have been done by injecting fuel in one port only and by monitoring the actual quantity of fuel that is ingested in the cylinder.
As shown in **Figure 4**, when SOI coincides with IVO, the fuel tends first to flow backward due to the lack of suction from the cylinder until piston descent creates enough depression: therefore, some expanded hydrogen escapes from the intake port where it was injected and remains trapped and diluted in the manifold, even once the intake valves are already closed. Instead, if SOI is placed well after IVO and therefore fuel injection happens in an established suction flow, the fuel entirely and readily enters the respective cylinder (**Figure 5**). It is interesting to notice that the same trend was correctly predicted even by a simpler CFD 1D analysis (GT-Suite) featuring emptying-filling cylinder models, as shown in the **Table 2**. This suggests that the dynamics of the intake system should be studied at least by 1D-CFD to correctly design the H_2 PFI and piping system.



(A) (B)

Figure 4 – Lambda distribution (A) 12 CA deg after SOI, coinciding with IVO, and (B) at IVC.



(A) (B)

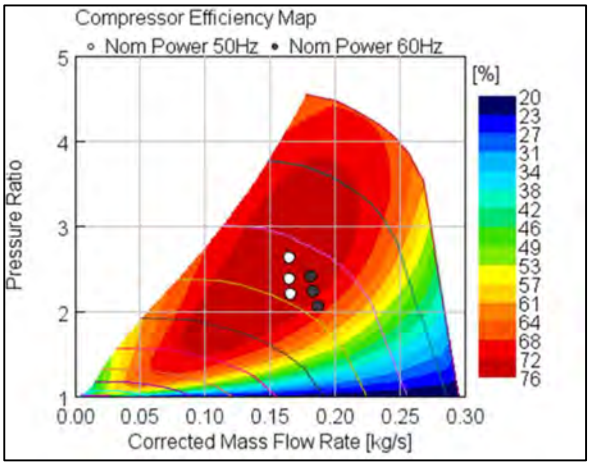
Figure 5 – Lambda distribution (A) 12 CA deg after SOI, with SOI placed 40 CA deg after IVO, and (B) at IVC.

Table 2 – Percent in-cylinder fuel mass of total injected mass for different SOI values.

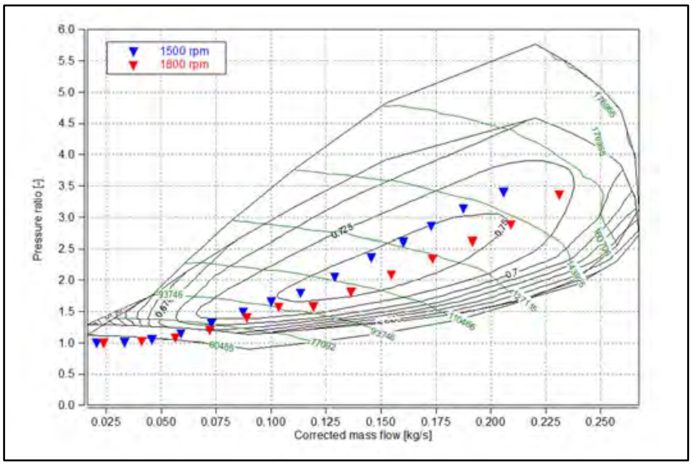
SOI	Percent in-cylinder fuel mass	
	CFD 3D (Converge)	CFD 1D (GT-Suite)
[CA deg] after IVO	[%]	[%]
0	73	87
40	93	98

As a next step, an analytical turbo-compressor matching activity has been performed for the H_2 GenSet application, due to the change in the fuel with respect to the donor Diesel application as well as to the very specific operating area of the GenSet with respect to a full-range application for either on-road and off-road. In fact, the GenSet operates at low speed in a narrow 150 rpm range centered around 1500 rpm and 1800 rpm, depending on 50Hz and 60Hz frequency, respectively. At these speeds, the turbocharger of the original Diesel engine is not able to provide enough boost to guarantee the required charge dilution, up to Lambda value of ~2.3. Thus, a smaller turbocharger has been selected, off-the-shelf, having a compressor wheel of ~63mm and a turbine wheel of ~50mm.

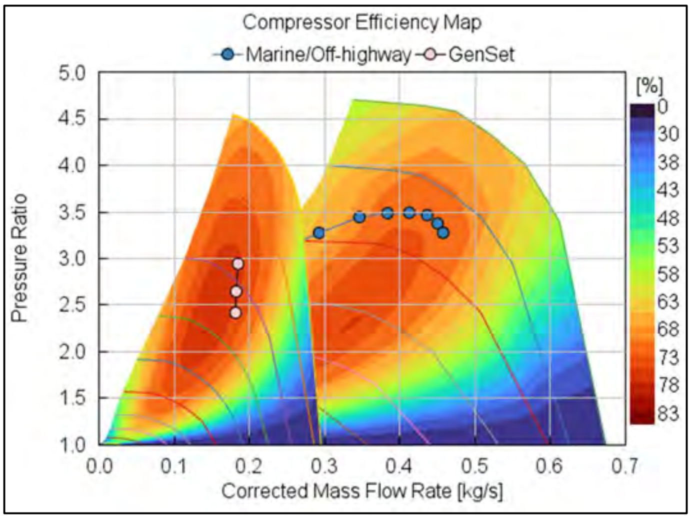
The selected turbocharger has its maximum efficiency close to the nominal power working points, as depicted in **Figure 6 (A)** for minimum (-10%), maximum (+10%) and nominal speed, at 50Hz and 60Hz. This ideal matching together with ball bearings technology for rotating parts guarantees low pumping and enables fast transient response as required by a GenSet. Finally, the favorable position of most engine operating points on compressor efficiency map ensures proper boost margin (and transient build-up) to meet the target lambda value also at altitude conditions **Figure 6 (B)**.
For the marine and off-highway engine applications, a maximum performance analysis was conducted as well, this time up to 3000 rpm, and – based on it – a slightly upgraded matching with an off the shelf Diesel T/C was selected (in particular, with better low-end torque capability).
Figure 6 (C) shows the overlapped comparison of the final selected T/C compressor maps for V8 6.6ℓ GenSet and Marine/Off-Highway, to highlight the quite different corrected mass flow ranges required by the two applications, that evidently cannot share the same T/C.



(A)



(B)



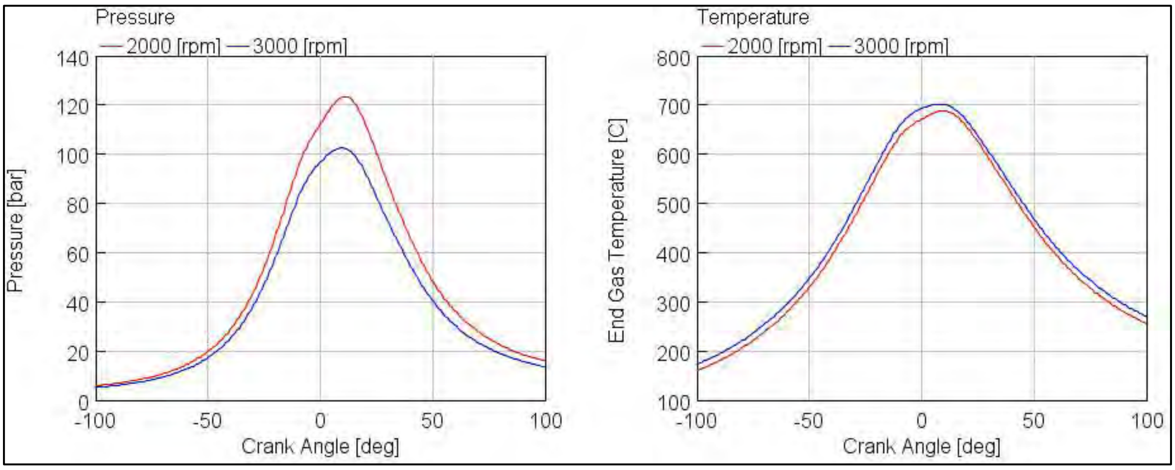
(C)

Figure 6 – GenSet simulated compressor efficiency at nominal power, at 50Hz and 60Hz (A). GenSet experimental operating points across the entire output range (B). Marine/Off-highway vs GenSet compressor maps with engine operating points (C).

Besides turbocharger-specific limits (temperature, pressure, speed, surge & choke lines), also engine knocking has been considered as a limit for the maximum boost. The knock appearance has been evaluated by Livengood and Wu approach that says that the auto-ignition of the end gas occurs when the $\int 1/\tau\, dt = 1$, where τ is the ignition delay, while the integral is called Knock Index. To evaluate the Knock Index, the ignition delay τ has been computed by analysis (Converge) for different values of pressure, initial temperature, EGR ratio and Lambda [13].

From the analysis, it resulted that the boost is limited by knocking at a lower value for 3000 rpm than for 2000 rpm, contrarily to what happens with other fuels, for which the higher is the speed, the less is the possibility for the end gas to auto-ignite. This analysis result, later confirmed also by tests, revealed that knocking is mostly influenced by temperature, in the Lambda range of usage of hydrogen. In fact, at 3000 rpm, in-cylinder temperature is slightly higher since the beginning of the compression phase, due to a higher gas-wall heat exchange at air inlet.

In **Figure 7**, indicating results at 2000 rpm and 3000 rpm at max performance, with compression ratio 12 and a safety factor of 0.75 for the knock index, are reported.



Engine speed	rpm	2000	3000
Knock Index	-	0.75	0.75
Boost Pressure	bar	3.6	2.9
Apparent Lambda	-	2.20	2.36
Burned Mass Percent (Residuals)	%	7	5

Figure 7 – Max performance analysis results at 2000 [rpm] and 3000 [rpm], with compression ratio of 12.0, limited by knocking (Limit Knock Index = 0.75).

3. Engine sub-systems design

3.1 Piston group and reciprocating

The original Diesel engine layout has represented a favourable basis for the H_2 project. Indeed, a Diesel piston has a quite generous compression height (as measured from piston top to piston pin bore axis, usually is in a range of 50-55% of the cylinder bore diameter) which is originally designed to withstand the high loads generated by Diesel combustion peak firing pressure. Within this space, the room for different designs of combustion bowl shapes has been found, still allowing to keep sufficient wall thickness from oil cooling gallery and ring grooves for an overall reliable piston structure (**Figure 8**).

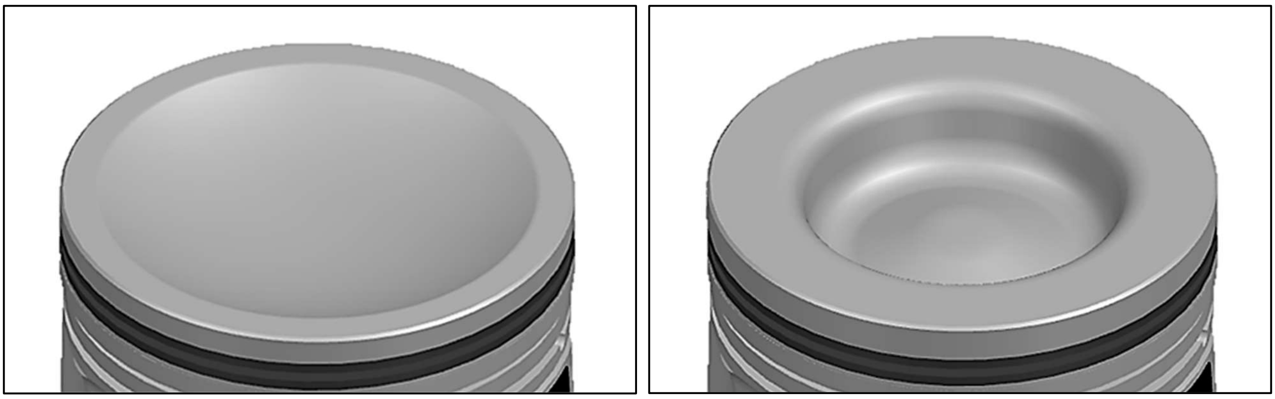


Figure 8 – Baseline (LEFT) and optimized (RIGHT) piston bowls.

In details, a quite flat piston top with just a slight semi-ellipsoidal bowl has been designed in a first step, aiming to realize a remarkable reduction of the engine compression ratio by means of increasing the squish height between the head deck and piston top. This first design has shown some limits due to charge oil contamination stemming from piston rings, in turn triggering combustion anomalies such as backfire and knocking. Moreover, to limit as much as possible the knocking tendency at high speed, the need of a further reduction of the Compression Ratio (CR) and improved in-cylinder charge homogeneity has been brought to attention by testing this first concept.

Therefore, a second bowl shape has been designed: the overall piston height has been increased leading to a peripherical reduction of the squish height and consequent squish strength. Moreover, piston central portion has been machined in depth thus finally achieving a "cup" shaped bowl with higher volume and lower compression ratio than the first version. This new shape has gathered, in addition to the benefit of the lower CR, an appreciable protection of the combustion from the residual oil on the cylinder bore surface [14].

In parallel to the work above on the bowl/piston top, a special attention has been paid on the piston lands, ring grooves and rings. The challenging task has been to limit as much as possible H_2 diffusivity to the crankcase by reducing the functional gaps (relying also on lower temperatures/lower thermal expansion in comparison with the original Diesel version). On the other hand, oil scraping capability has been optimized too, by working on the oil control ring design, always with the target to limit at a minimum the charge contamination by oil.

3.2 H_2 Injection System

A completely new injection system was designed, to replace the baseline diesel injection one. As a PFI architecture was selected, CNG-like, side-fed PFI injectors were used, properly modified to make them suitable for usage with Hydrogen: main modifications entail the internal materials, to avoid issues with elastomers and metals compatibility, and the orifice size, since hydrogen has higher specific volume compared to CNG.

Fuel rail and injector seats were integrated in the intake manifolds, leveraging the potential of additive manufacturing: aluminum was chosen as material.

As it will be described in more details in the following of this paper, mixing is paramount for performance and combustion stability of a PFI hydrogen engine: for this reason, the hydrogen channels driving the gas towards the combustion chamber have been designed through multiple loops of analysis and testing to achieve optimal volumetric efficiency and mixing performance. As a matter of fact, pipes have been designed to channel hydrogen from the tip of the injectors as close as possible to the intake valves, despite the design of the baseline intake ports is far from ideal for a PFI system (**Figure 9**).

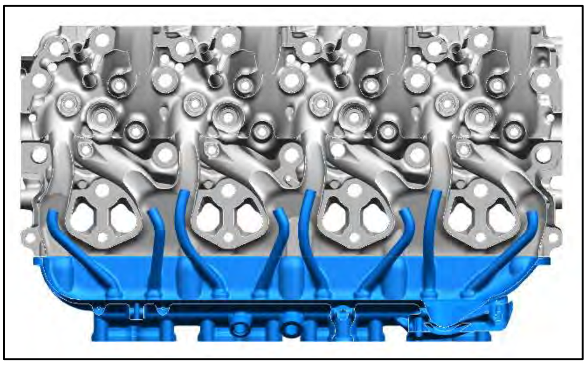
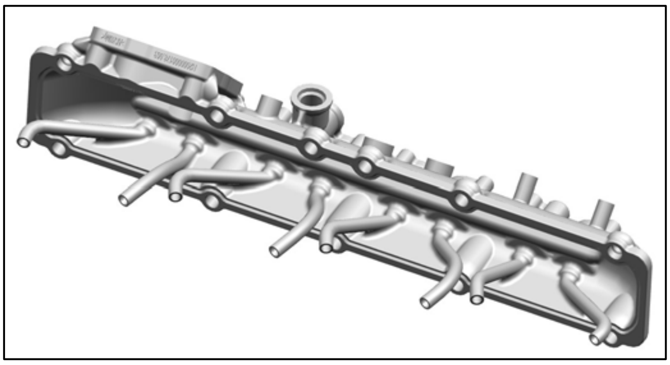



Figure 9 – H_2 V8 6.6ℓ PFI engine: cylinder head transparency with details of injection pipes (LEFT) and internal view of the intake manifold (RIGHT).

In PFI systems, it is important to guarantee that the H_2 fuel is well distributed inside the inlet manifold. In fact, a non-uniform mixing can cause several combustion anomalies, including misfire, knocking, preignition and backfire. This is due to the fact the H_2 can burn effectively only within a specific Lambda window, that – while wide with respect to other fuels – is anyway limited on both the slightly lean and the very lean sides. Too “high” Lambda values are causing misfire due to lack of reactivity, and, on the contrary, too “low” Lambda values are causing knocking (or even preignition) due to excessive reactivity of the mixture. Backfire indeed is also possible because the H_2 ingested inside the cylinder gets in contact with hot points; in this case H_2 ignites and burns before the inlet valve is closed, causing a sudden increase in inlet manifold pressure as well as heavy mass reverse-flow in all the inlet pipes back till the air filter. Noise and lack of torque are consequences of the above phenomenon, but air backflow means also fresh air reduction in following combustion cycles, causing pre-ignition and mega-knocking in several cylinders during next cycles (running at low Lambda), in a self-exciting phenomenon which is very intrusive to the engine smooth operation and even to its safety.

Good H_2 mixing is obtained paying attention to two different aspects:

1) H_2 distribution in the inlet manifold must be uniform as much as possible. Despite the complexity of designing individual ducts per each intake port, BMEP was found to be optimal with two ducts per cylinder.
2) Velocity of the air during the H_2 injection phase. As a matter of fact, a wrong SOI can cause high fuel unbalancing between the cylinders because a portion of H_2 is not able to enter completely into the cylinder; this H_2 portion will remain in the inlet manifold and will be usually sucked by the cylinder at the end of the inlet manifold. In the example reported below, an increase of pressure (between -280 and -200 deg) due to abnormal combustion is visible. Different angles of backfire are in correlation with H_2 SOI: when H_2 is injected, local Lambda min is reached inside the combustion chamber. Lower the local Lambda, higher the H_2 reactivity (**Figure 10**).

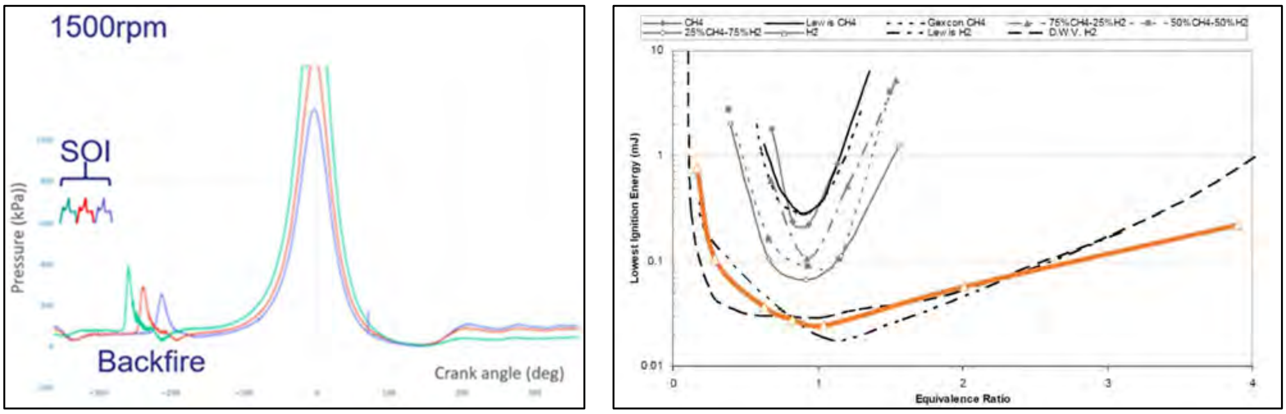


Figure 10 – Recorded in-cylinder pressure traces of backfire events for different SOI (LEFT) and H_2 backfire 'risk curve' (orange) as function of Equivalence Ratio - edited from [15] (RIGHT).

Therefore, it is important to guarantee that H_2 injection is well tuned with air velocity caused by valve opening (as highlighted by **Figure 11**, where SOI and EOI effect is measured by the increase of cylinder-to-cylinder IMEP variation).

As it is possible to see, too early SOI cannot be used (H_2 is not sucked by the fresh air motion), and too late SOI is causing a too late EOI (and H_2 has not enough energy to go completely inside the cylinder). It is interesting to note that EOI is not affected by rail pressure, but SOI, indeed, can be postponed with rail pressure increase. This is because the higher is the fuel rail pressure, the lower is the required angle to inject the desired amount of fuel.

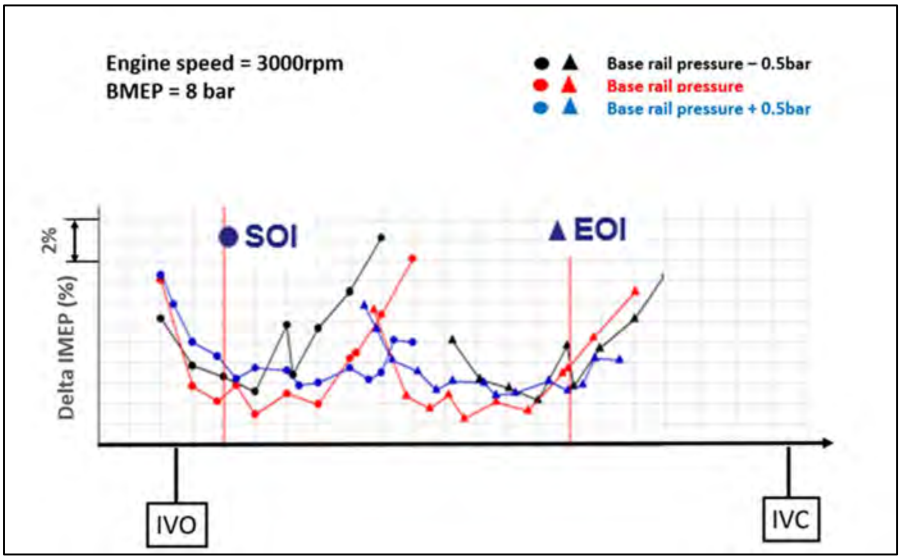


Figure 11 – Cycle-to-cycle IMEP variation depending on SOI, EOI and Rail Pressure, identifying the safe interval for EOI and SOI.

Rail pressure setting is another important parameter: it needs to be high enough to guarantee to reach the target power, but it must also be low enough to allow good idle management (opening of injector can't be too short, to guarantee good shot-to-shot stability).

For the selected injectors, the rail pressure should be in the range of 100kPa_rel and 800kPa_rel respectively at idle and in Full Load condition. Unfortunately, standard gas pressure regulator increases the rail pressure in the same way in which the inlet manifold pressure increases, and so is not able to respect the desired pressure (**Figure 12**).

Different solutions have been investigated to overcome the above-described limitation (pressure booster, double series spring stiffness), but the best choice has been eventually to use an active (electronic) rail pressure regulator, thanks to its operating flexibility.

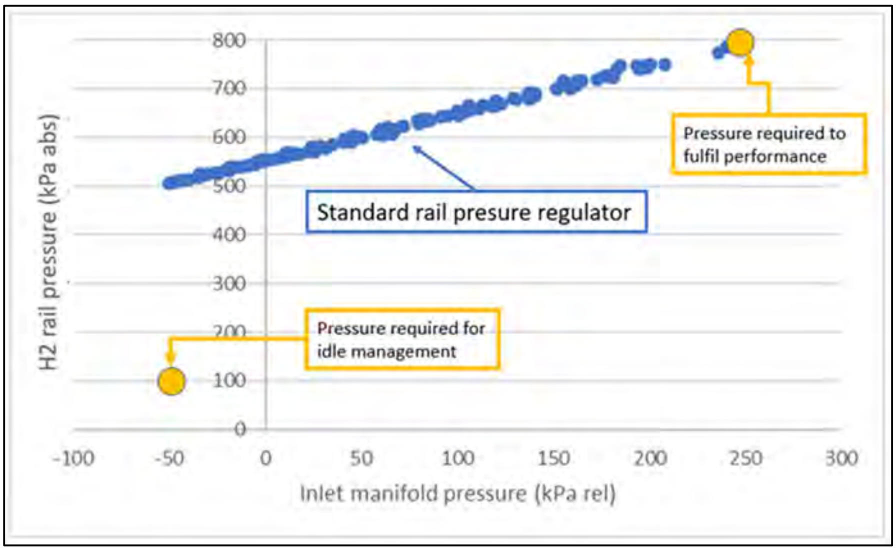


Figure 12 – H_2 rail pressure regulation achievable by mechanical regulator (blue) as function of engine inlet manifold pressure, compared to the range required by engine (yellow points).

Let's conclude this paragraph by saying that hydrogen feed from regulator is connected to one rail, then a standard stainless-steel pipe (safe for embrittlement) connects the two rails together to distribute gas to both engine banks. Hydrogen pressure and temperature are measured to properly compensate for all engine operating conditions and finely correct gas injection accordingly.

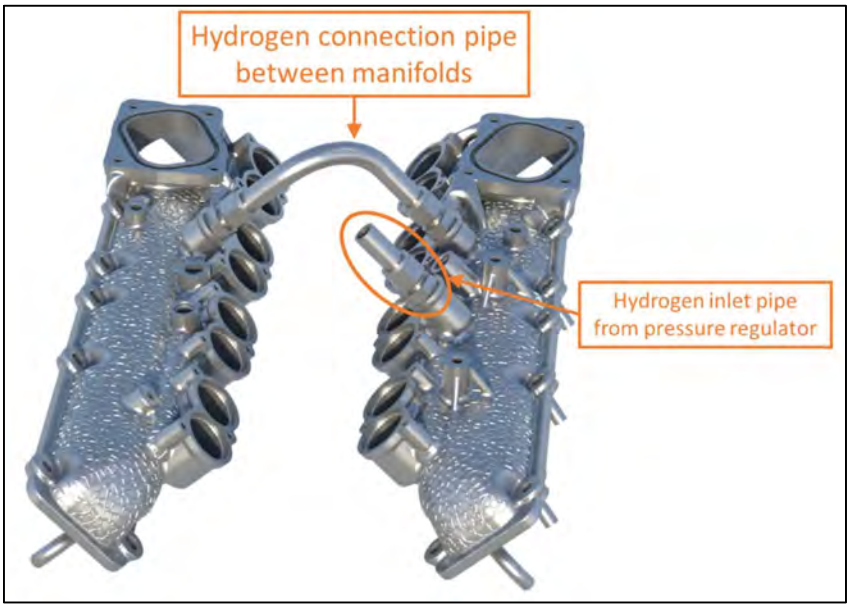


Figure 13 – H_2 feed from pressure regulator and connection between rails on the two banks.

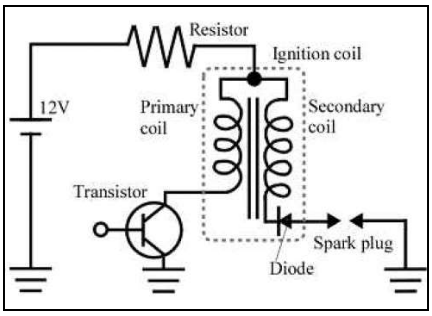
3.3 H_2 Spark Ignition System

The technology of ignition coil and spark plug used on the V8 PFI hydrogen engine is conceptually derived from standard spark-ignited gasoline / LPG / CNG applications, but

with some specific and important adaptations necessary to properly manage the ignition of the lean-burn H_2-ICE, while avoiding abnormal combustion events.
The adopted spark plug (**Figure 14**) features a central electrode, with a specific air gap, whose tip is aligned with the down face of the body used as ground electrode (a design so called "semi surface"); in addition, the spark plug internal body/insulator dimensions have been optimized for heat dissipation, making this design "very cold" (in the 'spark plug world jargon'). The positioning of the spark plug into the combustion chamber has been implemented to obtain the maximum efficiency about two main factors: the release of minimum energy necessary to ignite the lean & diluted hydrogen/air mixtures, and the ability of the spark plug to maintain its electrodes and its body as cold as possible. The experience in this application demonstrates that both spark plugs position, and its core design feature, are necessary to eliminate hotspots into the combustion chamber and potential unintended pre/post-ignition [14].

Figure 14 – H_2 V8 6.6ℓ PFI 'ultra-cold' spark-plug.

The ignition coil has been calibrated and controlled by the PUNCH controller, to deliver the correct energy, needed to ignite the hydrogen in all engine conditions and to avoid misfire. The control of the spark has been optimized with a specific feature capable to release quickly the residual energy delivered by the coil which successfully removes the unintended spark and protects the engine from any abnormal combustion events, such as knocking, pre-ignition and backfire (**Figure 15**).



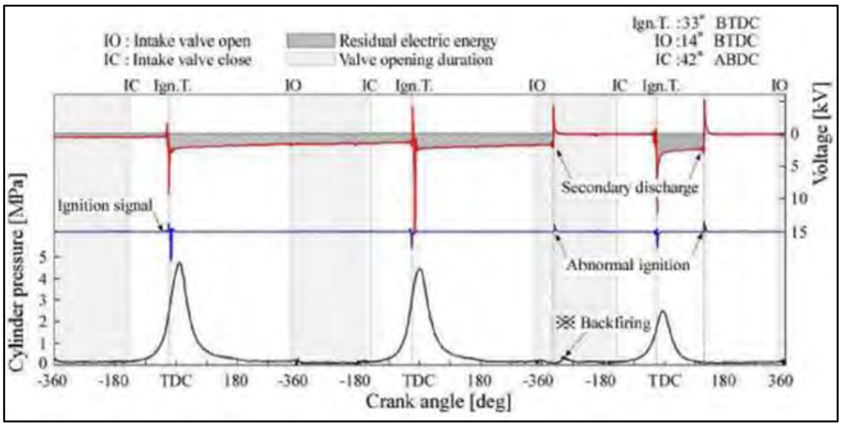


Figure 15 – System diagram of Transistor Coil Ignition (TCI) system (LEFT). Typical measurement result of residual electric energy in the TCI system of an H_2 engine (RIGHT). Reprinted from [16].

3.4 Charging and EGR System

To free-up the required space for the installation of the new injection system, the turbocharger assembly had to be relocated from the original position. The best trade-off between thermal and packaging requirements was found positioning the turbocharger in the rear of the engine, above the transmission interface.
The relocation drove a series of design changes, such as the need for new exhaust pipes and of a new support capable to operate as a manifold to collect gases from the two banks, as well as to support the cantilever mass of the turbocharger.
As previously discussed, a specific turbomatching was defined for each application, driving to the need of different turbochargers. The system was designed in a way that common exhaust pipes and manifold (green) can be used for both applications, installing the smaller turbo for Genset (red) with a dedicated adapter, and the bigger turbo for high output applications (blue) directly on the manifold.
The baseline Diesel EGR system (yellow) was maintained, as its performance are suitable for the hydrogen application.

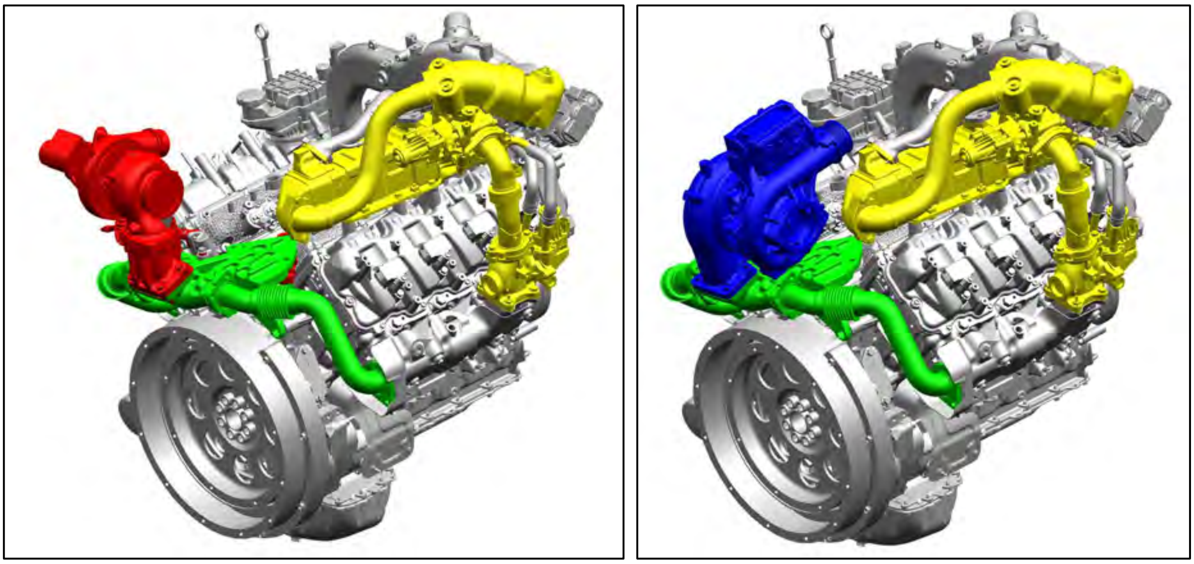


Figure 16 – H_2 V8 6.6ℓ PFI for GenSet (LEFT) and Marine/Off-Highway (RIGHT): charging (T/C is application-specific) and EGR circuits (EGR cooler and exhaust gas piping is common).

3.5 Lubrication and blow-by separation

Backfire is one of the main issues faced during the engine development and constitutes the most common combustion anomaly in H_2-ICE PFI engines as reported in the literature [14, 15]. Several parameters influence the backfire, and one of them is the engine oil formulation. Different oils have been tested, starting from Diesel based, moving forward a CNG formulation, and then selecting an oil specifically built for H_2. Mail learnings are:

- Avoid bases with glycerin (can causes emulsion thick in oil)
- Higher the flash point, higher the backfire robustness.
- Reduce Ca amount (used as an oxidation inhibitor: it neutralizes acids formed during combustion). Higher it is, higher is the backfire probability because it works like a catalyst for H_2 reaction. Different oils have been tested and moving from standard Ca content to a negligible amount, robustness against backfire has been increased. Mg quantity can be increased to recover the oxidation inhibition property.

Independently from the engine oil, particular attention must be put in the blow by system because oil droplets in the combustion chamber may affect the backfire robustness. Even if the standard passive blow by separator has a carry-over of just 1.5 g/h, this small amount was sufficient to cause 10% in power reduction. The same attention to reduce the amount of oil in contact with hydrogen must be devoted to piston cooling jet definition and piston rings selection: piston cooling jet flow must be reduced to minimum, and piston rings system must be designed prioritizing small oil consumption in comparison with all the other parameters (blow-by and friction) [17].

4. Engine performance, emissions, and efficiency

To reach the defined targets, it was necessary to overcome the main combustion anomalies such as backfire, pre-ignition and knock that may plague hydrogen combustion if non-optimized hardware & software are employed – as well-documented in the literature [14]. The backfire is one of the main issues faced during development activity and main reason for a conspicuous lack of power.
To prevent and improve the robustness to these phenomena it was necessary to eliminate, or minimize, of all those elements that could help to make the mixture air-hydrogen excessively reactive. These elements have been identified, as predominant on causing backfire and pre-ignition, in oil contamination and spark plug specifications (including its management). To avoid knock the key control parameters were compression ratio reduction, and heat rejection increase (both charge air and water temperature) to reduce charge temperature during the compression stroke.

4.1 Steady-state results

Thanks to the optimized design and engine operating parameters described in the previous section (spark plugs, ignition coils, CR, H_2 cylinder-to-cylinder distribution and in-cylinder homogeneity, SOI and SA tuning, blow by-system, piston rings, oil formulation), it was possible to reach a maximum power up to 277- kW (~42 kW/l) with the final configuration. This is an excellent value for a H_2 PFI engine based on most recent literature [1-8].

Phenomena	Parameters/Issues	Solutions
Backfire	H_2/oil backflow	Avoid H_2 & oil suction into combustion chamber
	Injection window	Enhanced H_2 trapping with optimal SOI - EOI
	Air/gas mixing	Avoid zones with low local Lambda
	Pre ignition	Avoid hot spots (spark plug, exhaust valves)
Knock & Pre-ignition	Intake air temperature	CAC efficiency and EGR/Lambda management
	Charge P & T	Reduce Compression ratio
	Hot spots	Avoid oil accumulation and ignition anomalies

Table 3 – Combustion anomalies: root causes & applied solutions.

Table 3 summarizes some of the most important root causes identified for combustion anomalies, and the consequent applied solutions in the engine development process.

As the Authors already reported in [12] regarding a smaller bore single-cylinder engine, also the larger bore multi-cylinder V8 engine exhibited a trade-off between NOx emission level at WOT and the corresponding performance (**Figure 17**). Peak torque (BMEP) can achieve high values across a wide range of speeds for H_2-PFI, competitive with the donor Diesel engine application, and safe from any combustion anomaly. The right balance of max performance and NOx at engine out depends on the aftertreatment strategy. The usage of well-known Ad-blue SCR technology for emission control will allow achieving high specific performances while being compliant with the most stringent emission standards. [1, 7, 8]

To safely achieve such high BMEP values (more than 18 bar in a wide rpm band for peak torque), charge dilution proven to be the most important parameter, as illustrated in **Figure 18**. If boost can be increased (within the T/C limits, and thanks to improved CAC efficiency / heat rejection) in a proportionated fashion, and thus keeping high Lambda (>1.8), the consequent power improvement is almost linear to the achieved boost level, with minimal penalty on NOx and even small BSFC gains.

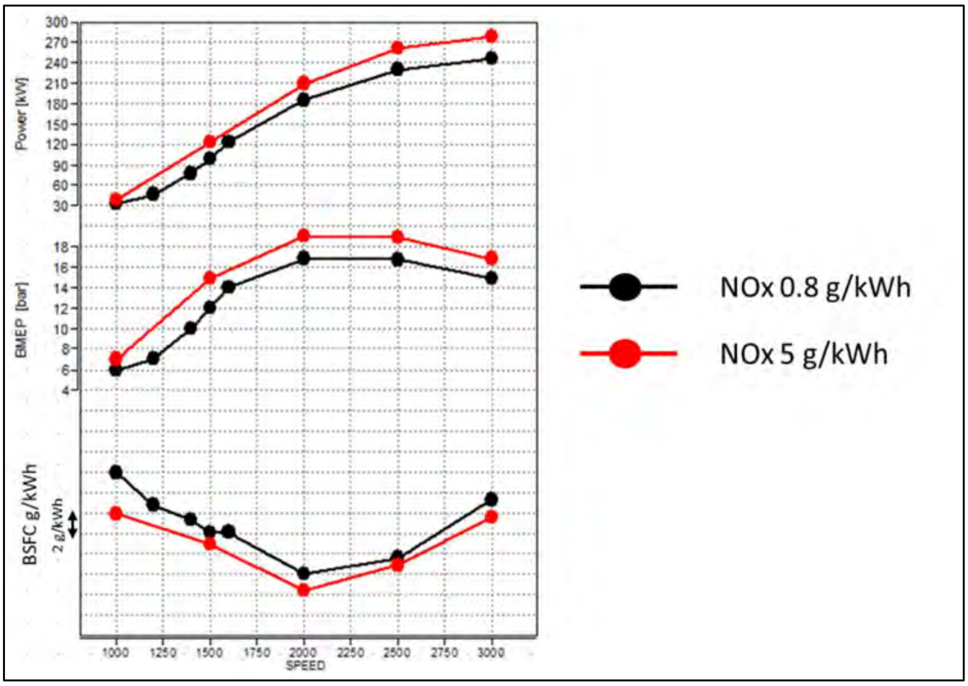


Figure 17 – H_2 V8 6.6ℓ PFI: WOT performance curve.

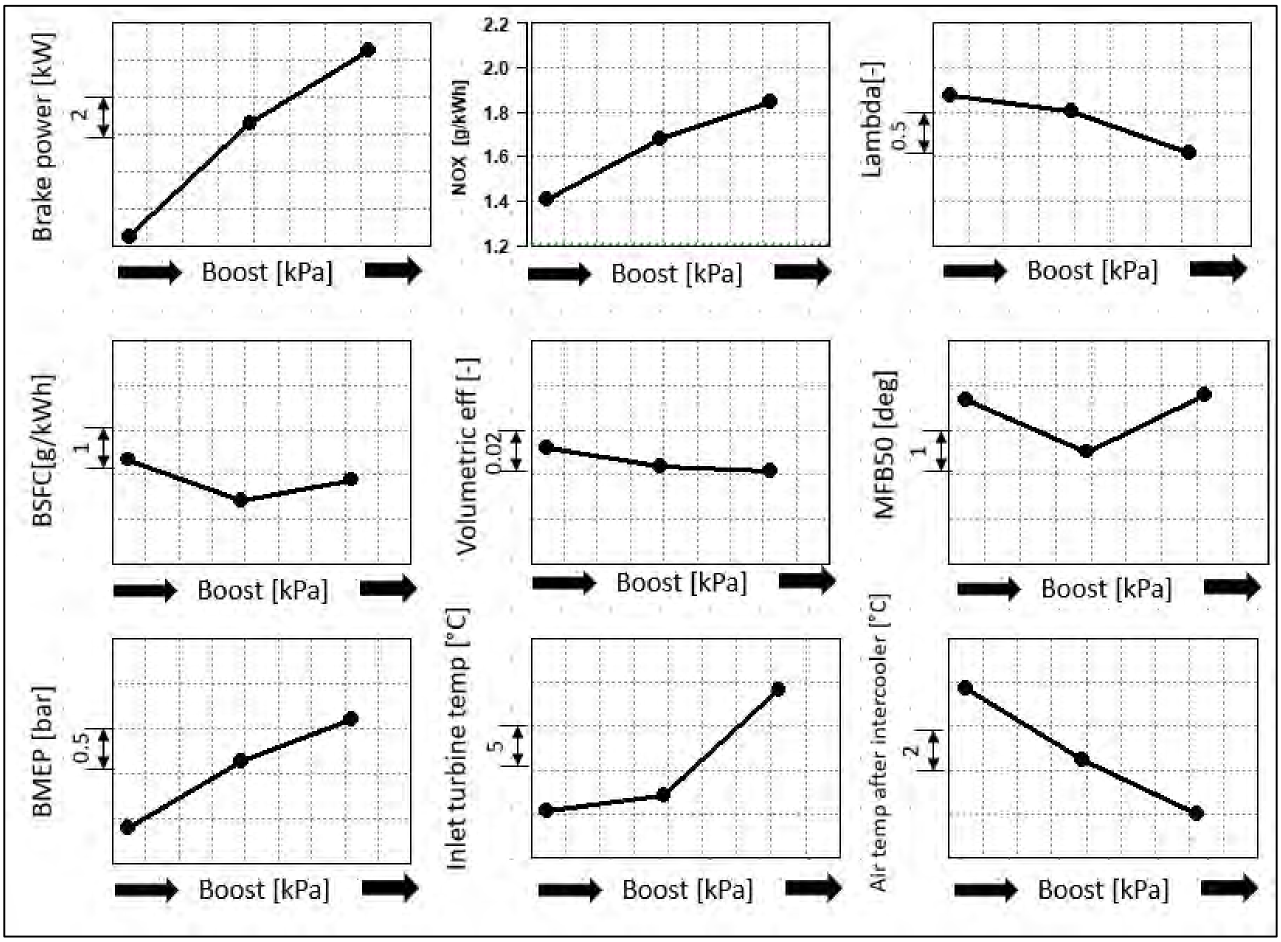


Figure 18 – H_2 V8 6.6ℓ PFI: effect of boost pressure on engine performance main operating parameters @ 90°C coolant.

However, as shown in **Figure 19**, if boost is pushed to the T/C limits and knock limitation starts to occur (especially for standard 90°C coolant temperature control), the power gain is completely offset by the progressive MFB50 retard (required to maintain knocking under acceptable MAPO values and to reduce NOx emission), with significant BSFC deterioration. In this case, the possibility to control the cylinder head coolant temperature to a lower value than the standard one (for example 70°C coolant as shown by the red curve in Figure 19) is a good enabler for efficiently achieving higher performance levels. It is also interesting to observe that the H_2-PFI engine overall efficiency optimization requires relatively low cylinder head temperature with respect to block temperature, to reduce the knock onset and avoid the thermal efficiency losses that stem from MFB50 retard. Therefore, lowering the cylinder head temperature gives more benefit on thermal efficiency, and more than offsets the mechanical efficiency losses due to oil viscosity increase in the rotating and reciprocating components of the engine top. This effect, while visible to a small extent also on the original Diesel engine regarding efficient NOx emissionization, is much more evident in the H_2-PFI engine.

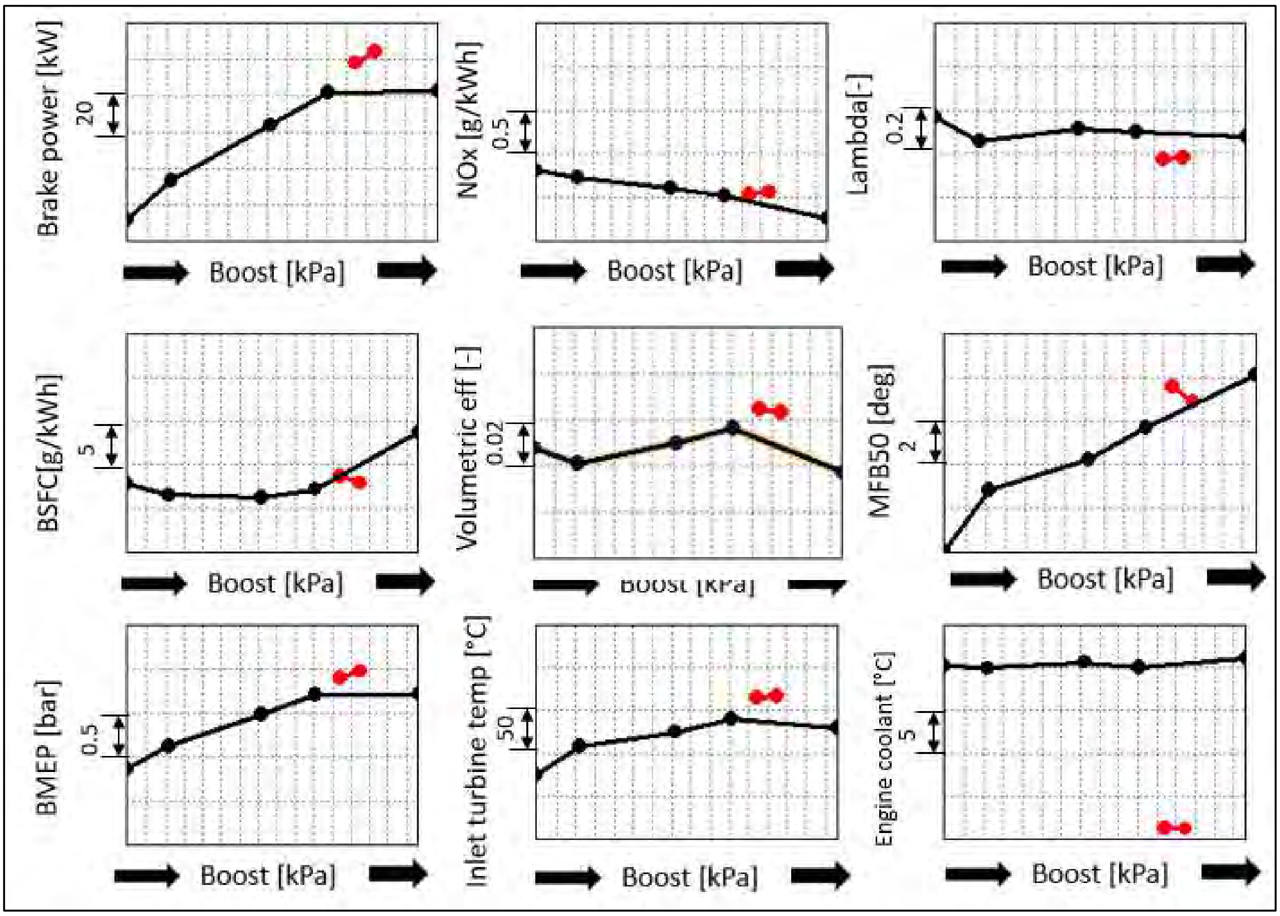


Figure 19 – H_2 V8 6.6ℓ PFI: effect of boost pressure on engine performance main operating parameters @ 90°C (black) and 70°C (red) coolant temperatures.

4.2 EGR effect

As discussed in several technical papers [7, 8, 12], EGR usage in hydrogen engines is still subjected to controversies, as not all studies agree on its advantages and suitability in contrast to a pure dilution through charge leaning. In the previous study on SCE already referenced [12], the Authors of the present paper found that in the smaller bore (~84 mm bore) engine investigated the effects of either using EGR or increased boost were virtually the same on achieving efficient and clean combustion at high loads. At the time, the Authors were also debating about the secondary effects of the actual T/C matching (and the consequent pumping) on the results, due to the SCE engine working with simulated boundary conditions.

For the medium-bore (~103 mm bore) of the present work, being an MCE engine with an actual T/C, the Authors found slightly different results depending on the operating point and application (GenSet vs Marine/Off-Highway). In fact, for the GenSet application which operates in a narrow & low speed range (1500-1800 rpm nominal), as reported in **Figure 20**, the application of EGR is deteriorating the engine efficiency for the same NOx level vs a pure fresh air charge dilution, indicating that the T/C operating point shifts to a not favorable operating region too close to the surge line.

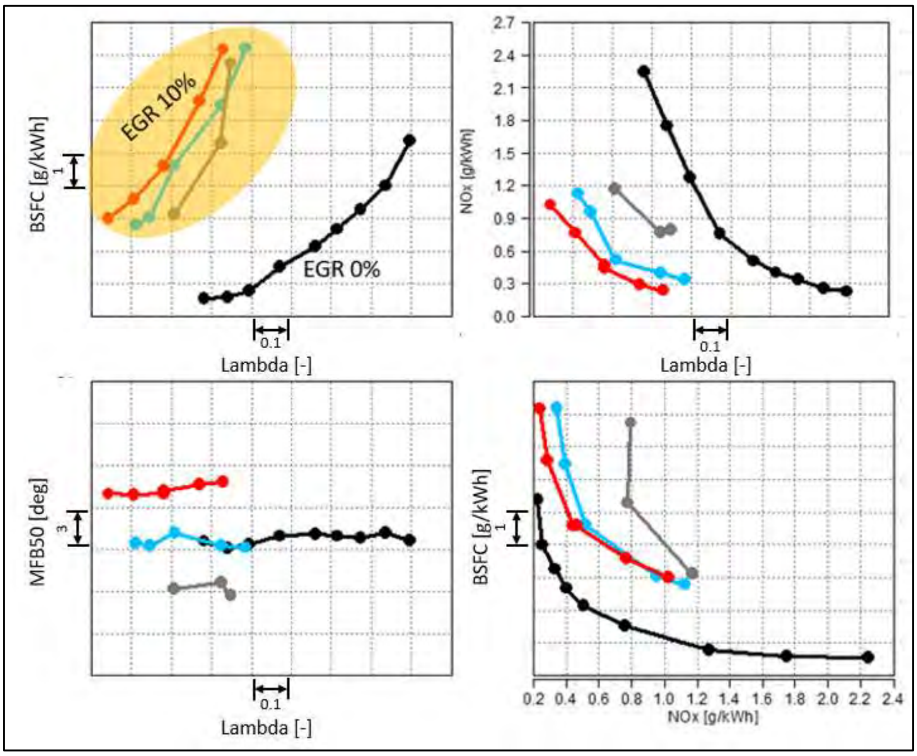


Figure 20 – H_2 V8 6.6ℓ PFI for GenSet application: effect of EGR on rated power @ 1800rpm.

On the other hand, at high speed typical of Marine/Off-Highway application, the EGR effect on the rated power @ 3000 rpm for NRMM standard was found beneficial. In fact, as **Figure 21** shows, in this speed range the usage of EGR provides benefits to the combustion through reduced charge reactivity and tendency to knock more than it affects T/C pumping. Therefore, in the high-speed range, the EGR usage leads to a favorable combination of higher power, lower NOx and better BSFC than the pure fresh air charge dilution.

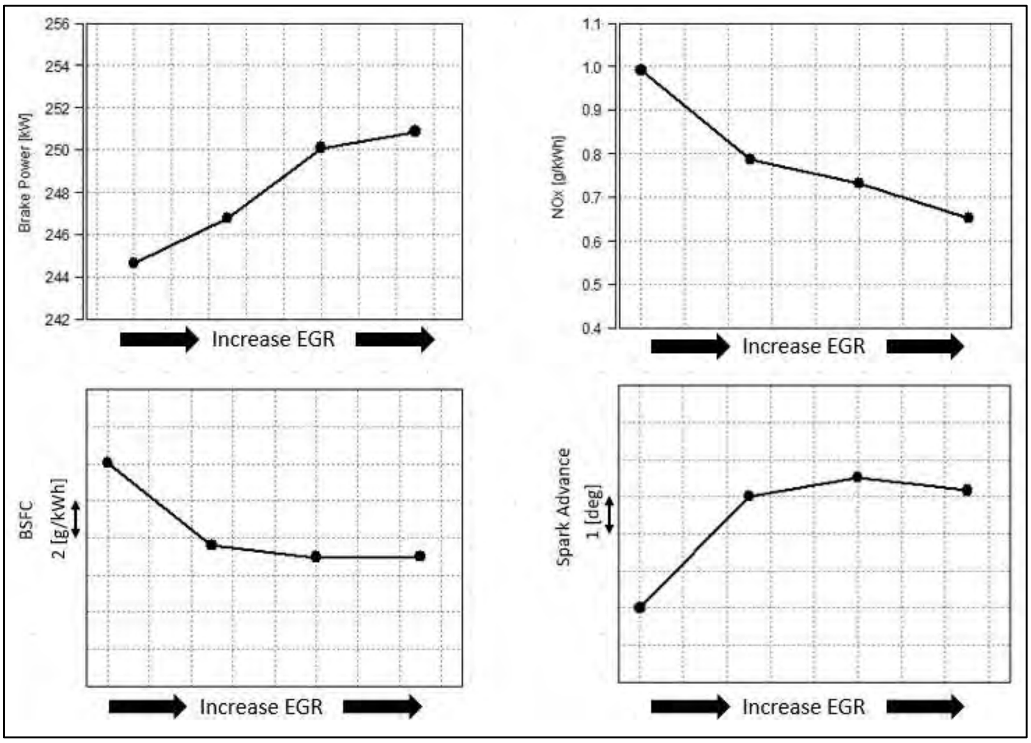


Figure 21 – H_2 V8 6.6ℓ PFI for Marine/Off-Highway application: effect of EGR on rated power.

Building on the optimized engine HW described in the Chapter 3 – whose capability has been illustrated in Figure 17 – a thorough thermodynamics calibration activity has been performed, aided by Design of Experiments (DOE) techniques. In the screening of new HW and combustion concepts, such techniques proved to be quite valuable because of the uncertainty on the actual interactions (and hence best settings) among the charging / EGR / combustion parameters, which may depart from well-known SI CNG applications due to the leaner combustion, and the different chemical features of hydrogen fuel vs methane. This turned out to be the truth not only in steady-state conditions, but also for the transient manoeuvres where a different set-up of torque reserves and build-up were applied.
Figure 22 shows the main thermodynamics parameters realized by the optimized combustion system for Marine/Off-Highway application, which explores the widest range of engine speed. The calibration of the H_2 V8 PFI was carried out with the aim of maximizing the overall efficiency within a given NOx limit as in Table 1. In particular, the goodness of new combustion chamber described in Chapter 3.1 was manifested in the freedom to choose an efficient ignition timing without incurring in knock events.
In addition, the selection of high Lambda values, to satisfy low NOx requirements and to reduce the reactivity of hydrogen mixture, leads to the achievement of exhaust temperature at turbine outlet very well matched for SCR aftertreatment systems: the overall variation across the map is much smaller than Diesel one (around 150 - 200°C for H_2-PFI), thanks to the corresponding narrower Lambda excursion.
It is also interesting to notice that the peak efficiency area is achieved in a large area of the map, boding well for the fuel consumption of such engines under various operating schedules. Finally, as discussed in detail in [12], we can observe how reaching the peak WOT conditions requires a deterioration of a set of combustion parameters, starting from the Lambda which is reduced to the minimum acceptable (knock and NOx emissions) and MFB50 which is retarded to the maximum acceptable (BTE).

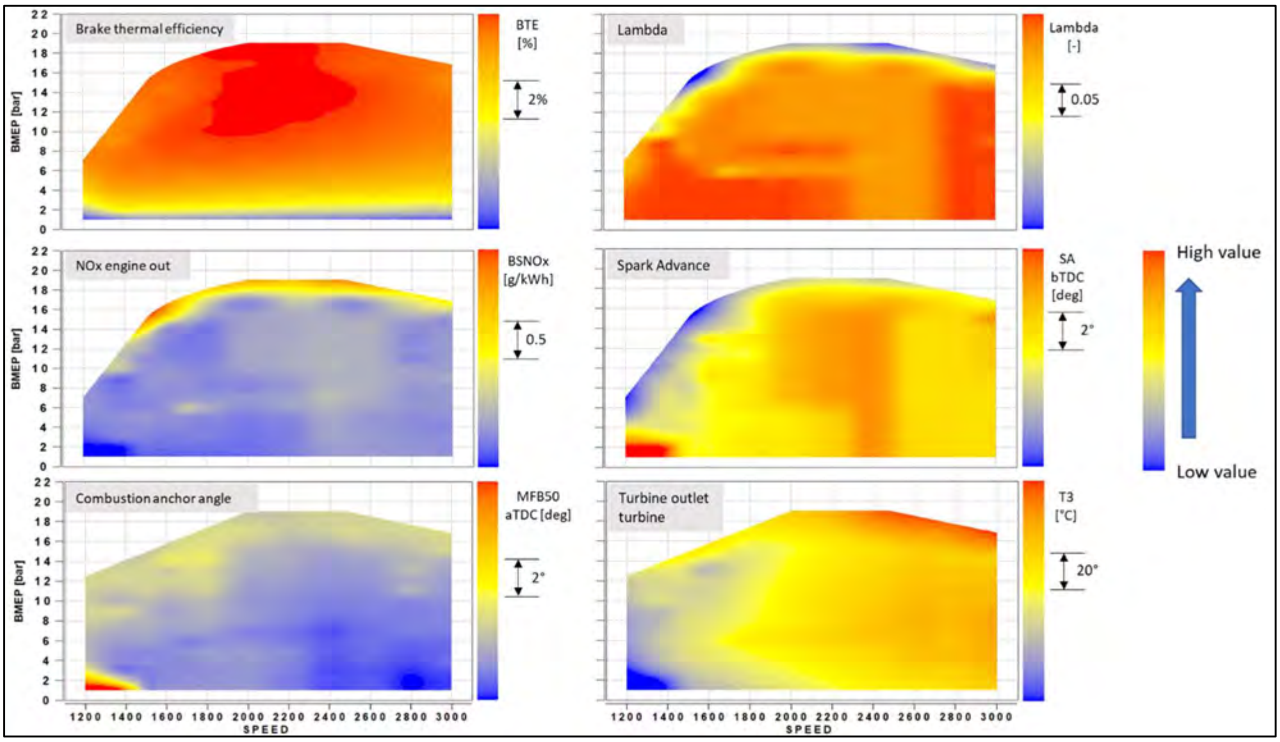


Figure 22 – H_2 V8 6.6ℓ PFI: engine maps for main thermodynamics parameters.

4.3 Transient results

The goals for transient control are torque and boost build up in terms of response time, mitigation of NOx emissions and fuel consumption.

The parameters to control are boosting, throttle valve (as applicable, depending on the starting engine load), injection and ignition timing, with the goal to minimize Lambda variations. **Figure 23** shows an example of transient mode on GenSet application during a step load event from 8 to 50 kW at fixed engine speed.

Various boost and spark advance compensations were applied to investigate the impact on engine speed variations. The target was to minimize the engine speed variations and the best choice was a calibration set-up capable to achieve the target with the lowest response time but – simultaneously – with the shortest path to achieve the Lambda target.

The correct SA tuning allows to reduce the response time and to reduce NOx production during transient maneuvers. This is because retarding SA allows to reduce NOx (due to lower peak temperature during the combustion) and to increase the available energy to the turbine: this strategy is shown in blue line as 'SA retard'.

Another investigation was made in relationship with the calibration to be adopted before the torque increase request: closing further the VGT while appropriately setting the TVA to achieve the same mass flow of the previous case 'SA retard' in Figure 24: this second strategy is called 'Opt. VGT + TVA + SA retard' and, through a faster boost build-up, provides further improvement especially in the NOx emissionization. In conclusion, the key result is that the steady-state calibration must take in account not only the standard parameters (BSFC, NOx emissions, combustion noise), but also the boundary conditions capable to provide the best conditions to the turbocharger for the transient torque build up.

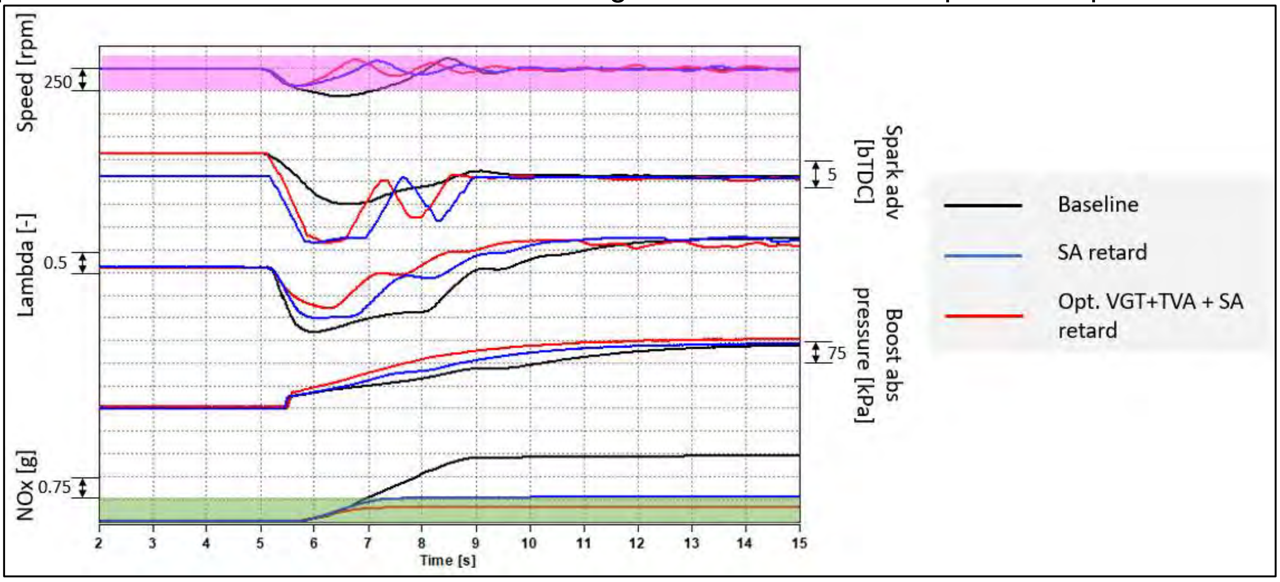


Figure 23 – H_2 V8 6.6ℓ PFI for GenSet application: G2 fast transient step-load optimization.

A further strategy to improve the engine transients, was the increase of Lambda value in the initial condition (**Figure 24**), indicated by the green curve 'Opt. Lambda + SA retard'. This latter, in a similar way to the boost increase followed by throttling described in Figure 24, allows to have an additional reserve of air that can be used to speed up the transient manoeuvrer in terms of torque and to minimize the loss of engine speed during the load step. The two promising strategies that have been identified, i.e. 'Opt. VGT + TVA + SA retard' and 'Opt. Lambda + SA retard', can be combined depending on the engine operating point at which the transient manoeuvre is initiated, as each one of them has its own pros and cons (for example the 'Opt. Lambda + SA retard' tend to be more efficient at very low load thanks to the lower pumping and heat losses, but its applicability to a PFI homogenous engine is limited by the acceptable combustion stability).

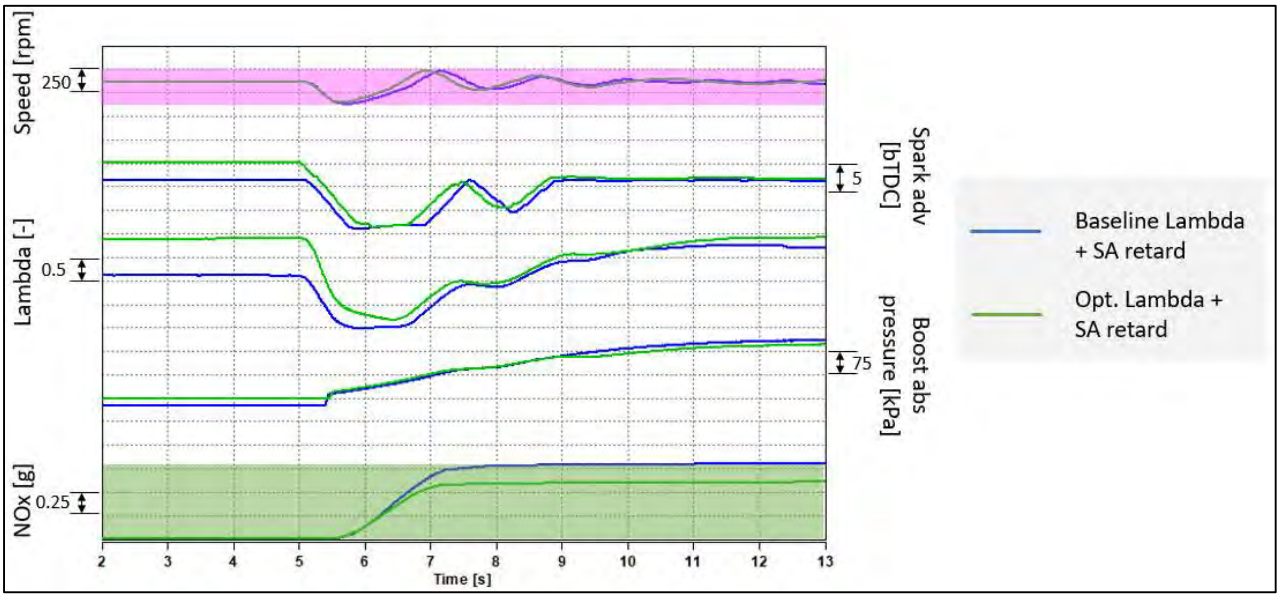


Figure 24 – H_2 V8 6.6ℓ PFI for GenSet application: optimization transient through initial Lambda.

5 Conclusions

The present paper summarizes the features of the PUNCH H_2 V8 6.6ℓ PFI engine for GenSet and Marine/Off-Highway applications. This hydrogen engine was developed starting from a state-of-the-art Diesel engine available in mass production. Key components and sub-systems were modified to achieve excellent hydrogen combustion in terms of performances, emissions, efficiency, and safety operation under any conditions and for the expected engine lifetime. This ambitious goal has required a complete re-design of fuel injection and ignition systems, piston bowl, as well as oil separation, venting and lubrication systems, and intake system. The PUNCH H_2 V8 6.6ℓ PFI comes also with a new engine control HW & SW developed in-house by PUNCH Group.
The results of up to 42 kW/l specific power, BTE more than 40% across a wide-range of the engine map, low engine-out NOx, and safe and reliable operation, bode well for a successful application in the field.

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7. Acronyms

ATS: After-Treatment System
aTDC: after Top Dead Center
bTDC: before Top Dead Center
BMEP: Brake Mean Effective Pressure
BS-FC: Brake-Specific Fuel Consumption
BS-NOx: Brake-Specific NOx (emissions)
BTE: Brake Thermal Efficiency
CA: Crank Angle
CAD: Computer-Aided Design
CFD: Computational Fluid Dynamics
CNG: Compressed Natural Gas
CR: Compression Ratio
DOE: Design of Experiments
EMS: Engine Management System
EGR: Exhaust Gas Recirculation
EOI: End of Injection
EVO: Exhaust Valve Opening
HW: Hardware
ICE: Internal Combustion Engine
IMEP: Indicated Mean Effective Pressure
IVC: Intake Valve Closing
LPG: Liquid Petroleum Gases
MAPO: Maximum Amplitude of Pressure Oscillations
MFB50: 50% Mass Fraction Burned
NOx: Nitrogen Oxides
NRMM: Non-Road Mobile Machinery
NVH: Noise, Vibration and Harshness
PFI: Port Fuel Injection
PFP: Peak Firing Pressure
SA: Spark Advance
SCE: Single-Cylinder Engine
SCR: Selective Catalyst Reduction
SOI: Start of Injection
SW: Software
TCI: Transistor Coil Ignition
TDC: Top Dead Center
TVA: Throttle Valve Actuator
T/C: Turbo-Charger
VGT: Variable Geometry Turbine
WOT: Wide Open Throttle