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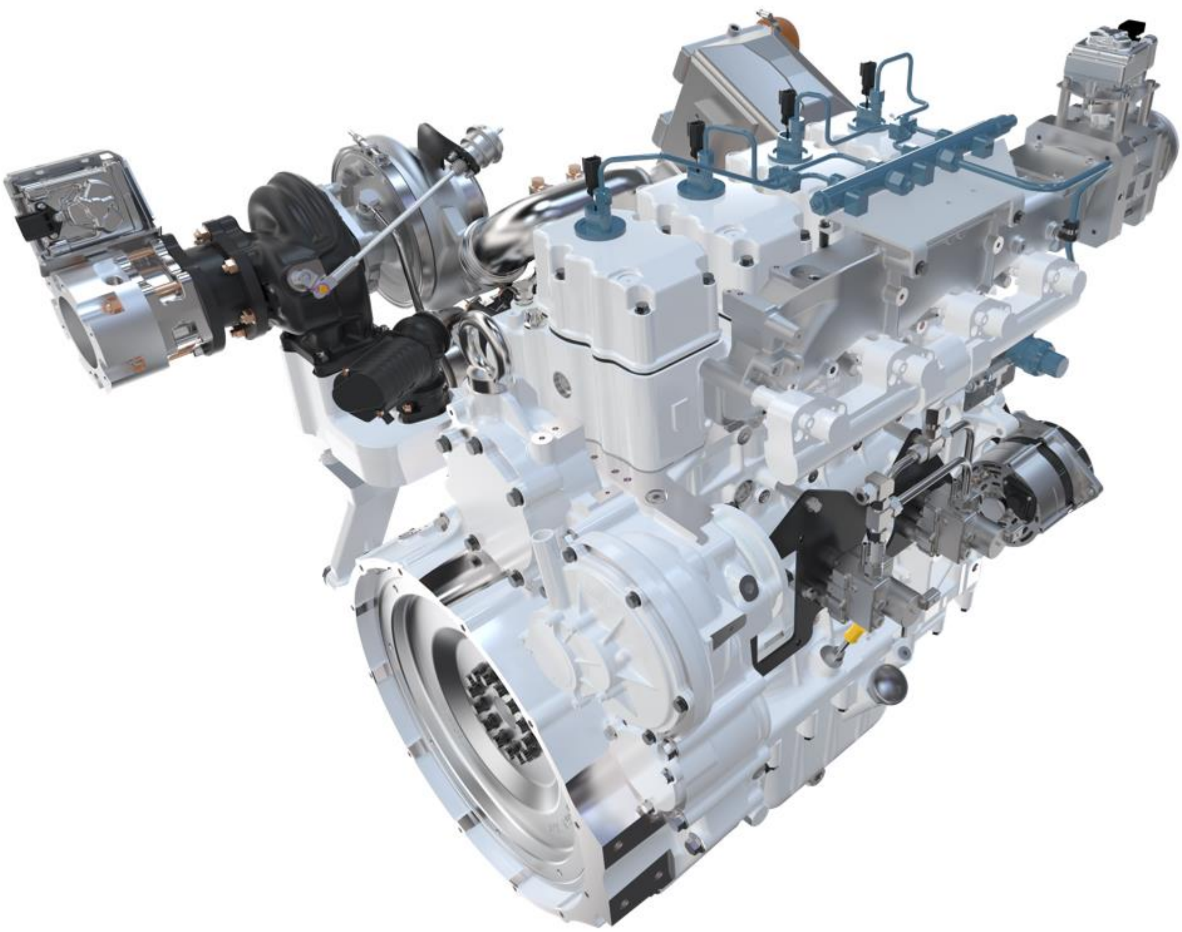
Final report

Hydrogen combustion engine with active pre-chamber (APC) technology



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Agent:

Liebherr Machines Bulle SA
Rue Hans-Liebherr 7
CH-1630 Bulle
www.liebherr.com

Authors:

Dipl. -Ing. Bakir Puran, Liebherr Machines Bulle SA, bakir.puran@liebherr.com

Dr. Nathan Peters, MAHLE Powertrain LLC, nathan.peters@mahle.com

BFE-Bereichsleiter: Dr. Carina Alles, carina.alles@bfe.admin.ch
BFE-Programmleiter: Stephan Renz, info@renzconsulting.ch
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Summary

Hydrogen combustion engines will play an important role in the decarbonization of certain sectors such as off-road vehicles and construction machinery. One of the challenging aspects of a hydrogen internal combustion engine (H2-ICE) is the limitation of the power density due to several phenomena and parameters in the combustion chamber. One solution to increase the performance and reduce emissions of the H2-ICE is the APC "active pre-chamber" technology. However, the design and control of the active pre-chamber is very challenging due to the complexity of the fuel path and the ignition system.

Several technical solutions concerning the control function, component design, and engine calibration for optimal operation with the technology APC were developed during this investigation.

The investigation of the technology shows significant improvement in terms of:

- Emission: NOx emissions reduced far below the current stage 5 regulation limit "0.4 g/kW.h"
- Dynamic and transient behavior of the combustion improved by 20% compared to a conventional H2 combustion engine (such as SI: Spark ignition) (table 6)
- Power density improved by more than 10% compared to direct injection (DI-H2, see Fig. 26)

The performance potential of the technology was demonstrated in steady state and transient application. The vehicle operation was evaluated in the NRTC "Non-Road Transient Cycle" mode.

Main Findings:

1. Complete integration of the technology without modification of the base engine achieved
2. High power density compared to port fuel injection, > 60 kW / Cylinder
3. NOx emissions below the level of the current emission regulation Stage V (< 0.4 g/kW.h)
4. Engine dynamic nearly equivalent to diesel engine (10 % - 20 % less dynamic than diesel engine)
5. Very high combustion stability demonstrated
6. Engine capable to run the NRTC "Non-Road Transient Cycle"

Technology of APC has shown a large potential regarding its application in hydrogen combustion engine. This technology offers the possibility of keeping the same engine size for replacement of the current diesel engine for different applications in Off-Road sector such as wheel loader, excavator and generally the earth moving vehicles. Several optimized variants of APC are currently under investigation in order to set up the final layout of serial development.

Additional insights for basic optical investigations (e.g., an optical candle) will be considered in the future.

This application of APC can be extended to other alternative fuel technology such as ammonia combustion engine and methanol combustion engine in order to improve the ignition behavior of such engines.

Zusammenfassung

Wasserstoffverbrennungsmotoren werden eine wichtige Rolle bei der Dekarbonisierung bestimmter Sektoren wie dem Offroad-Bereich spielen. Eine der Herausforderungen des Wasserstoff-Verbrennungsmotors (H2ICE) ist die Einschränkung der Leistungsdichte aufgrund verschiedener Phänomene und Parameter in der Brennkammer. Eine Lösung, um die Leistung des H2ICE zu erhöhen und die Emissionen zu verringern, ist die "aktive Vorkammer" APC. Die Konstruktion und Steuerung der aktiven Vorkammer ist jedoch aufgrund der Komplexität des Kraftstoffpfads und Zündsystems eine große Herausforderung.

Im Rahmen dieser Untersuchung wurden mehrere technische Lösungen entwickelt, wie z. B. die Steuerfunktion, das Komponentendesign und die Motorkalibrierung für einen optimalen Betrieb mit der APC-Technologie.
Die Untersuchungen mit dieser Technologie zeigen folgende Verbesserungen des Motor-Verhaltens:

- Die NOx-Emissionen wurden weit unter den aktuellen Grenzwert der Stufe 5 "0,4 g/kW.h" gesenkt.
- Verbesserung des dynamischen und instationären Verbrennungsverhaltens um 20 % im Vergleich zu herkömmlichen H2-Verbrennungsmotoren (z. B. SI) (Table 6)
- Verbesserung der Leistungsdichte um 10 % im Vergleich zu H2-DI (Fig. 26)

Das Potenzial der Technologie wurde im stationären und transienten Betrieb nachgewiesen. Zur Bewertung der Technologie im Fahrzeugbetrieb wurde der Testzyklus NRTC "Non Road Transient Cycle" herangezogen.

Haupterkenntnisse:

1. Die vollständige Integration der Technologie ohne Änderung des Basismotors ist erfolgt
2. Hohe Leistungsdichte im Vergleich zur Saugrohreinspritzung (PFI-Technology), > 60 kW/Zylinder
3. NOx-Emissionen unter dem Niveau der aktuellen Emissionsvorschriften Stufe V (< 0,4 g/kW.h)
4. Motordynamik nahezu gleichwertig zum Dieselmotor (nur 10 % - 20 % weniger Dynamik als Dieselmotor)
5. Sehr hohe Verbrennungsstabilität nachgewiesen
6. Motor für den Betrieb im NRTC "Non-Road Transient Cycle" geeignet

Die APC-Technologie hat in Bezug auf ihre Anwendung in Wasserstoffverbrennungsmotoren ein großes Potenzial gezeigt. Diese Technologie bietet die Möglichkeit, die gleiche Motorgröße beizubehalten, um den derzeitigen Dieselmotor für verschiedene Anwendungen im Off-Road-Bereich wie Radlader, Bagger und allgemein Erdbewegungsfahrzeuge zu ersetzen. Mehrere optimierte Varianten des APC werden derzeit untersucht, um das endgültige Layout der Serienentwicklung festzulegen.

Für die Zukunft werden weitere Untersuchungen für optische Grundlagenuntersuchungen (z.B. eine optische Kerze) in Betracht gezogen.

Diese Anwendung der APC kann auf andere alternative Kraftstofftechnologien wie Ammoniak- und Methanolverbrennungsmotoren ausgeweitet werden, um das Zündverhalten solcher Motoren zu verbessern.

Résumé

Le moteur à combustion à hydrogène jouera un rôle important dans la décarbonisation de certains secteurs tels que les activités non-routières et de la construction. L'un des défis majeurs du moteur à combustion interne à hydrogène (H2-ICE) est la limitation en termes de la densité de puissance due à plusieurs phénomènes et paramètres de la chambre de combustion. Une solution pour améliorer les performances et les niveaux des émissions du moteur H2-ICE est la technologie de la « préchambre active », APC. Cependant, la conception et la régulation de la préchambre active sont très difficiles en raison de la complexité du parcours du combustible et du système d’allumage.

Plusieurs solutions techniques telles que la fonction de contrôle, la conception des composants, la calibration du moteur pour un fonctionnement optimal avec la technologie APC ont été développées au cours de cette étude.

L'investigation avec la technologie montre une nette amélioration de la combustion en termes de :

- Emission : Les émissions de NOx ont été réduites bien en dessous de la limite réglementaire actuelle de stage 5 "0,4 g/kW.h"
- Amélioration du comportement dynamique et transitoire du moteur de 20 % par rapport aux moteurs à combustion H2 conventionnels (tels que l’allumage par bougie) (Table 6)
- Amélioration de la densité de puissance de plus de 10 % par rapport au DI-H2 (injection directe, voir Fig. 26)

Le potentiel de la technologie a été prouvé dans des applications en régime permanent et transitoire. Le fonctionnement du véhicule en utilisant le NRTC "Non-Road Transient Cycle" a été considéré pour évaluer la technologie.

Principales conclusions :

1. L'intégration complète de la technologie sans modification du moteur de base a été réalisée
2. Densité de puissance élevée par rapport à l'injection indirecte (PFI : Port Fuel Injection) > 60 kW / Cylindre
3. Emission de NOx inférieure au niveau de la réglementation actuelle sur les émissions (< 0,4 g/kW.h)
4. Dynamique du moteur presque équivalent à celle du moteur diesel stage V (10 % - 20 % moins dynamique que le moteur diesel)
5. Très grande stabilité de combustion démontrée
6. Moteur capable de réaliser le cycle transitoire dans le domaine non-routier NRTC "Non-Road Transient Cycle"

La technologie APC a montré un grand potentiel en ce qui concerne son application dans les moteurs à combustion d'hydrogène. Cette technologie offre la possibilité de conserver la même taille de moteur pour remplacer le moteur diesel actuel pour différentes applications dans le secteur des véhicules off-road, telles que les chargeuses sur roues, les excavatrices et, plus généralement, les véhicules de terrassement. Plusieurs variantes optimisées de l'APC sont actuellement à l'étude afin d'établir le schéma final du développement en série.

À l'avenir, d'autres études seront envisagées pour des recherches optiques de base (par exemple, une bougie optique).

Cette application de l'APC peut être étendue à d'autres technologies de carburants alternatifs telles que les moteurs à combustion à l’ammoniac ou au méthanol afin d'améliorer le comportement d'allumage de ces moteurs.

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Abbreviations

1D	One dimension
AC	Alternating Current
APC	Active pre-chamber
BLDC	Brushless Direct Current
BMEP	Break Mean Effective Pressure [bar]
BSFC	Break Specific Fuel Consumption [g/kWh]
CA	Crank Angle [°]
CFD	Computational Fluid Dynamics
DC	Direct Current
E-booster	e-charger
E-charger	Electrical charger
ECU	Engine Control Unit
E-Turbocharger	Electrical Turbocharger
FM_EC	Supplier brand 1_e-charger
FU_m_CYL	Injected fuel quantity [mg/stroke]
HIL	Hardware in the Loop
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure [bar]
LMB	Liebherr Machines Bulle
NOx	Nitrogen Oxide ($NO + NO_2$)
RPM	Rotation per Minute [1/min]
SIL	Software in the Loop
SOI	Start of Injection [°CA]
TB	Test bench
TDC	Top Dead Center
w/o	without
WP	Work package

1 Introduction

1.1 Purpose of the project

Due to their high degree of performance, efficiency and robustness, Diesel combustion engines have long been the standard solution for heavy-duty and off-road applications. The current legislation on emission regulation is reliably complied with using exhaust after-treatment systems. The increasingly strict emission standards on the one hand and the growing acceptance of alternative drive concepts on the other hand, has forced manufacturers of combustion engines to innovate in powertrain systems.

Several milestones have already been defined by EU-Regulations for On-Road "heavy-duty" sectors. CO_2 emission targets for heavy-duty vehicles from 2025 and 2030 respectively must reach a 15% and 30% reduction versus EU average CO_2 emissions during the reference period (1 July 2019 – 30 June 2020) as illustrated in Figure 1.

Regulation on CO_2 emission standards for heavy-duty vehicles

Until the revision is adopted, the Regulation on CO2 emission standards for heavy-duty vehicles applies. This Regulation entered into force on 14 August 2019.

As per the Regulation, from 2025 onwards, manufacturers will have to meet the targets set for the fleet-wide average CO_2 emissions of their new lorries registered in a given calendar year. Stricter targets will start applying from 2030 on.

The targets are expressed as a percentage reduction of emissions compared to the EU average in the reference period (1 July 2019–30 June 2020):

- From 2025 onwards: 15% reduction
- From 2030 onwards: 30% reduction

The 2025 target can be achieved using technologies that are already available on the market. The 2030 target has been assessed as part of the review of the Regulation.

As a first step, the CO_2 emission standards cover **large lorries**, which account for over 73% of all CO_2 emissions from heavy-duty vehicles.

Figure 1: EU Regulation for heavy duty vehicles / *Source : https://ec.europa.eu/clima/policies/transport/vehicles/heavy_en*

Currently, the CO_2 emission regulations for heavy-duty are based on a "Tank-to-Wheel" approach instead of a "Well-to-Wheel" approach. The latter considers the production, transport, distribution, and operation of the fuel.

In order to reach the target regarding the EU's 2050 climate neutrality, zero-emission technologies (ZE) and low emission technologies (LE) will be introduced by several OEMs in the future. Hydrogen, Ammonia and electric chemical carriers (such as Lithium batteries) are the most known energy carriers to realize a ZE (Zero emission) powertrain. Fuel cells (H2-FC) and combustion engines (H2-ICE) are the main powertrains which use hydrogen as an energy carrier.

H2-ICE: Internal combustion engine operating with H_2 is considered as a zero-emission powertrain since the only CO_2 emission results from burned oil particles in the combustion chamber. These CO_2 emissions are considerably below the regulated limit in this sector (Heavy duty On-road, CO_2 < 1 g/kWh). The engine out CO_2 emission is not the only performance indicator for H_2-ICE. An optimization of the technical concept of the hydrogen combustion engine should be performed according to the following additional indicators:

- Performance and NOx emission
- Power density

- Behavior in transient operation
- Robustness of the technical solutions regarding the components

To comply with applicable emissions regulations (for example stage 6 regarding NOx and particulates), exhaust gas after-treatment systems (EATS) must be used. These systems can be expensive and require maintenance. New combustion technologies can enable the use of simplified EATS. For example, further enleanment reduces the NOx engine out emissions which has a positive impact on the complexity of EATS system.

Current hydrogen internal combustion engines use either a direct injection (DI) or a port fuel injection (PFI) system. With both systems, engine performance (power density) decreases due to the prevalence of abnormal combustion inherent to hydrogen. This is not acceptable for some heavy-duty applications. In order to obtain the best performance and emission characteristics while keeping a high level of efficiency, it is necessary to develop a new ignition system that can meet these objectives. Active Prechamber injection and ignition strategy is one possible solution which allows improvement in engine performance under ultra-lean conditions.

Hydrogen as Fuel in ICE and challenges: Hydrogen as fuel has been the object of several investigations in different applications such as the aerospace, automotive sectors and industrial applications. A considerable experience in fuel compatibility with different materials and fuel properties has been gained in the past. The specific properties of hydrogen and those of several different fuels have been summarized in Table 1. The hydrogen fuel has a high gravimetric calorific value; however, its lower density reduces the volumetric heating value of the mixture, which leads to lower performance especially for some injection concepts such as hydrogen port fuel injection (PFI). This characteristic has a direct impact on the torque performance of the engine, which decreases with higher air/fuel ratio. This inconvenience is a challenging issue in both heavy-duty and off-road applications due to packaging requirements. H_2 direct injection (H_2 DI) is one of the possible ways to improve volumetric efficiency especially if the fuel injection occurs after closing the inlet valve.

	Unit	**Diesel**	**Methane**	**Hydrogen**	**OME_{3-5}**	**NH3**
Lower heating value	MJ/kg	42.8	50	120	18.5-19.5	18.6
Density @15°C, 1 atm	kg/m³	840	0.7 (gas)	0.09 (gas)	>1039	682
Minimum ignition energy at lambda = 1	mJ	0.24	0.29	0.02	N/A	680
Auto ignition temperature	°C	Min. 225	595	590	>210	650
Equivalent air/fuel ratio	[-]	14.7	17.2	34.3	6.1-5.6	6.05
Laminar flame speed	cm/s	40	42	230	N/A	7

Table 1 : Fuel characteristics [1], [2], [3], [4], [5], [6]. N/A: Not available

As we can observe in Table 1 the characteristics of hydrogen fuel compared to other fuels are:

- High combustion flame speed
- Wide flammability range
- High net calorific value
- Lower ignition energy
- Higher ignition temperature

The wider range of flammability offers the possibility to operate the engine in ultra-lean combustion region which leads to ultra-low NOx emission. However, the design of combustion for ultra-lean operation requires several adaptations in the engine architecture such as air and fuel path configuration.

While these properties offer good efficiency, the main drawback with current hydrogen ICE is their limited torque and power, which are incompatible with customer requirements. The lack of torque and power in hydrogen engines can be attributed to several factors, the main ones being the low density of hydrogen and abnormal combustions such as knocking. In the near future, priority must therefore be given to the development of hydrogen combustion engines with higher specific power while retaining the gain in terms of emissions and efficiency.

1.2 Introduction to application in off highway sectors

Liebherr engines are being used in several application in Off-highway and industry sectors, the vehicles have been grouped in the following 4 categories:

- Earth moving
- Mobile cranes
- Mining
- Rail and maritime applications

The vehicles can operate in very harsh working condition and require a high level of robustness:

- Hot and cold temperatures condition
- Higher transient application such as excavator
- Operation at higher humidity
- Vibration/Shock Profile
- Operation at higher altitude (5000 m)
- Lifetime requirement

Figure 2 shows several vehicles of Liebherr and their related operating characteristics.

Several engine sizes and configurations have been developed to fulfill the specific requirement of each application. Some applications require adaptation of certain components such as turbocharger to achieve high torque requirement at low speed and high-power performance at high altitude. Additionally, there are some applications, such an excavator, that have a high dynamic requirement in transient operation.

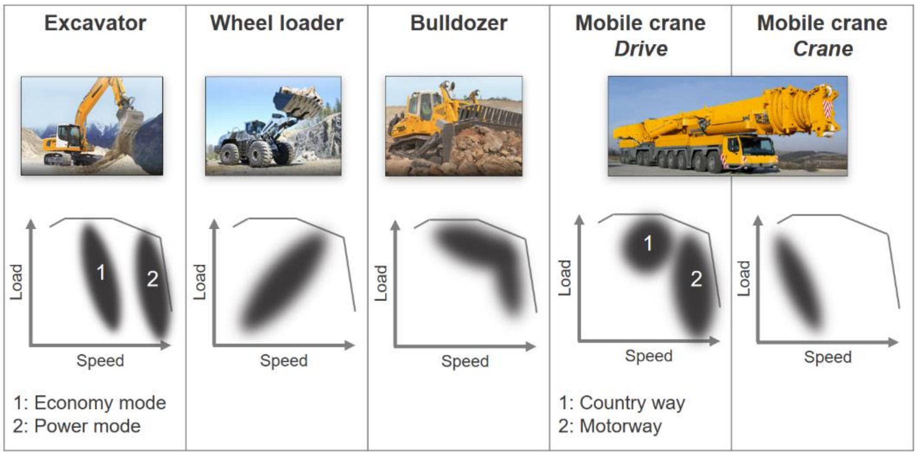


Figure 2: Liebherr's vehicles and their related operating characteristics

The current investigations are based on the engine family D96x and this platform is considered for hydrogen combustion engine activities. However, several H_2 hardware components and software developments can be applied to other engine families as well thanks to the Liebherr engine platform modular design.

1.3 Pre-chamber Combustion

A pre-chamber is a chamber proportionally smaller than, and directly connected to, the combustion chamber. Its historical uses have spanned low pressure fuel delivery, spark plug protection, and use directly as an ignition system.

Its use as an ignition system was first applied to spark ignited (SI) engines through Sir Harry Ricardo's patent for the Ricardo Dolphin engine, on which he began development in 1903. The most prominent automotive production applications of the pre-chamber concept in the latter half of the 20th century were produced by Honda (the Compound Vortex Controlled Combustion, or CVCC, engine), Volkswagen, and Toyota.

Images of the Ricardo Dolphin and Honda CVCC pre-chambers are shown in Figure 3. These pre-chamber concepts initiate the combustion process in the pre-chamber, with the products of this combustion process then transferring to the main chamber and subsequently causing those contents to ignite. Pre-chamber SI combustion concepts have been researched extensively for passenger car applications and for large bore off-road gaseous fuel applications.

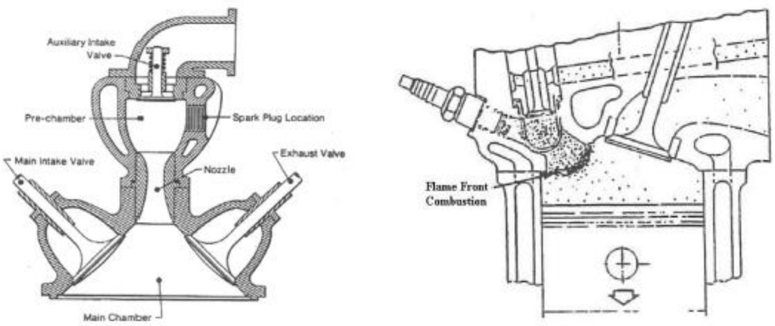


Figure 3: Ricardo Dolphin (left) and Honda CVCC pre-chambers (right) [7]

The pre-chambers described above contain the spark plug and an auxiliary fuelling source, allowing for a de-coupling of the chamber air/fuel ratios. These systems are known as active pre-chambers. Some pre-chamber concepts developed after the Dolphin concept, removed the separate pre-chamber fuelling feature. In these designs, known as passive pre-chambers, fuel is injected conventionally into the main chamber and piston motion during the compression stroke forces a volume of this fuel-air mixture proportional to the pre-chamber volume to enter the pre-chamber. This type of design reduces hardware, controls, packaging complexity, and provides a more stable combustion event compared to a conventional SI engine, though enleanment capability is limited when compared with that of an active pre-chamber. Generalized images of active and passive pre-chamber configurations are depicted in Figure 4.

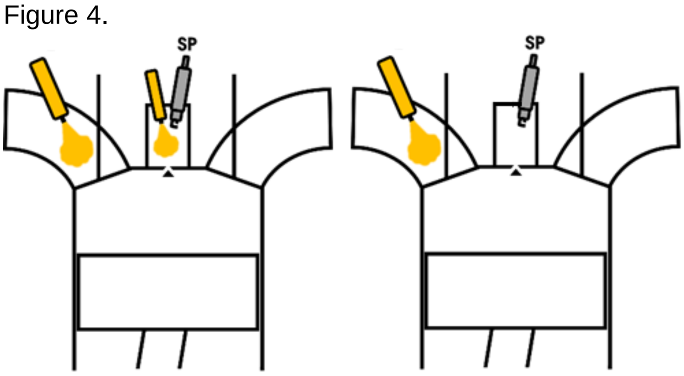


Figure 4:Generalized active (left) and passive (right) pre-chamber configurations

Jet Ignition

A chemical kinetically controlled pre-chamber-based combustion mode known as jet ignition was first researched by Nicolai Semenov in the 1950s, followed shortly by pioneering research by Lev Gussak. While it retains the pre-chamber combustor, jet ignition differs primarily from its antecedent by the manner in which combustion translates from the pre-chamber to the main chamber. In the case of jet ignition, a high degree of quenching of the pre-chamber combustion products occurs as the products exit through a multi-orifice nozzle. This allows the products entering the main chamber to manifest as high velocity jets of partially combusted radical species. After a short delay, these jets thermo-chemically initiate combustion in the main chamber.

Dilute SI Combustion

Dilute combustion in SI engines has been proven to be one of the most effective means to produce substantial increases in thermal efficiency. There are typically two methods of dilution explored in SI engines: lean operation (with air dilution) and dilution through exhaust gas recirculation (EGR). While homogeneous lean combustion necessitates the use of lean aftertreatment, the use of EGR allows the engine to maintain in stoichiometric operation, thereby enabling the use of a conventional 3-way catalyst for emissions control. The key trade-off to reduced aftertreatment complexity is the reduced thermal efficiency potential of EGR compared to lean air dilution, all else being equal, resulting from the differences in the ratios of the specific heat capacities.

Dilute combustion also reduces the likelihood of knock in SI engines. The colder combustion temperatures that result from dilute combustion reduce the temperature residence of end gases ahead of the flame front. Since knock requires a thermal mechanism, the likelihood of it occurring is consequently reduced. The thermal efficiency potential of a dilute system can then be compounded through the use of a higher compression ratio, enabled by the reduced knock likelihood, in addition to the reduced pumping and in-cylinder heat losses inherent to dilute combustion.

Traditional spark plugs are limited in their ability to enable dilute combustion. This limitation is manifested as misfire due to poor kernel formation and partial burning due to arrested flame development. The application of jet ignition distributes ignition energy throughout the combustion chamber, with flame fronts that propagate from the resulting ignition points needing to traverse less physical space and individually consume less fuel for the combustion process to achieve completion. The dilution limit extension is indicated by the significant reduction in burn duration with jet ignition versus conventional SI at similar dilution ratios.

APC (Active Pre-Chamber) with H_2

Active Pre-Chamber incorporates a direct fuel injector in the pre-chamber as shown in Figure 4. This enables precise control of fuel metering and spray targeting. The latter is necessary to minimize particulate formation during the pre-chamber combustion event. Direct fuel injection also provides the opportunity for injection late in the cycle. A fuel injection event that occurs too early can result in “over-mixing” as seen in Figure 5, producing an overly dilute mixture near the spark plug, posing a risk of misfire. A fuel injection event late in the compression stroke is therefore desired to ensure an ignitable mixture near the spark plug and maximize the quantity of auxiliary injected fuel that participates in the pre-chamber combustion event.

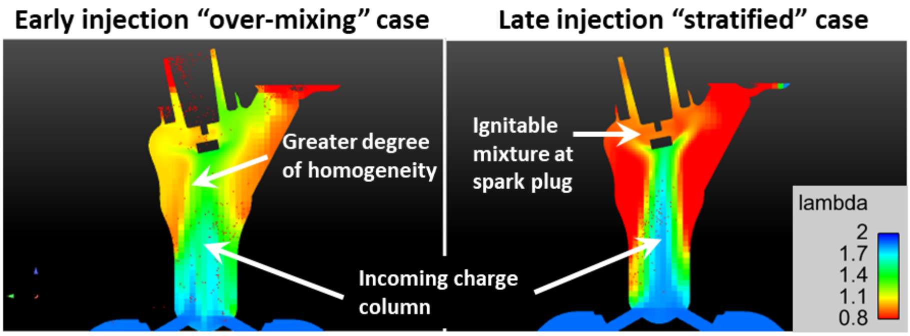


Figure 5: Mixture preparation with early (left) and late fuel injection (right) timing in the pre-chamber at time of spark with constant pre-chamber fuel quantity (Mahle power train) [8]

This innovation to the fuel delivery system for the active pre-chamber concept is viewed as critical for 1) successful operation with a wide variety of liquid and gaseous fuels, and 2) efficient, judicious use of the pre-chamber fuel in order to preserve the system efficiency.

1.4 Objectives

The goal of the project is the development, integration, and testing of the new ignition and injection unit (APC) for hydrogen combustion engine in order to confirm the potential of this new technology regarding:

- Performance and efficiency
- Emission
- Dynamic of the engine

Figure 6 shows the concept of the APC unit.

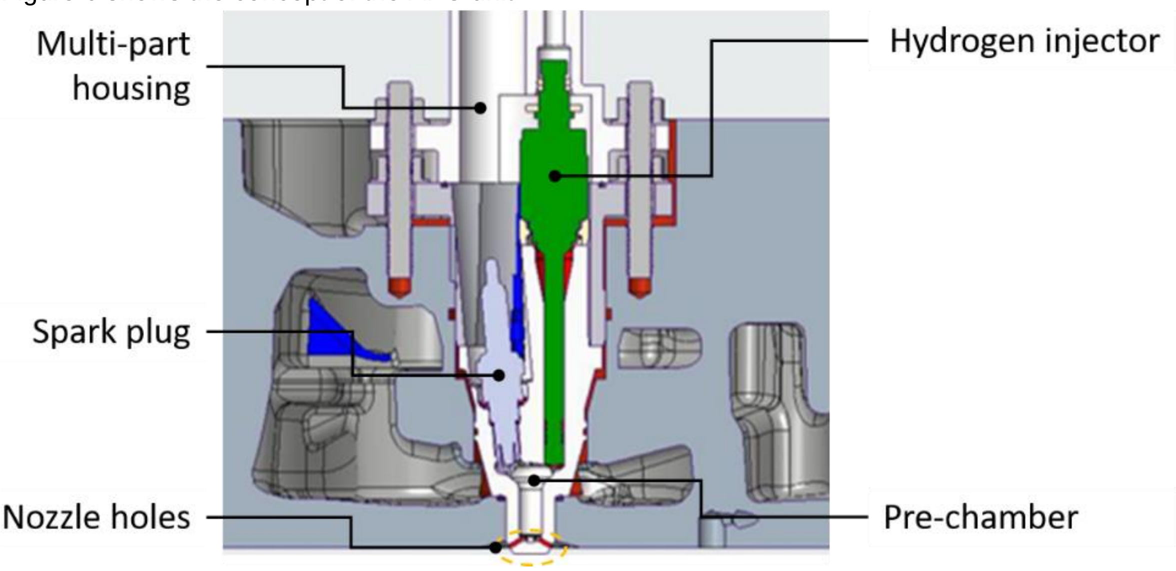


Figure 6: Concept of the Active pre-chamber injection. Image courtesy of MAHLE Powertrain.

The APC unit contains a hydrogen injector, a spark plug and the nozzle, which are integrated in a multi-part housing. This multi-part housing ensures easy assembly, as well as a good possibility of maintenance, so that internal components like the hydrogen injector or the spark plug can be exchanged without dismounting the unit.

The main objectives of the project are:

1. Ignition unit for zero emission engine without exhaust after-treatment system to fulfill EURO7
2. Perform ultra-lean combustion (lambda between 2.5 and 3) to obtain ultra-low NOx emissions
3. Improvement of knocking resistance limits
4. Increase the power density of the engine to reach the equivalent power displacement of the diesel engine
5. Achieve a high engine efficiency (target is set to 43 - 44%)
6. Obtain good engine dynamics to be the most equivalent to the diesel engine

These improvements are verified using CFD simulation and by performing several engine tests. Furthermore, one of the goals of the project is to investigate and to improve energy management and the efficiency of the internal combustion engines.
Due to the time plan, it was difficult to realize the planned modification of the single cylinder engine for optical measurements. Therefore, it was consciously planned to develop in advance a concept study for the realization of the optical single cylinder engine to be ready after the engine measurements with the APC technology.

2 Challenges and approach

2.1 Challenges

The current APC technology on the market is only designed and specified for motorsport applications running on gasoline and heavy-duty applications running on natural gas and thus have very different requirements and demands compared to heavy-duty and off-road applications with high power engine operating at relatively low speed with hydrogen.

To achieve the project goals, development is required in following fields:

Design and Simulation

The development of the APC unit requires a very good understanding of the thermal processes and fluid mechanics in the main combustion chamber and the active pre-chamber. The two chambers interact with each other during each engine work cycle, which influences the gas mixture, stratification, and lambda value in the pre-chamber. The mass of the hydrogen-air-gas mixture increases in the pre-chamber during the compression stroke of the engine, due to the constant volume of the pre-chamber, depending on the pressure difference between the pre-chamber and the main chamber. This is caused by the geometric parameters (number of holes, total cross-sectional area of the holes, chamber geometry) of the pre-chamber.

One of the challenges is to define the geometry of the pre-chamber (shape of the pre-chamber volume, number of holes, diameter of the holes, orientation of the holes) and to obtain a good knowledge of the lambda value in the active pre-chamber to determine the amount of pilot injection required at all engine operating points. In addition, the mixing behavior of the pre-injected mass in the pre-chamber and its stratification must be kept under control to prevent exceeding a certain lambda value during the ignition phase, which would otherwise increase the risk of knocking.

Other challenges are defining the spark timing, ignition, jet timing, jet velocity, spray angle and burn duration in the active pre-chamber. To achieve acceptable combustion rates, an optimum ratio between the volume of the pre-chamber and the total cross section of the nozzle holes must be determined.
Regarding the integration of the APC system in the cylinder head, it is important to ensure the proper cooling of the whole unit. This task will be performed by combining heat transfer calculation and structural analysis. The design of the active pre-chamber will be performed using target values of the pressure difference (between the main combustion chamber and the pre-chamber) and the temperature in the pre-chamber. All this CFD work will be performed for multiple engine operating points (Table 2).

Engine speed [rpm]	Engine load [%]
1000	100
1500	100
1900	50
1900	100

Table 2 : CFD operation points

CFD simulation conditions will be defined using results from the 1D-combustion model (AMESim, GT Power). The 1D-models are based on experimental data and theoretical background of the combustion. CFD data will be compared to combustion measurements with the APC unit.

Software

Besides the mechanical development, it is necessary to develop new software and the APC controls strategy. The controls strategy needs to allow engine operation in manual and automated modes with calibrated maps. The manual mode is useful for engine commissioning, first firing and initial calibration refinement to understand engine behavior, whereas the automated modes operate with defined injection and ignition maps based on engine speed and load.

Furthermore, the current Liebherr ECU is not capable of controlling more than 6 injectors, so an alternative ECU hardware solution is mandatory.

Engine and test bench commissioning

The engine and the hydrogen test bench are completely new which need to be commissioned both at the same time. Generally, this situation is not ideal and needs to be well coordinated. To check all test bench functionalities and the security measures before testing the hydrogen engine, a well calibrated and operational hydrogen engine is needed. As this cannot be realized at the same time, many sequential steps and loops during this commissioning process are necessary.

Backfire

Hydrogen engines with port fuel injection have the tendency to backfire, caused often by hot residual gases or hot spots in the combustion chamber, where the injected hydrogen can pre-ignite during the injecting event. Due to the very fast hydrogen flame velocity, the flames can pass the intake valves and access the intake manifold. Once this issue occurs, in most cases the regular engine operation cannot continue as this affected combustion chamber usually remains in the backfire mode. Furthermore, the other combustion chambers start to have significant knock events due to increased intake air temperature with cylinder pressure peaks higher than maximum limit.

Nevertheless, there are still process parameters like lower combustion temperature, optimized injection timing or certain exhaust back pressure regions which should reduce the risk of backfire or at least move the backfire region towards higher engine load.

2.2 Approach

WP1: Project definition, design of the APC unit and the fuel paths, CAD integration on engine, CAD integration of the new turbocharging system, design analysis and release.

After the project definition, the first phase of the project was the pre-design of the APC unit and the CAD modification of the cylinder head to create different integration concepts. Furthermore, the sealing design between combustion chamber, APC, engine coolant and lubrication oil were defined. After the concept evaluation, the best solution was selected, and the final design (3D CAD) was fixed. The final design defines the dimensions of the APC unit and the seat in the cylinder head. Furthermore, the matching of the new turbocharging system defines its design, selection, and integration on the engine. WP1 was accomplished with CAD releases.

WP2 CFD simulation of the APC unit, cold flow analysis, evaluation, and selection for procurement

In the second phase, the focus was set on the CFD simulation to define the size of the pre-chamber and the nozzle geometry and thus to analyse the cold flow behaviour of the APC unit. Moreover, the cooling and thermal behaviour of the APC unit needed to be analysed as well using CFD tools.

After the simulation analysis of different APC versions, 3 most promising APC definitions were chosen for final geometry definition. The focus was on the pre-chamber volume, nozzle geometry and the number of nozzles. WP2 was accomplished with procurement release.

WP 3 Procurement

After the final geometry definition of the APC unit, the focus in this phase was set on finding appropriate suppliers, hardware procurement, regular monitoring of the manufacturing progress and hardware delivery. The quality of the delivered hardware was checked, and the several sub-systems were pre-assembled in order to ensure the fitting capability.

WP 4 Investigation for optical component testing

In this phase, it was planned to perform a concept study of possible single cylinder modification in terms of optical combustion investigation. Potential suppliers should be contacted and offers collected to estimate time and cost for such a realisation.

As the result of WP 4, the study would include the analysis of state of the art, requirement specifications, implementation support, proposal for the technical realisation and a detailed recommendation.

WP 5 Engine testing

In this phase, the original diesel engine is converted into a hydrogen engine with the APC technology and ready for the testing on the engine test bench. After the engine commissioning, the testing is performed in several steps:

1. Engine calibration in PFI + SI configuration, finalized by the steady-state reference measurements
2. Engine calibration in passive pre-chamber configuration (APC injector not injecting), finalized by steady-state and NRTC measurements
3. Engine calibration in active pre-chamber configuration (APC injector injecting), finalized by steady-state, load steps and NRTC measurements
4. Engine calibration in active pre-chamber configuration (APC injector injecting) and EGR, finalized by steady-state, load steps and NRTC measurements

WP 6 Measurement analysis and report

The last phase contains the measurement analysis and evaluation, which is finalized by a project report. Furthermore, the results are defining the required specification on the APC technology and are important for the potential transfer to series development.

3 Basics and boundary conditions

3.1 Liebherr Diesel engine

The Liebherr diesel engine D964 with 4 cylinders in inline-arrangement and a displacement of 9 liters has a maximum power of 300kW at 1900 revolutions/min. The engine features for regulated and unregulated markets identical performances, the same requirements for the machine's cooling system and the same interfaces when installed. This enables the customer to use the same device design for different emission standards. The engine D964 applies in various Liebherr applications including the off-road sector.

This engine type was used as basis platform for the conversion into a hydrogen engine with the APC technology. The main specifications of the diesel engine are listed in Table 3.

<table>
<tr><td>Configuration</td><td>In-line engine</td><td rowspan="12"></td></tr>
<tr><td>Number of cylinders</td><td>4</td></tr>
<tr><td>Flywheel housing</td><td>SAE 1 / SAE 2</td></tr>
<tr><td>Bore</td><td>135 mm</td></tr>
<tr><td>Stroke</td><td>157 mm</td></tr>
<tr><td>Displacement</td><td>9.0 l</td></tr>
<tr><td>Rated power</td><td>200 - 300 kW</td></tr>
<tr><td>Rated speed</td><td>1500 – 1900 rpm</td></tr>
<tr><td>Max. torque</td><td>1739 Nm</td></tr>
<tr><td>Dimensions (L/W/H)</td><td>1015 / 838 / 1116 mm</td></tr>
<tr><td>Dry weight</td><td>735 kg</td></tr>
<tr><td>Auxiliary outputs (PTO)</td><td>3</td></tr>
<tr><td>Emissions standards</td><td colspan="2">EPA Tier 0 (Fuel consumption optimized) / EPA Tier 4f / EU Stage IIIA / EU Stage V</td></tr>
</table>

Table 3 : Main characteristics of the Liebherr Diesel engine D964 – 9L

The decision to use the D964 as the base engine was made due to the following reasons. As this engine is well known for its robustness, it was a suitable candidate for testing the new APC technology. Furthermore, finding the performance limits of the engine means also operating often in the range of knock limit with large cylinder pressure peaks as a consequence.
Another reason concerns the total project cost. A 4-cylinder engine allows steady-state and transient measurements by keeping the amount of the necessary hardware and spare parts at a minimum. Measures on a single cylinder engine could be even less expensive, but as the potential analysis of APC technology needs to be done also for transient engine operation, the single cylinder engine was no option.

3.2 Engine schematic of the APC engine

Figure 7 and Figure 8 show two schematics of the engine H964 APC with 2-stage turbocharging system. The first engine version is without EGR, whereas the second one is equipped with EGR. Normally, the gas mixture in the combustion chamber can be diluted by increased lambda using the 2-stage turbocharging system. This leads to lower combustion temperature with lower knock risk. The other possibility is to use/add EGR in order to reduce the combustion temperature.

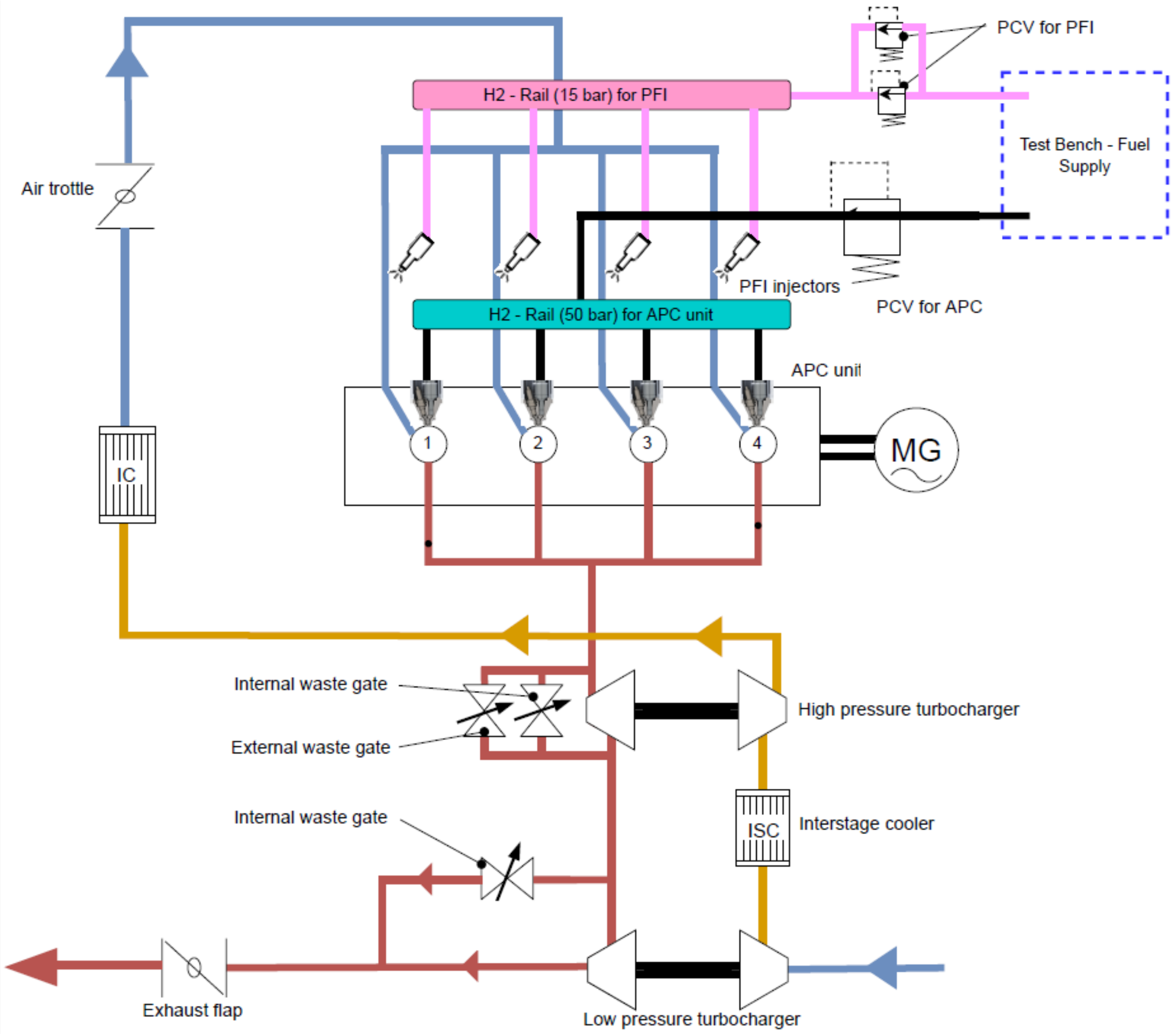


Figure 7: Schematic of the H964-APC engine without EGR application

The air path includes two compressor stages with integrated interstage cooler, intercooler, air throttle and intake manifold. As the APC engine has PFI injectors for the main injection quantity and the APC injectors for pilot injection, the engine needs two separate fuel paths. The PFI path contains two in parallel connected pressure control valves (PCV), one H_2-rail and four PFI injectors. The rail pressure is controlled between 4 and 15 bar. The APC fuel path contains a mechanical PCV, which operates at constant pressure (50 bar), an H2-rail and 4 APC units.

The exhaust path is equipped with an additional external electrical waste gate installed in parallel to the internal waste gate of the high-pressure turbocharger stage. With this electrical waste gate, the boost pressure can be controlled much more accurately compared to pneumatically controlled internal waste gates. Therefore, both internal waste gates always remain in closed position.

Figure 8 shows the engine configuration with EGR. The EGR path, consisting of the EGR valve, EGR cooler and piping, has only been added to the previous configuration.

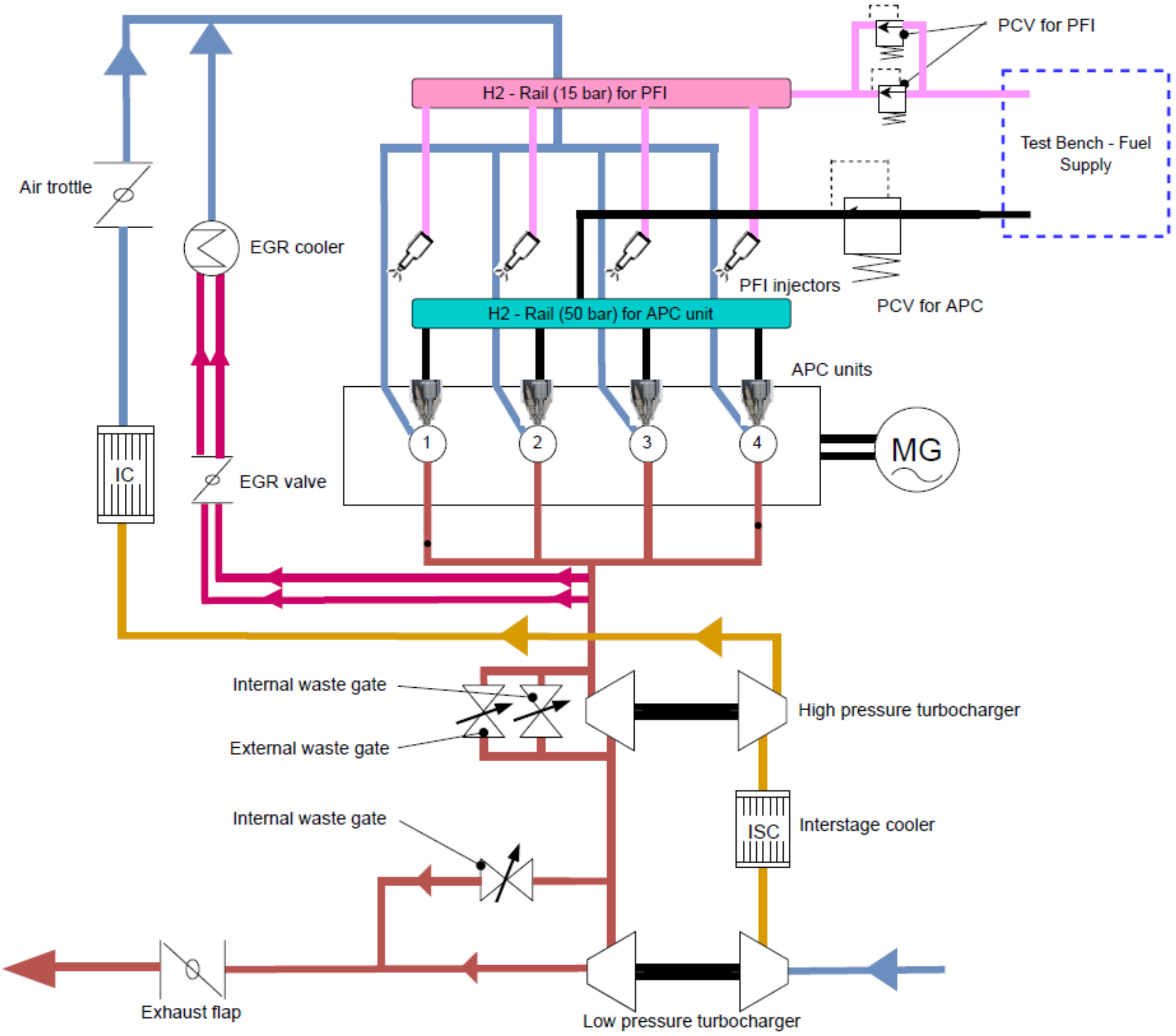


Figure 8: Schematic of the H964-APC engine with integrated EGR

4 Simulation

Two analysis approaches were used to evaluate performance of the three engine variants: 1D and 3D computational fluid dynamics (CFD). First step, the 1D model of the full engine was established using LMS Amesim. This model was developed based on the SI natural gas variant of the D966 engine and was used to modify the initial and boundary conditions for a SI H2 engine variant.

Second step, a predictive CFD model was established using the output from the 1D model: initial state and boundary conditions. Using the software “Converge”, a CFD model was developed. Model used is a Reynolds averaged Navier-Stokes (RANS), specifically developed to simulate jet ignition combustion. First, the model was validated against single cylinder engine results with successful correlation between micro parameters (pre-chamber geometry and pre-chamber fuelling parameters) and macro parameters (engine speed, load, compression ratio, and lambda). More details of the development of this model are provided in [9].

Finally, using GT Power and Amesim, several 1D models were developed to build the behaviour of the hydrogen combustion, specifically knock, correlated to the CFD model. The combustion of the 1D model was then exercised with this correlated behaviour, to both provide higher accuracy initial and boundary conditions for CFD simulations, and to extrapolate the results of the CFD model.

The combined analysis effort provided good insight into active pre-chamber combustion behaviour with hydrogen, an improvement in relative performance with PC and APC over SI and provided the expected efficiency and enleanment potential for the APC engine.

Three operating conditions were simulated. These are listed in Table 4.

Operating point	**N°1**	**N°2**	**N°3**
Speed [rpm]	1900	1000	1900
Fueling quantity [mg/cycle]	72	77	72
Lambda	1.8	2.3	≥ 2.3
EGR [%]	29	7	0
Spark Timing	1° before TDC	variable	variable

Table 4: Operating conditions for simulation

4.1 Simulation results

Figure 9 describes the cylinder pressure of the conventional spark ignition, pre-Chamber (PC) and active pre-chamber for the operating point 1.

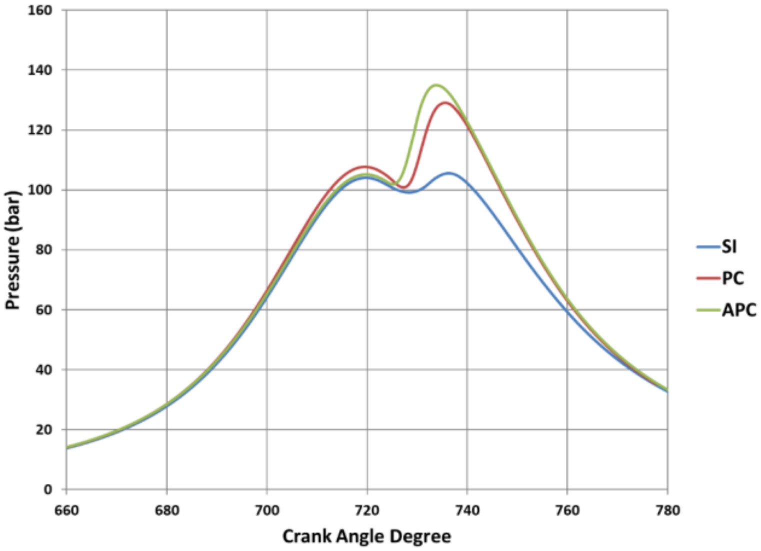


Figure 9: Simulation results of the cylinder pressure for the operating point 1

Figure 10 shows the combustion rate of the three combustion methods. The combustion speed with pre-chamber and active pre-chamber are also higher than the conventional spark ignition combustion.

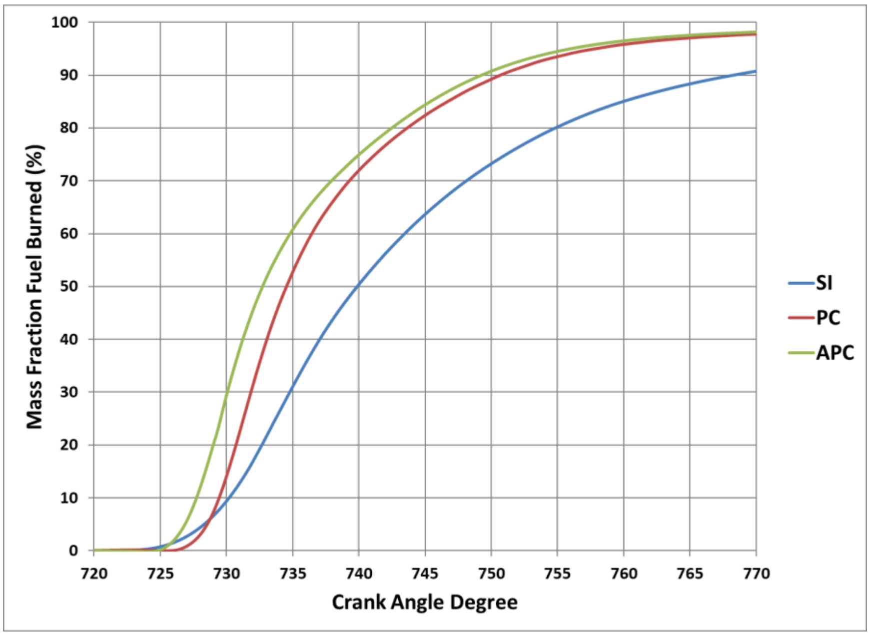


Figure 10: Simulation results of the combustion rate of the three combustion methods

CFD simulations were performed for three ignition system configurations (SI, PC, APC) at a same operating point of 1900 rpm, wide-open-throttle (WOT), λ=1.8, and 29% EGR. Using an ignition timing of 1 °BTDC to ensure that knock was avoided for the three configurations. At this operating condition, successful main chamber combustion was achieved in the simulations. Figure 9 and Figure 10 show the results from these simulations.

As shown in the mass fraction burned (MFB) graphic in figure 10, the higher peak pressure for APC compared to SI and APC is due to faster burn rates. This results in significantly different combustion phases for the three simulations. In comparison between PC and SI, inclusion of the pre-chamber reduced respectively CA0-50 and CA50-90 by 29% and 39%. The further reduction in early and mid-burn time with APC compared to PC can be attributed to a 33% reduction in pre-chamber combustion time (2 degrees crankshaft angle) due to the addition of fuel inside the pre-chamber. Two main sources are responsible for the small reductions in later burn rate segments with APC. The first is due to the difference in the phasing of the combustion. When conditions in the cylinder are less favorable to maintain combustion, the early phase of slower combustion with PC means that the later phase of combustion occurs further away from TDC. The second source of faster late combustion with APC is better jet penetration into the main chamber due to the faster combustion of the pre-chamber [10].
Figure 11 shows, that the models exhibited good agreement for both in-cylinder pressure and knock behavior.

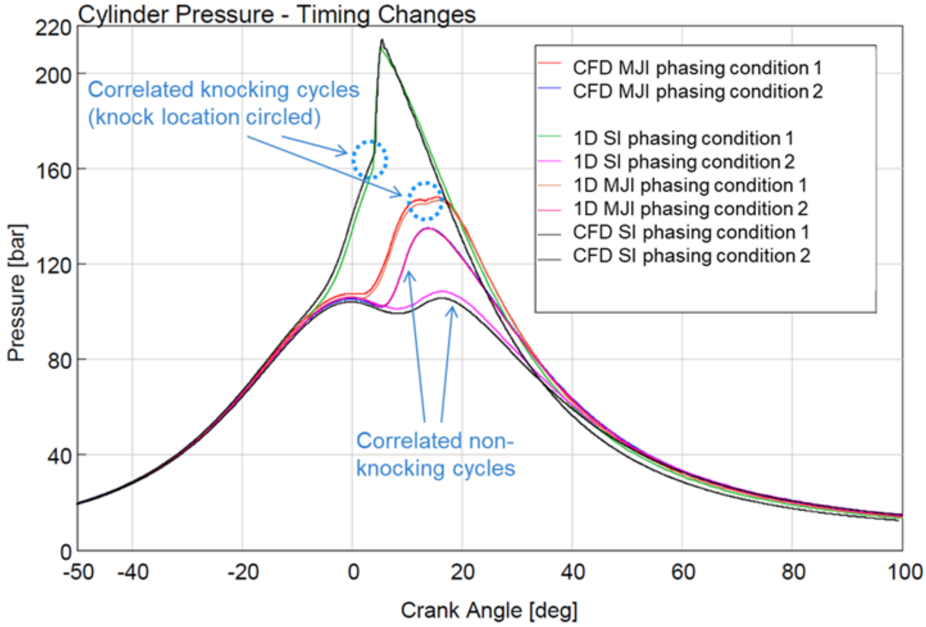


Figure 11: Correlation between CFD and 1D in-cylinder pressure for various engine configurations at Operating Point 1

The comparisons of key performance metrics such as Indicated thermal efficiency (ITE) between APC and SI were masked in CFD simulations by knocking and, for non-knocking cycles, differences in combustion phasing. The ignition timing was scanned in the 1-D model with the correlation to the CFD model, to define the knock-limited CA50 for both ignition configurations. It was defined as the CA50 at which knocking occurred at MFB = 95%. This would be considered a small knock and would have minimum impact on engine performance and mechanical durability.

Table 5 shows the results of the comparison between SI and APC for knock-limited performance from the 1-D combustion model.

	Indicated thermal efficiency [%]	**Knock limited CA50 [°ATDC]**
Spark ignition	42.7	7.5
Active pre chamber	43.8	7.5

Table 5: Comparison of knock limited combustion phasing and predicted ITE for SI and APC

As shown in previous publications, operating the engine with an active pre-chamber configuration improves combustion stability over spark ignition at the same operating point, as shown by a reduction in the coefficient of variation (COV) of IMEP [11,12].

Figure 12 shows a comparison of the flame fronts initiated by the SI and the APC at several points all along the combustion process. During this first phase of combustion, an asymmetric growth of the SI flame front is visible, influenced by the swirl in the diesel style combustion chamber. This behavior can result in a greater cycle-to-cycle variation in burn rates for the SI, which has a big impact on conditions in the cylinder (temperature and pressure) [10]. The unpredictable change in these parameters can be detrimental to hydrogen engines, because they are very sensitive to knocking and pre-ignition.

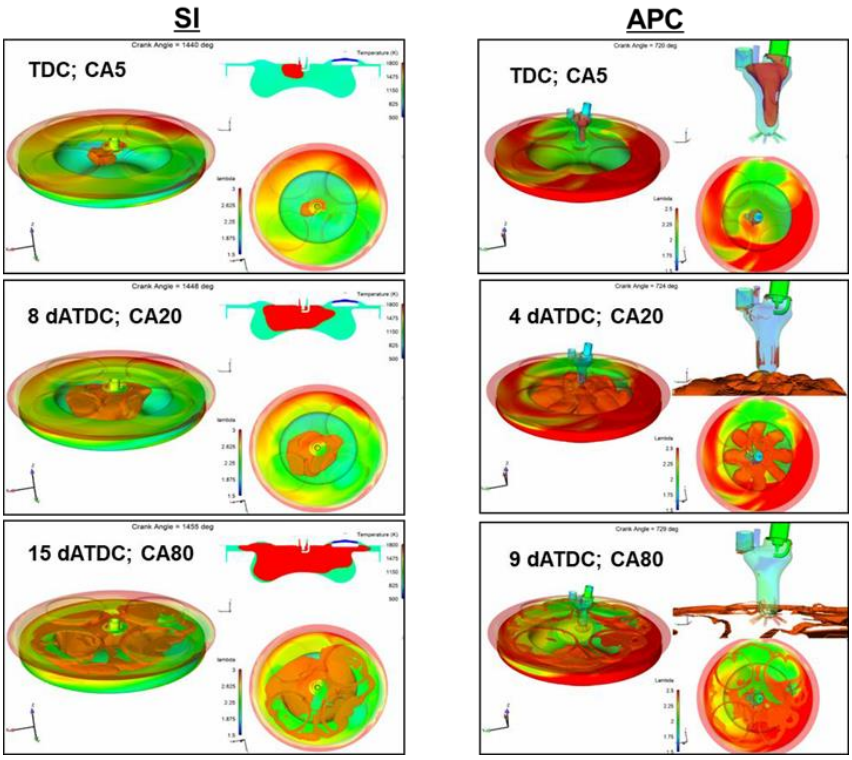


Figure 12: Qualitative comparison of flame propagation behavior between SI and APC at Operating Point 2

Table 6 displays comparative burn duration results for APC and SI with different dilution ratios. In each case, APC produced significantly reductions in CA10-90 burn durations at same conditions. Beyond the λ=2.3 condition, SI engine did not achieve combustion, but APC engine was able to produce rapid burn durations to at least a λ=3 condition, illustrating the lean limit extension capability of APC.

	Units	SI λ = 1.8 EGR: 29%	APC λ = 1.8 EGR: 29%	SI λ = 2.3 EGR: No	APC λ = 2.3 EGR: No	APC λ = 3 EGR: No
CA 0-10	[°CA]	12	10	15.2	8.8	11.6
CA 10-50	[°CA]	10	6	12	4.3	6.6
CA 50-90	[°CA]	29	15	13.2	6.2	14.6
CA 10-90	[°CA]	39	21	25.2	10.5	21.2

Table 6: Comparison of burn duration segments between SI and APC at Operating Points 1-3

The knock limit was not significantly improved by the reduction of the burn rate with active pre-chamber as it does with conventional fuels. Instead, for H2, the most effective way to reduce knock sensitivity is to change the charge composition; either by increasing EGR or lambda.

The results, shown in Figure 13 and Figure 14, that enleanment is the primary combustion-related enabled for increasing the compression ratio in this combustion system. Due to the knocking behavior of hydrogen, a relative increase in compression ratio is expected using enleanment compared to the lambda-compression ratio relationships encountered with gasoline and other fuels.

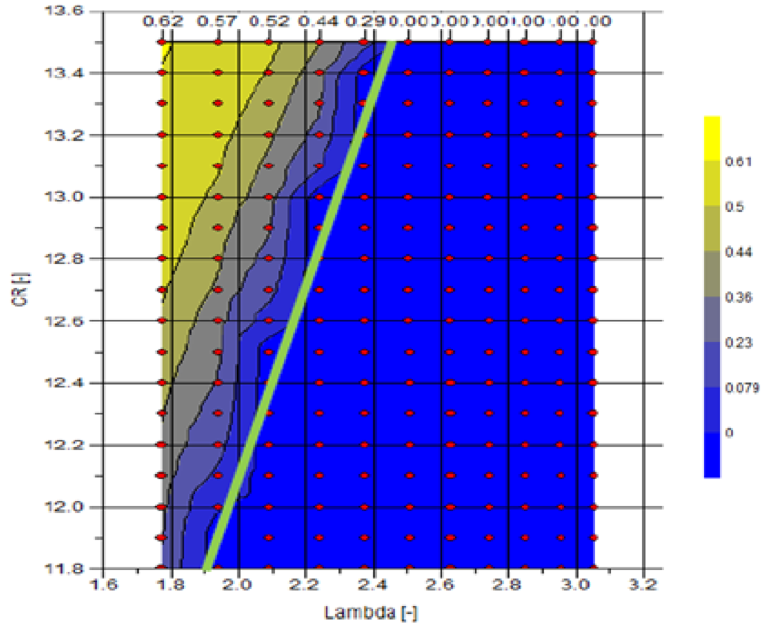


Figure 13 : Predicted knock limited (green line) lambda across varying compression ratios

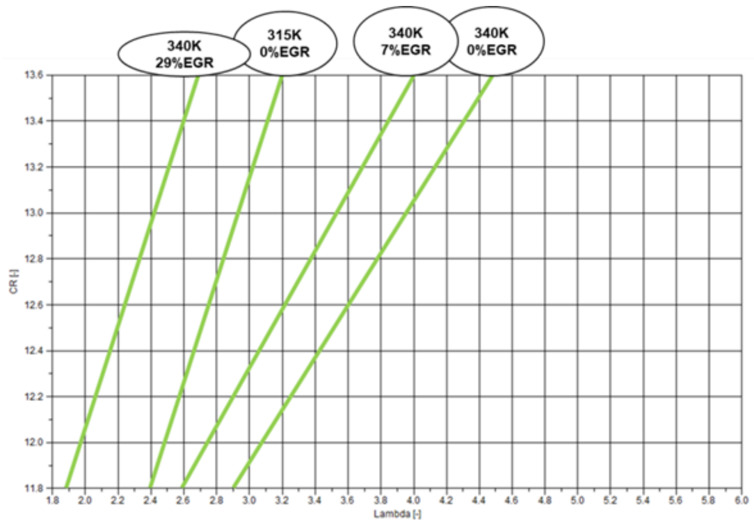


Figure 14: Knock limited (green lines) intake manifold temperature across varying compression ratios with multiple EGR levels

5 Design

To convert the diesel engine into a hydrogen APC engine, several design modifications of the basis engine were necessary:

- Modified cylinder head for APC unit
- PFI fuel path
- APC fuel path
- 2-stage turbocharging air path
- EGR path
- New piston for lower compression ratio

In the following chapters, the main modifications are described more in detail.

5.1 APC engine

Figure 15 shows a CAD model of the complete engine.

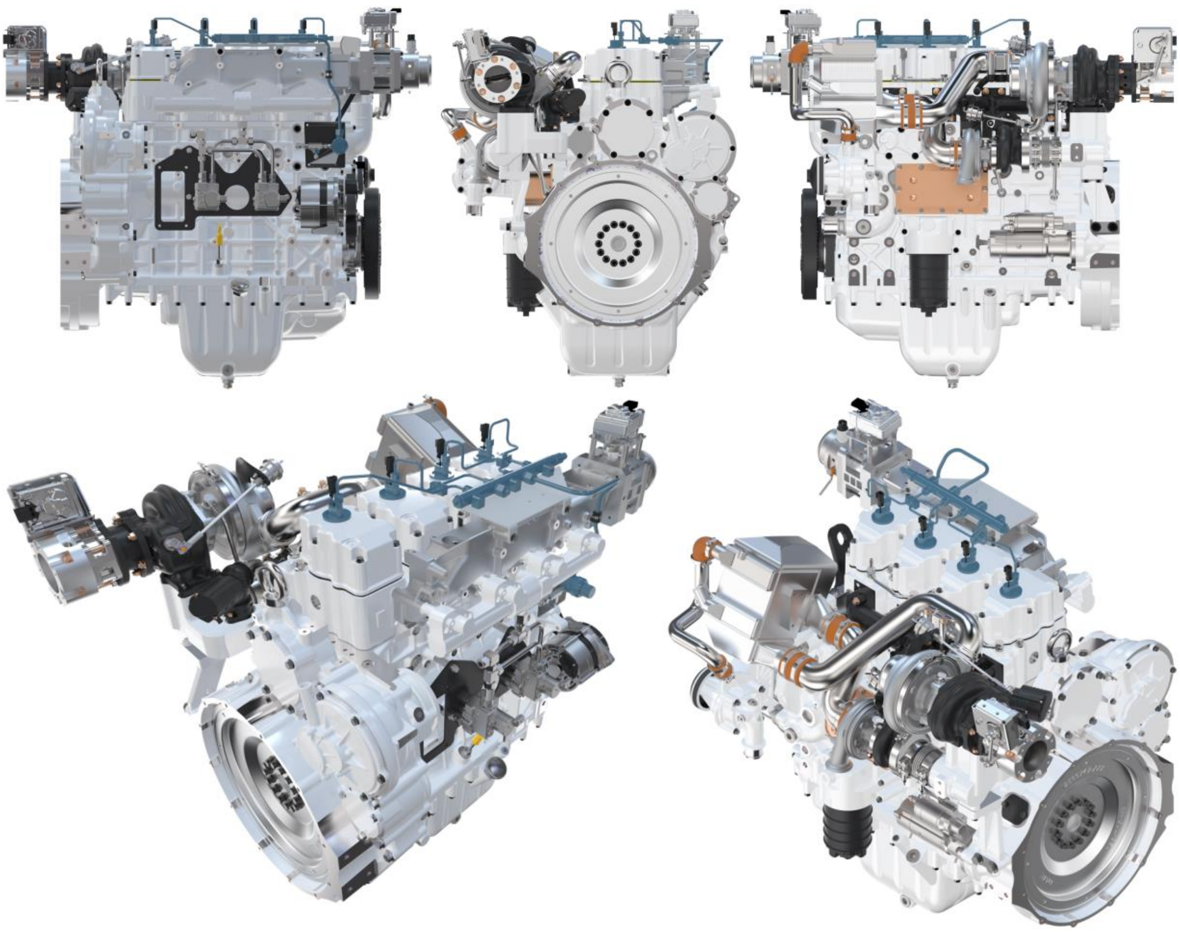


Figure 15: Converted Diesel engine into the hydrogen engine H964 APC

5.2 APC Unit and cylinder head modification

The design layout of the APC unit is constrained by several restrictions. On the one hand, the internal pre-chamber dimensions and the nozzle holes need to be optimized to ensure the best functionality of the APC technology. On the other hand, the injector and the spark plug need to be integrated in the unit as well. This task is challenging due to the limitation of external unit geometry, mainly caused by the cylinder head design.

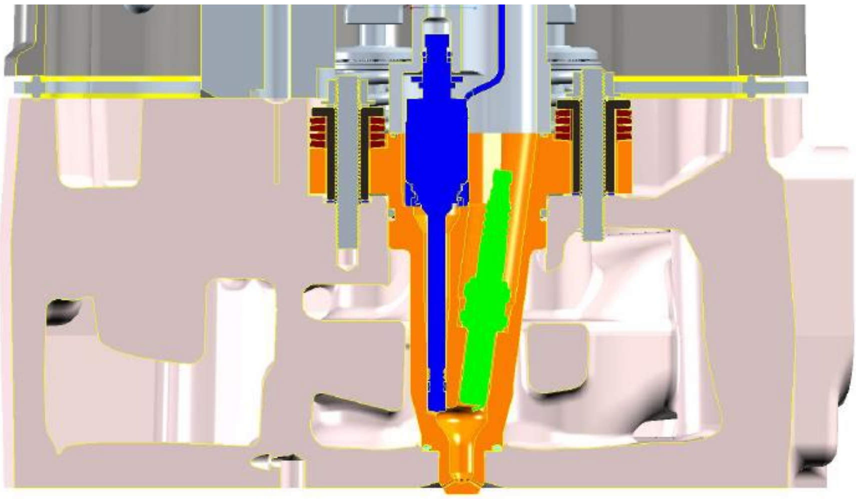



Figure 16 : Integrated APC unit in the cylinder head (left) and APC unit (right)

The original cylinder head of the diesel engine was modified to integrate the APC unit. The APC unit was placed in the same position as the diesel injector. Since the unit requires more space than the diesel injector, material had to be removed from the cylinder head. The air and exhaust gas paths in the cylinder head must not be modified, and a cooling water jacket must be provided for cooling the APC unit to prevent the injector and spark plug from overheating. The position of the valve train shall also not be modified. This resulted in severe limitations for the design of the outer contour of the APC unit. Figure 17 shows the machining of the cylinder head to insert the APC unit on the left, and the APC nozzle when mounted on the right.


Figure 17 : Modified cylinder head (left) and installed APC unit in the cylinder head (right)

5.3 PFI fuel path

Figure 18 shows the PFI path. The H_2 inlet pipe comes directly from the test bench and feeds the pressure control valves (PCV). These electrical valves regulate the pressure in the rail that feeds the PFI injectors and are piloted from the test bench. The rail is basically two aluminum-drilled blocks with a connector to the pipe. The PFI injectors are described below.

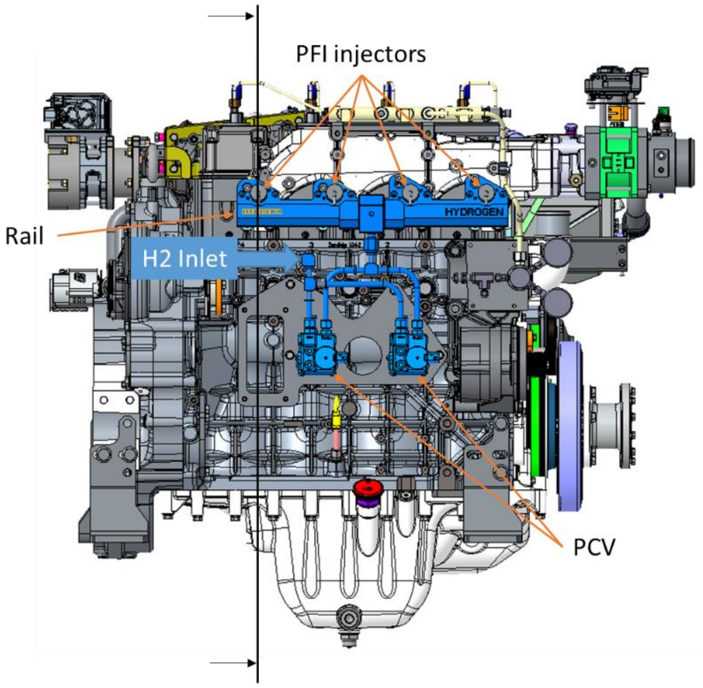


Figure 18: PFI fuel path in blue (left), section of PFI injection (right)

5.3.1 PFI injectors

The PFI "port fuel injector" is symbolically illustrated in Figure 19 with highlighted area 1. This injection transports the main fuel to create the delivered work of the engine. There are several configurations to integrate the injector in the intake manifold or the cylinder head.
Several aspects such as the sealing, the homogeneity of the mixture air / fuel and backfire intensity can be optimized by a correct selection of the injector and orientation in the intake manifold.

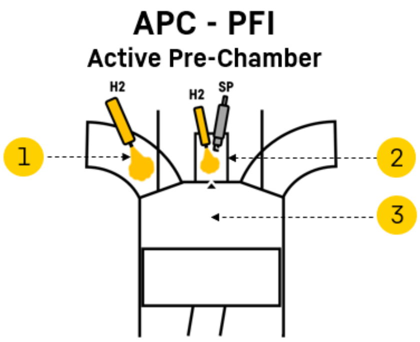


Figure 19: Port fuel injection with active pre-chamber

5.4 APC fuel path

Figure 20 shows the APC fuel path. From the H_2 inlet pipe, the manual pressure regulator ensures a constant pressure in the rail. A pipe screwed from the rail to each APC sleeve. This aluminum sleeve is very convenient to replace the spark plug and the APC injector, since it is screwed with two screws to the APC unit. The APC injector is shown below.

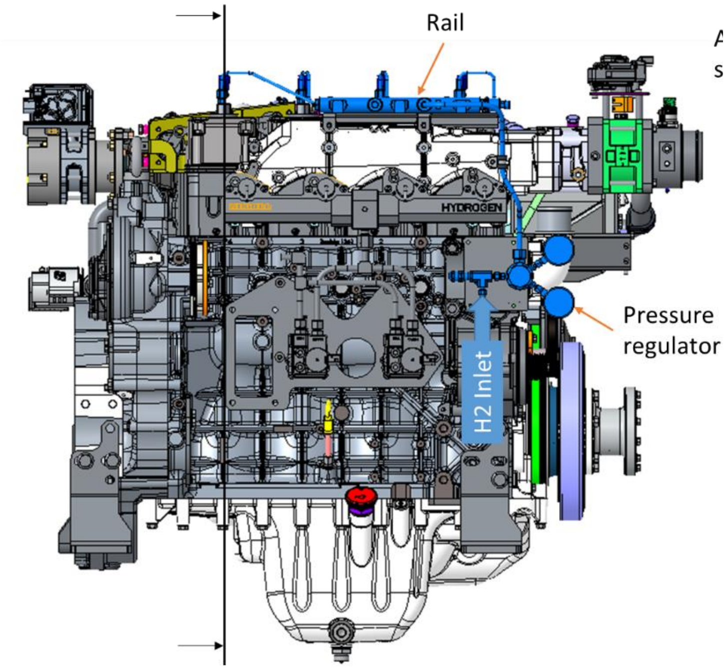


Figure 20: APC fuel path in blue (left), section of APC injection (right)

5.4.1 APC injectors

The PFI "port fuel injector" is symbolically illustrated in Figure 19 with highlighted area 2. This injection has to ensure a precise injection since behavior of the combustion in the prechamber is impacting the efficiency of the main combustion and also the emission characteristics in term of NOx of the engine.

5.5 Two-stage turbocharging system

Two stage configuration allows the engine to run in ultra-lean combustion.

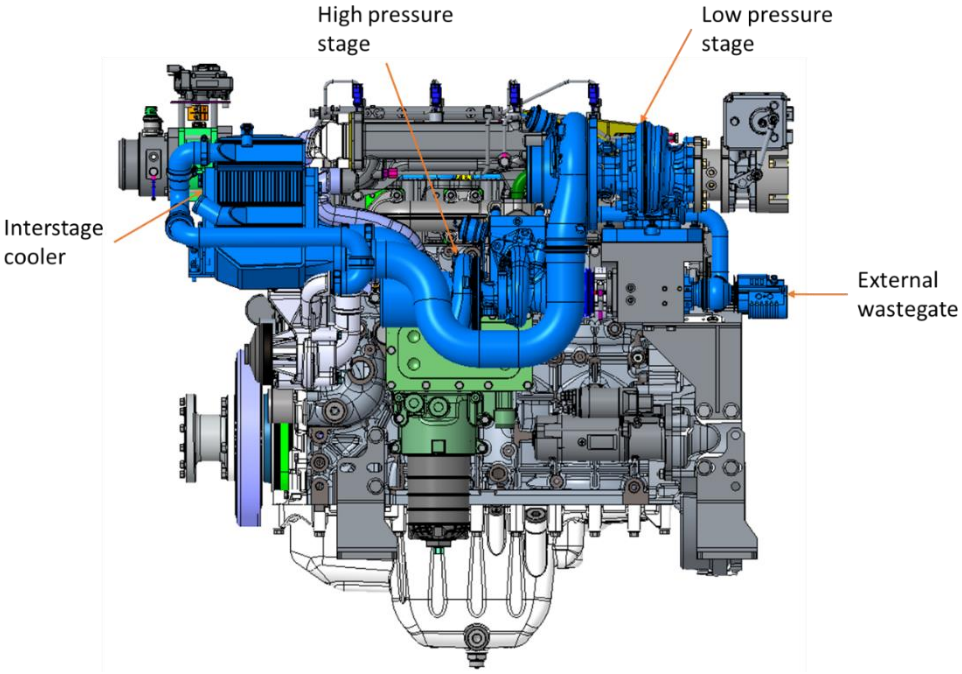


Figure 21: Charged intake air path in blue

This configuration “Two stage turbocharger” is composed of following hardware:

1. Low pressure turbocharger
2. Interstage cooler
3. High pressure turbocharger

5.5.1 Electrical waste gate

To ensure precise control of the boost control an electrical waste-gate has been additionally used. The waste-gate is embedded with power electronic.

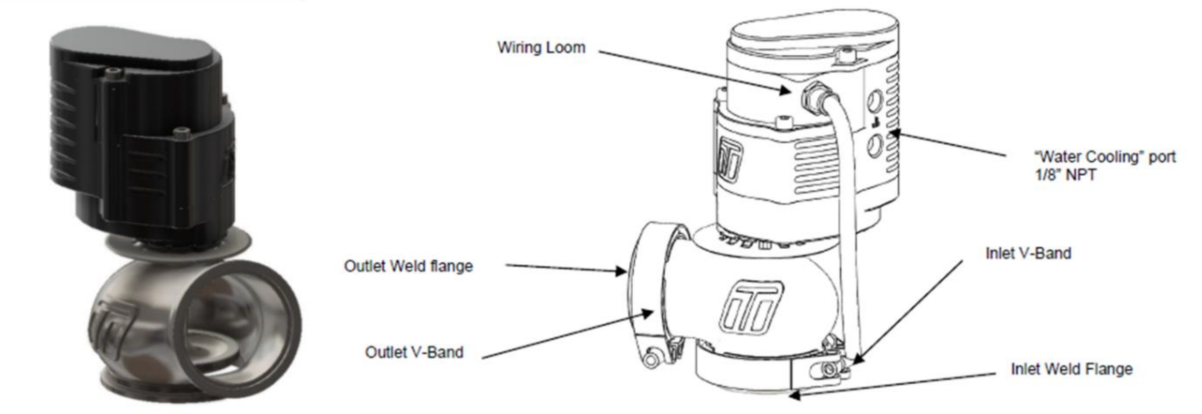


Figure 22: External electrical wastegate

5.6 EGR path

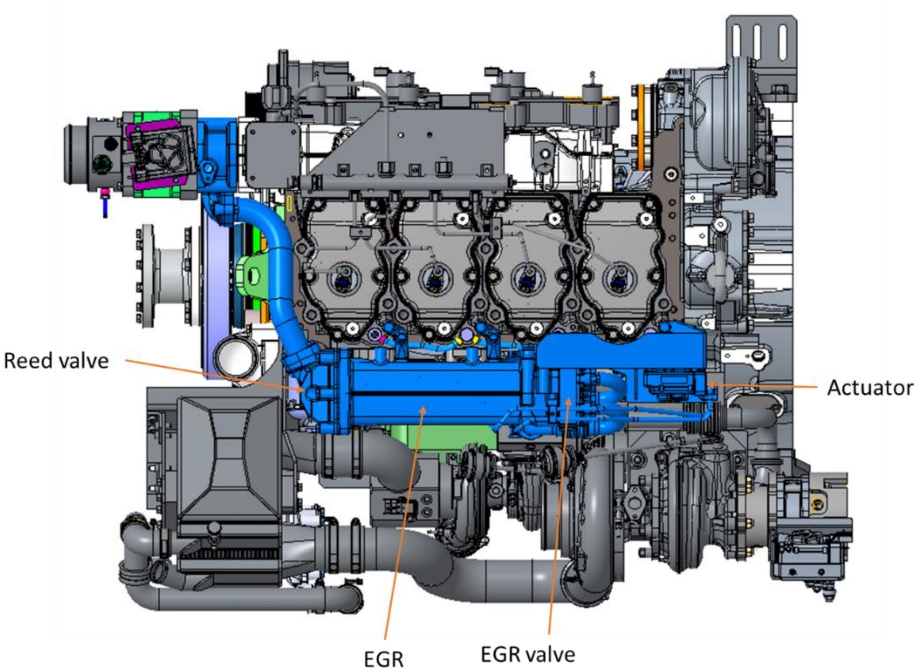


Figure 23: EGR path in blue

5.7 Piston with low compression ratio

Several compression ratio and different material has been investigated with the Hydrogen combustion engine. The piston bowl geometry has been also optimized in order to ensure the *best homogeneity* which leads to lower NOx emission and *better performance* regarding the engine efficiency. Aluminium piston has shown a potential regarding the combustion temperature; however, this technology will have an impact on the engine efficiency which requires further geometry optimization. Current investigations simulation & measurements are based on compression ratio between 10.5 – 13. One of the main benefits of APC is the capability to increase the compression ratio due to lower residuals which could generates anormal combustion and lead to reduction of power density. At the end a best piston configuration has been selected and used for the detailed investigation.

6 Measurements

In order to verify the defined targets (see chapter 1.4) and thus the new APC hydrogen engine, it is necessary to test the engine under several operating conditions and analyse its behaviour. The following subchapters describe these operating conditions and give details regarding the main calibration steps. Furthermore, this chapter discusses the main results and shows the potential of the APC technology and the limits.

6.1 General Information

6.1.1 Engine calibration control

The main actuators that control the APC engine has similar architecture as a gasoline engine control. (Figure 24). The ECU software is divided in several blocks, the Air mass flow control, the Lambda control, the APC control, the Ignition control, and the Exhaust (EGR + exhaust flap) control. The exhaust flap is located after the turbine outlet. Figure 7 also shows the general schematic overview of the engine.

The software always includes two modes: steady state and transient state. The measurements were performed manually at steady state initially to fill the engine mapping. Once all the feedforward maps were complete, it was possible to activate closed loop controls to perform transient steps.

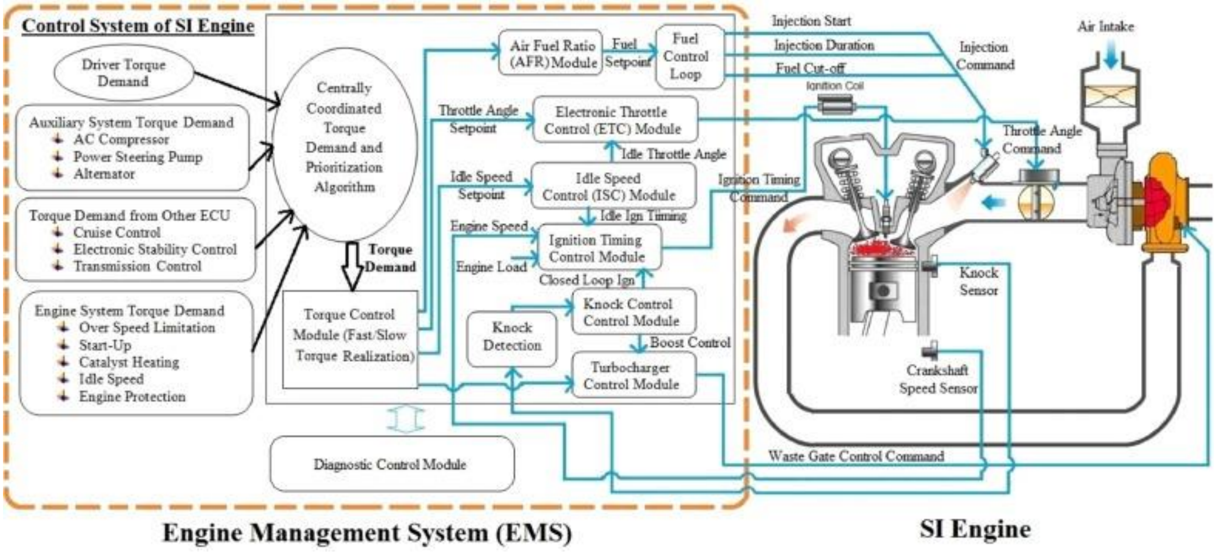


Figure 24: Main calibration parameters for the steady state and transient state [13]

6.1.2 Injection and ignition overview

The working cycle of one cylinder with APC is performed in several steps with typical injection strategy such as the gas engine phases.

In the first step, the PFI injectors start to inject the main quantity into the intake port. The injection window for the PFI injectors is limited by the inlet valves trough the opening and closing timing. The injection pressure varies depending on engine operation point.

In the second step, APC injection occurs which is controlled by end of current (EOC) and time of current (TOC). The timing of APC injection is set in the way to ensure appropriate lambda in the pre-chamber. Previous simulations show a typical range of EOC between 40 – 65°CA before TDC. Too early EOI would enlarge the time for gas exchange between the main and pre-chamber so that the lambda in the pre-chamber would be higher at spark event. Too late EOI would reduce the time for gas exchange and impact the mixing behavior in the pre-chamber. This could lead to knocking due to a rich mixture.

6.2 Steady-state measurements

6.2.1 Introduction

The steady-state measurements will be presented in the next section. The analysis and discussion will concentrate on the following challenges: power output, efficiency, and emissions. As presented in chapter 1.4, the aim is to have a hydrogen-fueled engine with equivalent performance of a diesel engine.

All steady-state measurements are a 30 second averaged measure of the stabilized engine operating conditions. Certain points will not satisfy this constraint, but these points will be further discussed. In chronological order, the engine configurations tested were:

- 1. Reference measurement: Engine configuration PFI + SI
- 2. APC measurement without EGR: Engine configuration PFI + APC
- 3. APC measurement with EGR: Engine configuration PFI + APC + EGR

The measurements started with a PFI+SI configuration, as the APC technology and the hydrogen test bench were new at Liebherr Machines Bulle. As the PFI+SI configuration has already been tested on another engine, it simplified the engine and test bench commissioning. The main hydrogen injection is done by port fuel injection (PFI) and the ignition of the gas mixture in the combustion chamber using spark ignition (SI). To perform spark ignition in the main chamber, special APC dummy without pre-chamber has been designed, where only the spark plug can be integrated (see figure 25).

Figure 25: Prechamber (left) and dummy prechamber for spark plug (right)

APC measurements are divided into measurements with and without EGR. In the configuration without EGR, the EGR path is not only deactivated, but completely removed from the engine to avoid EGR leakage, as the EGR valve is not 100% sealed in closed position.

Table 7 shows the speed ranges of the different engine operating conditions, which all engine configurations need to comply.

	Engine speed range [rpm]
Idling	700
Max torque	1300 – 1500
Max power	1900
Max speed	2100

Table 7 : Engine speed ranges for 4 different operating conditions

6.2.2 Results

Figure 26 shows the percentage of power over Diesel engine that DI-H2 and APC-PFI technologies can achieve. In this case APC-PFI technology reaches 90% of the Diesel power, 10 % higher compared to the DI-H2 technology.

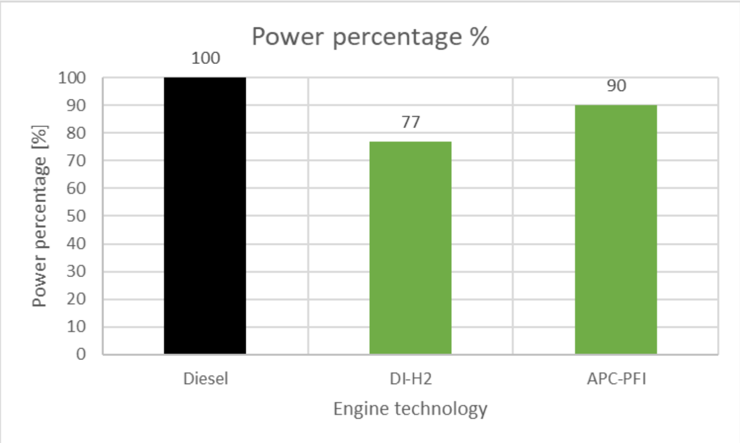


Figure 26: Power percentage comparison

H_2 PFI+APC configurations and the Diesel equivalent engine map are shown on the same figure for a better visual comparison.

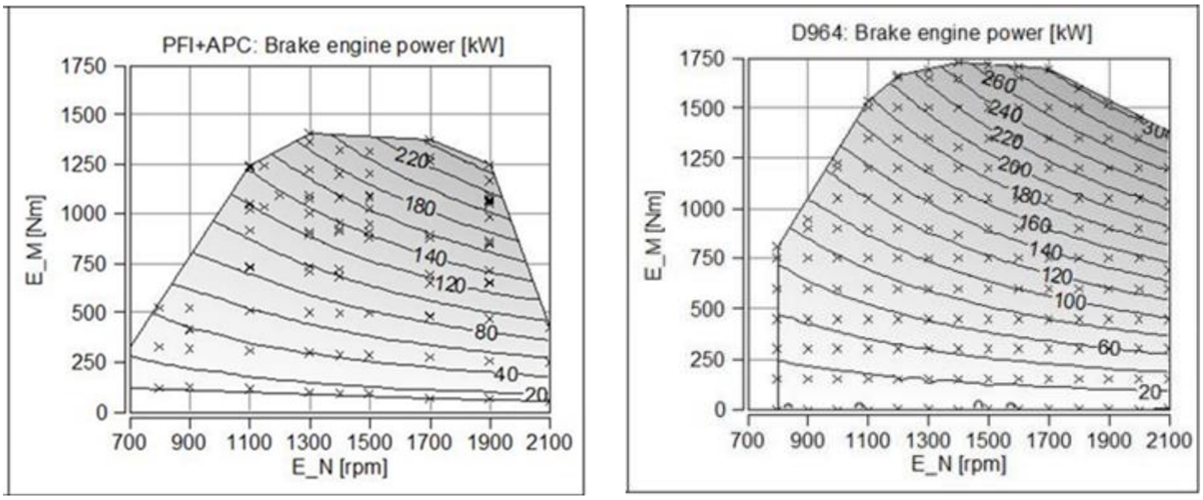


Figure 27: Output engine power

As expected, APC engine power is lower compared to that of Diesel as shown in Figure 27, but with EGR there is the possibility to increase the power as shown in Figure 28. Table 8 summarizes the results.

	Engine Power	Engine torque
Unit	kW	Nm
PFI + APC	250	1406
PFI + APC + EGR	300	1700
Diesel	304	1727

Table 8: Comparison between technologies

The injection in the pre-chamber ensures a Lambda close to stoichiometric in the latter, enabling a very fast and reproducible ignition in the pre-chamber. The flames exiting the pre-chamber are then igniting the chamber mixture homogeneously and rapidly. This advantage results in accurate and reproducible in-cylinder pressures.

At first sight it would seem that the power density of the Hydrogen engine has reached that of the Diesel engine. Indeed, the PFI+APC+EGR configuration reaches a similar power, torque and BMEP limits as the D964 engine. However, the points framed in red (left, figure 28) are not "stabilised" measurements for a long period of time. Therefore, it is reasonable by considering a margin against “abnormal combustion” to say that the PFI+APC+EGR reaches a stabilized power of 250kW-265kW and maximum torque of 1512Nm.

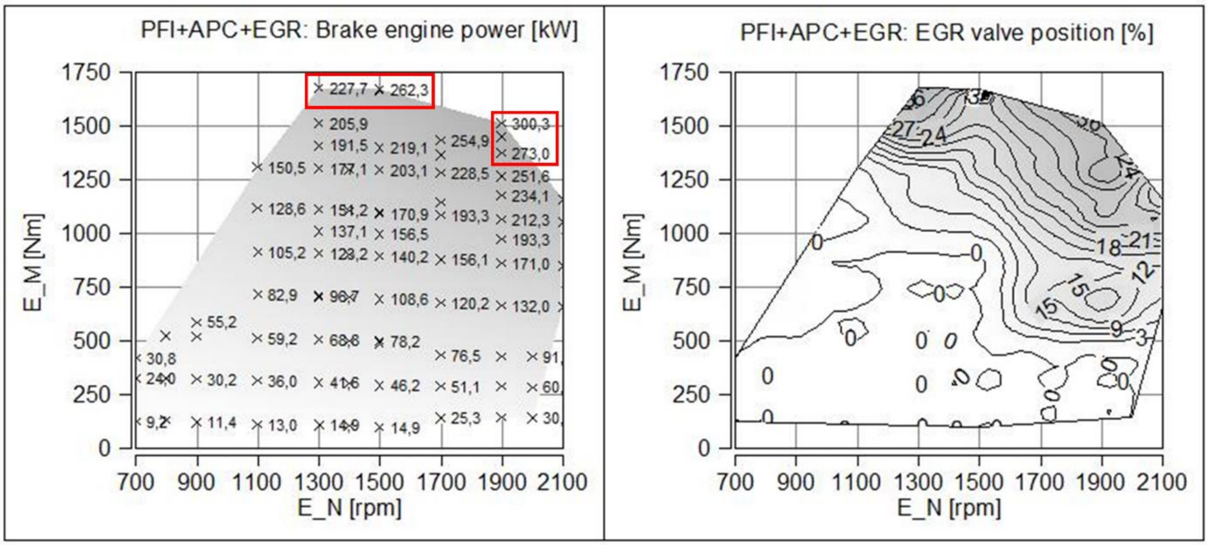


Figure 28: EGR position and operating points to clarify

Backfire was the main issue to increase the engine power. One of the root cause for backfire was valve overlap. The main injection started when the intake valve opened. If the exhaust backpressure is too high, residual hot gases can flow back to the intake port and ignite the injected hydrogen, which is called backfire. Backfire was very sensible to any small disturbance. From boost control, spark plug failure, APC quantity injection, cylinder pressure variation, exhaust temperature and backpressure. At high loads, any small variation quickly induced backfire. Therefore, the APC injectors had to be oiled twice a day (to avoid APC injector leakage or poor injection) due to needle seizure. The spark plugs were replaced relatively often due to quick wear and flashovers. The external wastegate was calibrated with a map of PID values. Thanks to a precise calibration, the wastegate was very precise during the closed loop boost control. If the external wastegate opened or closed to quickly, backfire would probably arise at high loads.

Adding EGR enabled a stable operation up to 250kW, and short periods at 300kW. Figure 28 shows that the EGR valve opens at high loads. Although the EGR main use is at high loads, it also had an impact on the lower loads. Indeed, the EGR path had a leakage, which allowed a much more stable engine operation up to 200kW. EGR gases are inert, therefore cooling down the in-cylinder temperature of combustion and by consequence the exhaust gas temperature. In addition, the EGR path reduces the exhaust backpressure since the exhaust gases have another “bypass” to flow. Both these effects helped reduce backfire at mid-load and full load because of lower igniting temperature or less backflow from the exhaust valve to the intake valve.

To resume, APC technology (combustion stability), EGR and properly functioning components allowed the port fuel Hydrogen injection engine to achieve for short periods the power and torque output of the equivalent Diesel engine. Another component was necessary to achieve these high powers and will be discussed in the next paragraphs.

The second necessary factor to increase power was the double stage turbocharger. Hydrogen has a high energy density related to the fuel mass, but a very low volumetric density compared to liquid injected diesel. Plus, hydrogen is port-fuel injected for this engine, which reduces the volumetric efficiency even more. Therefore, the double stage turbocharger provides enough boost pressure to avoid being limited by the air path. The maximum relative boost pressure achieved for the PFI+APC+EGR was over 4 bar compared to conventional boost pressure of diesel engines 2 bar - 3 bar.

The Lambda variation is depending on the load level. The goal was to have a Lambda range as large as possible for two reasons.

Firstly, one main goal of the APC technology was to achieve ultra-low NOx emissions to avoid installing an exhaust after-treatment system on the machine. The NOx limit is defined by transient driving cycles with defined emission and performance limits. For example, the NRTC (Non-Road Transient Cycle), which has been chosen to test the APC engine, currently limits the NOx emission at 0.4 g/kWh. The test will be specified at chapter 6.4 NRTC. NOx has different formation mechanisms, but the most prominent factor for the oxygen and nitrogen to react is high temperature (Zeldovich, thermal NOx formation [14]). Therefore, while operating in a lean mixture, the flame temperatures will be lower, hence lower NOx emissions. The steady state raw measurement of NOx emission shows almost in all engine Maps values below 0.4 g/kWh, which is already a good indicator that the NOx emissions for different vehicle application are low. However, it is not possible to draw conclusions with the transient cycle since most NOx are emitted during dynamic load steps and dynamic Lambda variations. This will be further treated in the following NRTC chapters.

Secondly, dynamic response will be better if the lower loads are calibrated with a high Lambda. The engine will be able to “jump” from a high to a lower Lambda setpoint quickly. The injection delay is very small compared to the air path “lag”. Therefore, during a load step it will be possible to inject a large quantity of hydrogen to gain torque rapidly. The load step topic will be further discussed in the Load Step chapter 6.3.

To summarize, the double stage turbocharger allows the engine to reach the maximum loads and run with a high Lambda in the overall map, which consequently leads to low NOx emissions. In counterpart, the double stage requires a more complex software control and more hardware, such as an additional turbo obviously, an additional wastegate and an additional inter-stage cooler.

Since the overall CA50 was set around 8° CA and 11° CA, theoretically the maximum engine efficiency should be reached. Here are some considerations that could further improve efficiency.

The principal limiting factor for efficiency increase is the port fuel injection. The volumetric efficiency for port fuel injection is estimated between 72-77% and direct injection is 85-95%. Therefore, switching to DI would increase overall efficiency.

The APC injected quantity at 1100 rpm and 1240 Nm is 1.2% of the total injected quantity. The APC quantity was modified but has never improved the engine efficiency. So, it seems that the quality of the “flames” is more important for the engine efficiency than the injected quantity in the pre-chamber.

The engine efficiency with this technology is enhanced by the increased combustion speed due to jet ignition. Currently the values around 42 %, however a large potential for increasing the efficiency is possible by introduction of several parameters’ optimisation of engine.

Both compressors’ efficiencies were also analysed to verify if an optimization was possible at the 40.7% eff. operating point. The “low” pressure compressor was estimated at an efficiency of 75% and the “high” pressure compressor at 80%. The inter-stage cooler was cooling down the air at 85°C, and could be optimized, but the HP compressor is already at maximum efficiency. Therefore, an engine efficiency gain seems unlikely with the air path.

6.3 Load steps

6.3.1 Introduction

Load step measurements are useful for the characterization of the engine dynamic, which is very important in heavy-duty applications. It shows how fast the engine torque is able to follow and reach the torque set point. For example, an excavator which needs a sudden increase in torque when the excavator starts to dig. Alternatively, as a dumper needs to accelerate abruptly when loaded to reach its unloading point quickly, to be as efficient as possible all day long. In this measurement campaign, several load steps have been performed with the PFI+APC configuration (without EGR) and the results compared with an equivalent diesel engine D964.

Some of the calibration parameters are specifically modified to carry out load steps. Since the engine is in a dynamic mode, the parameters need a “dynamic correction” directly implemented in the software. These parameters are lambda control, start of ignition (SOI), position of the external wastegate (both internal wastegates are always closed) and intake air flap. Dynamic lambda correction enables the lambda control to reduce to a lower set point (defined by user) while the torque set point is not reached. Dynamic ignition correction will delay ignition timing while torque is increasing. The wastegate and the throttle will respectively close and open by a factor of the torque set point difference of the measured torque.

A last point to add deals with the emissions that the engine emits during the load step. We were more focused on NOx, but there are also other emission values that need to be within limits (CO_2, etc…). Here the NOx measure has a delay of ~5 seconds due to the time lag of the exhaust gases to reach the FTIR analysis bay. A NOx measurement is presented in the following results for informative purposes. These load steps were performed to get as close as possible to the dynamics of the diesel engine. There is no optimization for NOx emissions. It should be noted, however, that by focusing on the NOx measurement, it was possible to define some limits to improve the NRTC cycle results (cycle including dynamic and pollution constraints) which will be explained in the next chapter.

6.3.2 Results

The boundary conditions of the load step were the following:

- 1300 rpm (constant)
- From 15% to 100% of the “alpha” (simulated pedal) set point

6.3.3 Analysis

The major factor in the load step slope is the Lambda dynamic correction. The results have been evaluated with three step loads. For the time delay comparison, the reference has been set at 1170Nm (90% of 1300Nm). Below is the time to reach this reference torque for all curves:

- Diesel: ~ 3 s
- APC (Lambda min. 1.4): 3.2 s (Without optimisation and Post injection)
- APC (Lambda min. 1.7): 3.6 s (Without optimisation and Post injection)

As Lambda is reduced, the slope of the load step increases. In other words, the lower the lambda value, the quicker the engine reaches the desired torque. It seems logical that with a lower Lambda value and an equal air quantity entering the engine, more torque will be produced since more fuel is injected per stroke. With this in mind, all there is left to do is to “control” this Lambda step as precise as possible. By varying the constants of the feedback loop of the Lambda control, it was possible not to undershoot the Lambda set point (which was critical). It was also relevant that the Lambda value stayed constant as long as possible. This Lambda control assured a constant torque slope during the load step.

For a Diesel engine Lambda is limited by the “smoke“ limit. For the hydrogen engine, knocking was the limiting factor. Indeed, it was necessary to introduce dynamic ignition angle to lower Lambda to values below 2. This method “adjusting dynamically the ignition delay” allowed to avoid knocking, however it has a negative effect on efficiency.

Initially, the double stage turbocharger was dimensioned to provide enough boost pressure for the APC engine. Therefore, with a second optimization loop, the LP and HP turbochargers could provide better dynamic response time. The performed measurement shows that the time to set point of the relative boost pressure is almost equal for each load step. All the wastegates (internal and external) are closed to not waste any energy from the exhaust gases. The external e-wastegate only opens when the boost pressure reaches the boost set point. Since the lower lambda value is limited by knocking, the air path is currently limiting the time to set point. Hence, the second optimization loop of the LP and HP turbochargers.

Last but not least, the time to set point could be improved by adding post-injection. It has been shown on another in-house H_2 engine that the added post-injection energy boosts the turbines, thus increasing the compressors speed. New design of APC “APC Only” using a single injector for ignition and main fuel transport has been also designed to guarantee the post injection which helps for transient and dynamic improvement.

6.4 NRTC – Non Road Transient Cycle

6.4.1 Introduction

The NRTC (Non-Road Transient Cycle) is a standard cycle used to test the performance and emissions of an off-road engine. In fact, this test is mandatory to obtain the certification that allows a company to sell the engine. In addition, each continent/country has different emission limits according to the regulations of the continent/country. Furthermore, the emission regulation varies for the engine category (power output) and the emission stage (law implementation year). The regulation chosen for the APC engine is the following: Norm EPA 40 CFR 1039 Tier 4 NRTC. This cycle was specifically chosen to demonstrate the engine dynamics and the low NOx emissions, that are limited to 0.4 [g/kWh] for the latter. Finally, the NRTC cycle is defined as both a cold and a warm start. However, the hydrogen engine only performed the warm cycle since the maps were not calibrated for a cold engine.

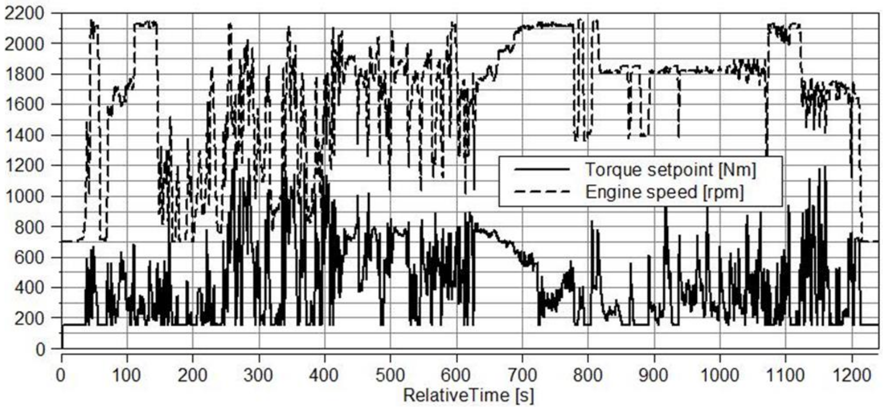


Figure 29: NRTC warm cycle

Figure 29 shows the NRTC performed by the APC engine. The normalized cycle is originally a percentage of engine torque and engine speed. Therefore, it has to be denormalized for each specific engine map. The map used for the APC engine is shown in figure 30. The engine was in PFI+APC configuration, since EGR was installed later the latter. The max. power of 220kW started at 1500 rpm until 2100 rpm. This max. power was chosen because the engine could run without considerable backfire up to this power. Unfortunately, the cycle cannot be compared to the equivalent Diesel engine since the max. power and engine map are different. Figure 30 shows the scatter of the engine operating points as well during the NRTC cycle. The engine mostly operates at mid-load and high speed. Finally, for this measurement campaign, the engine is speed driven by the electric motor. Therefore, the engine is calibrated to achieve the torque set point.

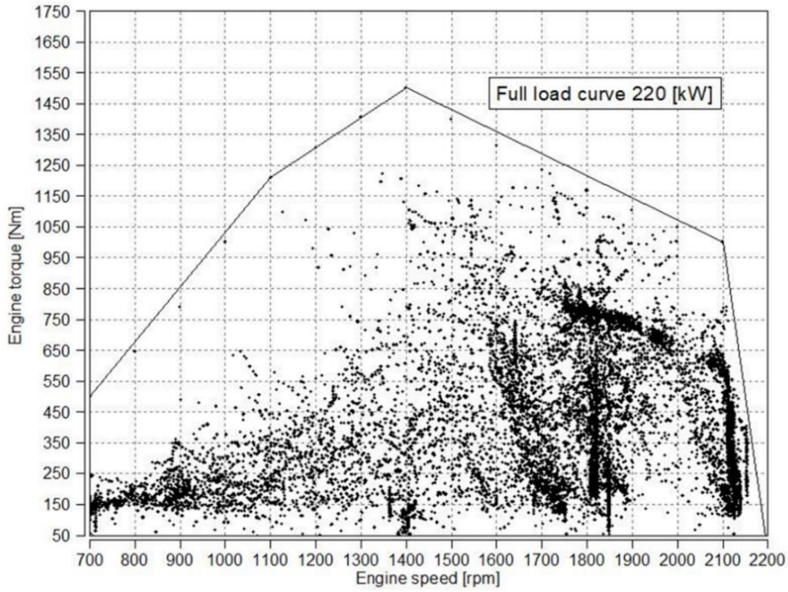
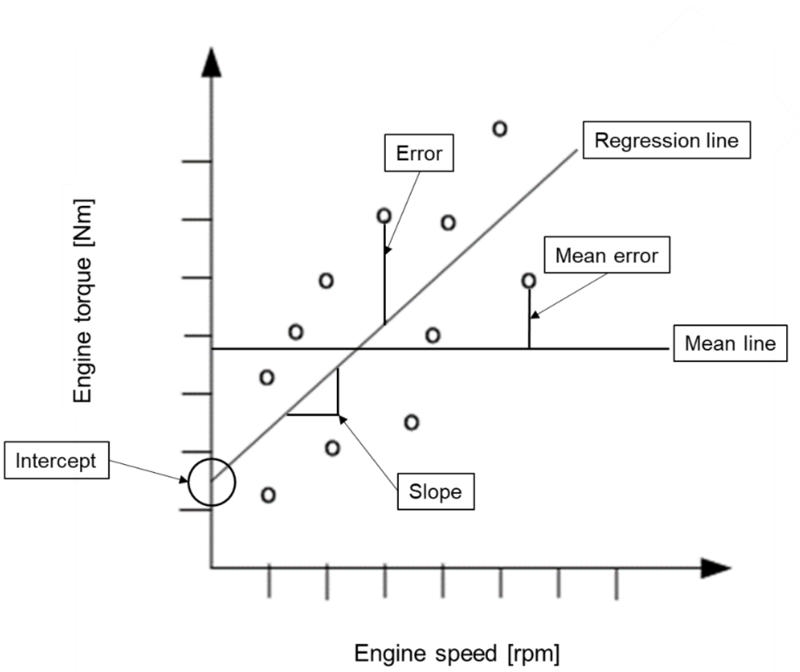


Figure 30: Operating points during the NRTC cycle

Before analyzing the results, a final subject must be discussed. Many parameters must be verified to validate the NRTC cycle. The parameters considered for the APC measurements were the speed, torque, and power accuracy. To do so the following statistical tools were used: Slope, Intercept, SEE (standard error of estimate) and R2 (R squared) (see Figure 31). The results of the four tools must be between the lower and upper limits. The latter statistical tools are calculated as follows: all brake torque points from the cycle are plotted on a torque versus speed graph and generate a regression line. The slope (slope of the line) and intercept (intersection of the regression line and the y-axis) are straightforward results from the regression line (ax+b). The standard error of estimate (SEE) is, in a simplified way, the “mean” error between the measured points and the regression line. The R squared (R2) will determine how far the measures are from the regression line, with a result from 0 to 1. For example, if all points are on the line, R2 will be equal to 1.



$$\text{SEE} = \sqrt{\frac{\sum(Errors)^2}{(number\ of\ measures) - 2}}$$

$$\text{R2} = \frac{\sum Mean\ errors^2 - \sum Error^2}{\sum Mean\ errors^2}$$

Figure 31: Statistical concepts to validate the NRTC cycle

To summarize, the engine speed, brake torque and brake power had to be valid to ensure the engine's performance. Then, the goal for the engine-out raw NOx total average emissions, measured from the FTIR bay, is to be under 0.4 [g/kWh].

6.4.2 Results with transient cycles

Several NRTC cycle measurements have been performed, knocking or backfire are the main parameters that have been optimized using hardware and software update. As written above, the goal was to have all parameters OK under the condition of NOx emissions values below 0.4 [g/kWh]. Several loops of optimization have been carried out regarding the NOx results. The table below shows the different steps of optimization with the corresponding NOx obtained values.

In the following table NOK means Not OK.

Cycle number	1	2	3	4	5	6	7	8	9	10	11	12
NOx emission [g/kWh]	1.33	1.41	0.64	0.65	0.47	0.58	0.52	0.41	0.46	0.41	0.37	0.24
Number of NOK parameters	3	2	5	4	5	5	6	5	5	4	6	1

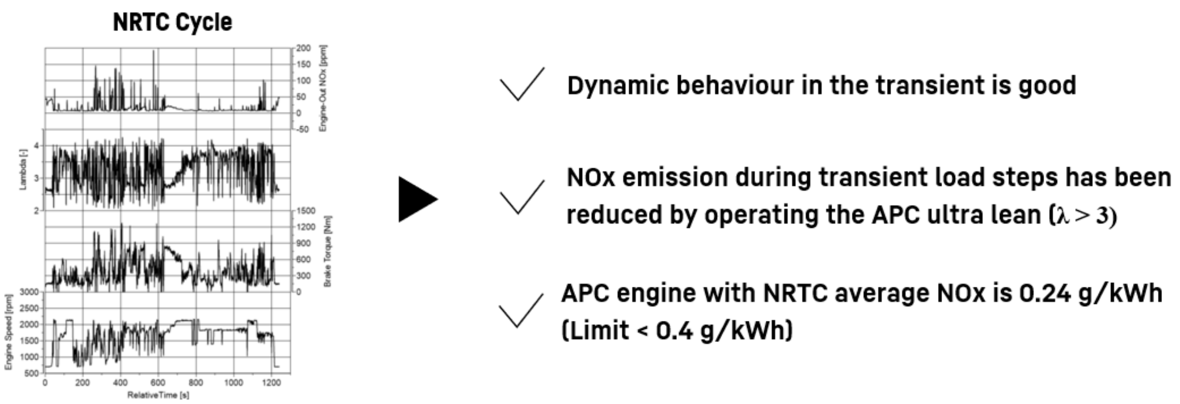
Table 9: All completed NRTC results

Figure 32 shows which parameters are OK. Engine speed (E_N) is obviously all OK since speed is piloted precisely with the electrical motor. Therefore, the torque (E_M) was the failing factor of the cycle, including the power (E_P_B) since power is torque multiplied by speed.

The main result of NRTC cycle has been shown in the figure below:

				NOx: 0.65 [g/kWh]		NOx: 0.41 [g/kWh]		NOx :0.29 [g/kWh]	
Name	unit	Lower Limit	Upper Limit	Value	Validation	Value	Validation	Value	Validation
E_N_Slope	rpm	0,95000	1,0300	0,99300	OK	0,99300	OK	0,99300	OK
E_N_Intercept	rpm	-70,000	70,000	11,315	OK	11,406	OK	11,413	OK
E_N_SEE	rpm		104,23	55,372	OK	54,463	OK	54,631	OK
E_N_R2	-	0,97000		0,98200	OK	0,98200	OK	0,98200	OK
E_M_Slope	Nm	0,83000	1,0300	0,83500	OK	0,85700	OK	0,87300	OK

Figure 32: Results overview of three different cycles



This result shows that with an optimal strategy defined with best hardware and software configuration, we can achieve the NOx emission below the current limit regarding the stage V regulation in Off-Road sector.

7 Conclusions and outlook

Conclusions

Transformation of a diesel engine to hydrogen combustion engine has been investigated in these studies using the active pre-chamber technology (APC). Hydrogen will be one of the main vectors for decarbonization of the power train and especially in Off-Road sector. One of the main limitations of the hydrogen combustion engine is the power density which is strongly dependent on the injection strategy and limitation due to knocking in the combustion chamber. For example, port fuel injection is limited in volumetric efficiency and suffers from combustion instability.

A specific combustion model with hydrogen has been created. Several measurements and CFD simulations were performed later on to validate the 1D Model performed in AMESim platform.

This 1D Model has been used to identify the main components of the engine and to define the control strategy.

Several phases in the measurement campaign have been used to evaluate the technology in the test bench. Steady state measurement has been performed in order to build the engine Maps characteristics and create a base for the calibration for transient operation. In the early phase we observed that the

NOx emissions are very low and demonstrate that the technology is very promising in emission and dynamic. This Behavior has been confirmed later in the real transient cycle NRTC which demonstrates a NOx emission below the current stage V emissions regulation (< 0.4 g/kW.h) Several loops of optimization has been applied in order to avoid the anormal combustion and improve the power density of the engine. At the end we achieved to develop an engine which has nearly the same power density as diesel and can demonstrate the equivalent dynamic of the current diesel engine by keeping the NOx emissions in very low interval.

Outlook

Technology of APC has shown a large potential regarding its application in hydrogen combustion engine. This technology offers the possibility of keeping the same engine size for replacement of the current diesel engine for different applications in Off-Road sector such as wheel loader, excavator and generally the earth moving vehicles. Several optimized variants of APC are currently under investigation in order to set up the final layout of serial development.

The effort to build the optical infrastructure is too high and measurements will not be performed on the single-cylinder engine. However, a simplified installation (e.g., an optimal spark plug) is being studied and will be considered in the future for basic optical investigations.

This application of APC can be extended to other alternative fuel technology such as ammonia combustion engine and methanol combustion engine in order to improve the ignition behavior of such engines.

8 National and international cooperation

Cooperation with FHNW regarding the subject:

Review of Optical Measurement Techniques and its Requirements on Optical Access to the Engine Cylinder with Focus on Hydrogen Combustion in Internal Combustion Engines

The complete document has been integrated into the annexes below.

9 Publications

- M. Bunce (Mahle Power train), B. Seba, Simulation investigation with active pre-chamber for hydrogen combustion engine, International congress on combustion engine, Baden Baden March, 2021

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