Hydrogen as a promising Engine Fuel Alternative

Prof. Dr.-Ing. **Hermann Rottengruber**, IEPS-OVGU, Magdeburg;
M.Sc. **Aristidis Dafis**, IEPS-OVGU, Magdeburg
M. Sc. **Dmitrij Wintergoller**, IEPS-OVGU, Magdeburg
B. Sc. **Bastian Rabe**, IEPS-OVGU, Magdeburg

Abstract

The reduction or complete elimination of CO_2 emissions and other pollutants in the transport sector is a central issue for both industry and environmental policy. To ensure compliance with legislation for emissions [1], alternative fuels are playing an increasingly important role in reducing exhaust emissions. One possible method to rapidly, sustainably and cost-efficiently decimate emissions in the transport sector is a modern internal combustion engine that runs on hydrogen (H_2). Hydrogen, as a carbon-free fuel, represents a meaningful alternative to conventional fuels and can be used as a key element in overcoming the challenges of the energy transition. Furthermore, hydrogen can be expected to make a remarkable contribution to the ideology of zero emission and the sustainable decarbonization of the transport sector.

The injection of hydrogen into the combustion engine is the same as with the conventional fuel. A distinction is made between two basic processes. The fuel can be injected either through Multi-Point-Injection (MPI) into the intake, manifold or directly into the combustion chamber (DI: Direct-Injection). The behaviour of hydrogen as a fuel in the combustion engine is far more complicated than that of conventional fuels. Due to the high ignitability of hydrogen, higher demands are placed on the ignition system and the combustion chamber design. In MPI with external mixture formation, backfires can occur as combustion anomalies caused by local hotspot formation. These backfires must be prevented. Backfiring is characterized by the ignition of the fuel-air mixture back into the intake duct. Furthermore, due to its lower density, the injected hydrogen displaces the air in the intake manifold. This is caused by the strong expansion of the hydrogen, after it has been injected into the intake manifold. As a result, the volumetric efficiency of the combustion engine is reduced.

Furthermore, performance advantages can be achieved with hydrogen in comparison to petrol engine combustion. The specific power is increased by 15 % with stoichiometric combustion using cryogenic hydrogen and external mixture formation, and by 17 % with direct injection of hydrogen [2, 3]. In contrast to MPI, the filling loss is avoided with H_2 direct injection and internal mixture formation. To exploit the full efficiency potential, it is therefore advantageous to inject hydrogen directly into the engine combustion chamber. These results in decisive advantages

in efficiency for vehicles with high performance requirements compared to driving with fuel cells or with the conventional combustion engine [4].

1. Hydrogen as fuel

When comparing the physical properties of hydrogen with those of conventional fuels, several advantages and disadvantages arise for the operation of a hydrogen combustion engine. In general, hydrogen is the chemical element with the lowest density. At ambient pressure, its density is 70.8 kg/m³ in the liquid state and 0.09 kg/m³ in the gaseous state [3, 6]. In contrast, the density of liquid diesel at ambient pressure ranges between 820 and 845 kg/m³ [6]. In hydrogen engines using intake port injection, hydrogen expands to the intake manifold pressure level after injection. This expansion is accompanied by a volumetric increase of the gas, which displaces fresh air and therefore negatively affects the cylinder filling.

To avoid these filling losses, it is advisable to inject the hydrogen fuel directly into the combustion chamber. This causes the fuel to expand within the cylinder, which increases the air charge. As a result, both torque and engine power increase.

The laminar flame speed of hydrogen is approximately 230 cm/s. When compared to conventional fuels, which exhibit laminar flame speeds around 40 cm/s [6] hydrogen burns significantly faster. This property leads to extremely short combustion durations. The short combustion duration enables energy conversion under conditions close to an ideal constant-volume process, resulting in a high thermal efficiency. Furthermore, it promotes stable combustion, especially under lean fuel-air mixture conditions [3].

In addition to high thermal efficiency, the high flame speed also leads to increased peak pressures and temperatures in the combustion chamber. This, in turn, can result in increased noise emissions, NO_x formation, and wall heat losses [3]. he rapid combustion is also related to the high ignitability of hydrogen, resulting in a wide ignition range of $0.13 < \lambda < 10$ [3, 6]. This allows for lean-burn operation across the full engine map under lambda control.

Another major advantage of using hydrogen as a fuel—besides the wide ignition limits—is its carbon-free nature. Due to the absence of carbon atoms, the combustion of H_2 theoretically produces no CO_2, CO, or unburned hydrocarbons (HC). In real-world combustion, however, small amounts of carbon-containing exhaust emissions may occur. These emissions are primarily due to lubricating oil entering the combustion chamber and participating in the combustion process. Nevertheless, such emissions are typically at the detection limit of current conventional CO_2 measurement systems and are therefore considered negligible [7].

The only relevant pollutant emissions from hydrogen combustion are nitrogen oxides (NO_x). Especially high NO_x emissions can occur under slightly lean conditions, at very high combustion temperatures, and with long residence times in the so-called post-flame zone of the combustion chamber. To ensure low-emission operation of a hydrogen engine, it is therefore recommended to operate with a fuel-air ratio of λ > 2.2 [3]. At this air excess level, the combustion temperature in lean operation stays below the NO_x formation threshold of approximately 2200 K [3].

A fundamental issue in hydrogen combustion engines is the susceptibility to combustion anomalies. The most well-known anomaly is knocking. The tendency of a fuel to knock is described by the methane number (MN). Methane, by definition, has a methane number of 100 and is thus highly knock-resistant. Hydrogen, in contrast, has a methane number of 0. The main reason for this is its extremely low minimum ignition energy of 0.017 mJ [3].

Due to this low ignition energy, hydrogen is prone to pre-ignition and backfiring. Backfiring can occur in engines with external mixture formation, where the air-fuel mixture ignites on hot surfaces (hotspots) while the intake valves are still open, causing combustion to propagate back into the intake manifold. This can be avoided by employing direct hydrogen injection. However, the phenomenon of pre-ignition remains an issue. Pre-ignition is the spontaneous ignition of the fuel-air mixture by hotspots in the combustion chamber, independent of the spark timing. Pre-ignition is classified into knock-induced and non-knock-induced pre-ignition. In the knock-induced case, the mixture ignites after the intended ignition point; in the non-knock-induced case, ignition occurs before it. Both lead to rapid pressure rises and high temperatures, which can result in mechanical damage and engine overheating. Therefore, during combustion chamber design, potential hotspots as ignition sources must be carefully identified and eliminated.

In addition to its low ignition energy, hydrogen has a relatively high autoignition temperature of 585 °C, compared to 250 °C for diesel fuel. Therefore, achieving a stable autoignition in a hydrogen engine is only possible with a high compression ratio and, in some cases, preheating of the intake air [3].

The thermal efficiency η_th depends on the compression ratio ε and the isentropic exponent κ:

$$\eta_{th} = 1 - \frac{1}{\varepsilon^{\kappa-1}} \quad (1)$$

$$\text{with } \kappa = \frac{c_p}{c_v} \quad (2)$$

The isentropic exponent is a temperature- and pressure-dependent parameter and describes the dimensionless ratio of the specific heat capacities at constant pressure (c_p) and constant volume (c_v). At ambient conditions, the isentropic exponent for a hydrogen-air mixture is approximately κ ≈ 1.4, which is higher than for a gasoline-air mixture, typically around κ ≈ 1.35 under stoichiometric conditions. A higher κ results in increased thermodynamic efficiency, as shown in Figure 1. Therefore, hydrogen offers an efficiency advantage due to its favorable isentropic properties.

It is important to note that in lean-burn operation, excess air is present in the combustion chamber. This surplus air can absorb heat without participating in the combustion process, thereby reducing the combustion temperature. As the combustion temperature decreases, the isentropic exponent κ increases, which in turn enhances thermodynamic efficiency during lean operation.

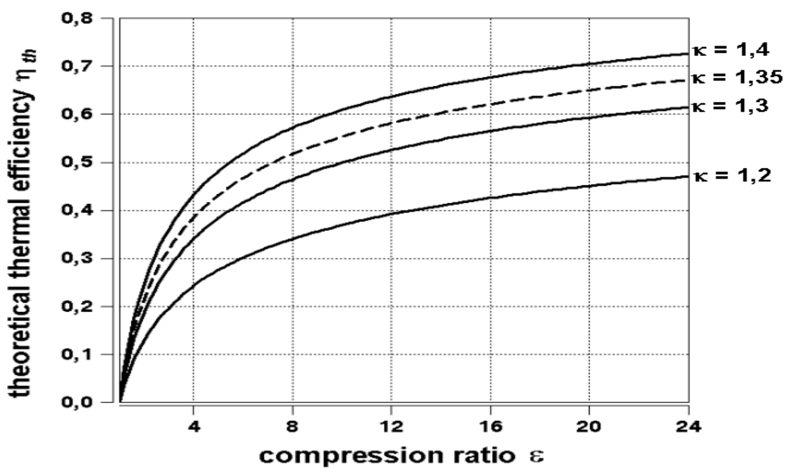
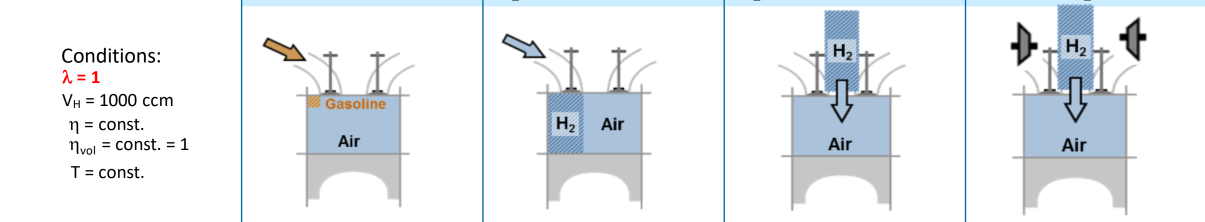


Figure 1 Thermal efficiency as a function of the compression ratio and the isentropic exponent

2. Engine Concepts

s previously mentioned, hydrogen can be injected into the engine either via port fuel injection (PFI) or direct injection (DI), similar to conventional fuels. Depending on the injection strategy, different advantages in specific power output can be achieved. In an H_2-PFI engine with external mixture formation, the specific power is reduced by 18%, see Figure 2. However, when hydrogen is injected into the intake manifold in cryogenic (deep-cooled) form, the specific power increases by 15% [3]. With internal mixture formation using H_2 direct injection, specific power is increased by 17% compared to conventional port fuel injection in spark-ignition engines.

In addition to port and direct hydrogen injection, a hydrogen combustion engine can also be equipped with a turbocharger, enabling even higher peak power outputs. In this case, specific power increases of over 100% can be achieved (see Figure 2). Similar to gasoline or diesel engines, various boosting systems can be applied. Significant benefits across the entire engine speed range can also be achieved by using a variable geometry turbocharger (VGT) [8]. As mentioned in Chapter 1, DI is preferred due to reduced combustion anomalies and higher thermal efficiency [9].



	Gasoline PFI	H_2-PFI	H_2-DI	Charged H_2-DI
Fuel Volume [ml]	17	296	420	420
Air Volume [ml]	983	704	1000	1000
Mixture Calorific Value [MJ/m³]	3.9	3.2	4.5	up to 7.8 @ Π ≈ 1.8
Power output [%] (compered to gasoline n/a)	100	82	117	up to 200

Figure 2 Performance enhancement of various engine concepts

In stoichiometric operation with hydrogen port injection and spark ignition, backfiring may occur, as discussed earlier. When hydrogen is injected directly into the combustion chamber, knock tendency increases due to the high in-cylinder pressures and temperatures during stoichiometric combustion. The use of a turbocharger further increases in-cylinder temperatures. To counteract these high temperatures, lean mixtures can be applied. For example, with λ ≈ 2 and port injection, only a low mean effective pressure (IMEP) is achieved. In contrast, higher IMEP values can be reached with DI and spark ignition. With additional boosting, both IMEP and engine power can be further increased. Moreover, exhaust gas recirculation (EGR) can be employed to lower the in-cylinder temperature.

Besides DI with spark ignition via a spark plug, autoignition can also be used to initiate combustion. In this case, ignition is triggered by compression and the resulting increase in internal energy. This ignition concept brings both advantages and disadvantages compared to spark ignition. In general, a hydrogen compression ignition engine with short combustion durations and ignition delays is feasible. It allows for a higher efficiency compared to other ignition concepts, mainly due to reduced compression losses. Additionally, high power density under knock-free conditions can be achieved.

However, disadvantages include high thermal loads, since short combustion durations result in higher peak temperatures. Moreover, autoignition is associated with strong cyclic variations, caused by the inconsistent ignition timing of the fuel-air mixture from cycle to cycle.

Another drawback is the high NO_x emissions. This is attributed to both the elevated peak temperatures and the inhomogeneous mixture distribution in the combustion chamber. In compression ignition, hydrogen is injected shortly before top dead center (TDC) — typically at 5° to 15° crank angle (CA) before TDC for passenger cars and 6° to 12° CA before TDC for commercial vehicles [10] This does not allow sufficient time for proper air-fuel mixing, resulting in a locally rich mixture, even though the global mixture may be lean. This leads to the formation of diffusion flames in the post-flame zone, which strongly promote NO_x formation. To perform injection close to TDC, high injection pressures of around 300 bar are required. These are currently under development in hydrogen injector research.

With DI and spark ignition, the hydrogen is injected earlier during the compression stroke. This reduces the required injection pressure and enables the formation of a homogeneous air-fuel mixture. As a result, both global and local lambda values remain lean, which significantly reduces NO_x emissions. Furthermore, by adjusting the start of injection (SOI) according to engine load, NO_x emissions can be further reduced, as shown in Figure 3 [3]. Consequently, lean-burn operation with spark ignition currently represents the optimum solution.

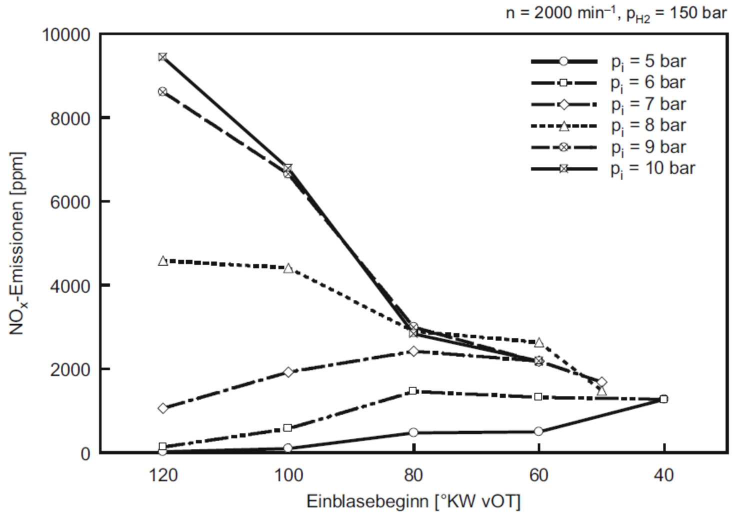


Figure 3 Adaptation of the start of injection to the NOx emission behavior with H2 direct injection [3]

3. Component Requirements

The components in hydrogen-fueled combustion engines that come into contact with the fuel must be specifically adapted to the unique material properties of hydrogen. These include hydrogen embrittlement, low lubricity, and high diffusivity. Hydrogen, in its atomic form, can penetrate the material structure and diffuse through most metals. Moreover, hydrogen dissociates at the material surface, leading to deformations in the microstructure caused by local stress concentrations, which in turn result in material embrittlement.

Suitable materials for hydrogen internal combustion engines and fuel-contacting components include austenitic steels, aluminum alloys, nickel-based alloys, selected polymers, and ceramics [11]Due to hydrogen's poor lubricating properties, adequate materials and designs are also necessary for hydrogen injectors and valves. Accordingly, both PFI (Port Fuel Injection) and DI (Direct Injection) hydrogen injectors, as well as the valve assemblies, must be designed to support self-lubrication.Because of the lack of lubrication, as well as the increased chemical and thermal stresses, armored valve seats are used. In addition to these seats, modified valve guides can reduce hydrogen diffusion by improving lubrication in the valve stem guide area. Further thermal load reduction at the valve head can be achieved by modifying the valve stem geometry.

This is particularly relevant for the exhaust valve, which is subjected to high thermal loads. To withstand high temperatures, sodium-filled hollow valves are used. The sodium inside the valve stem transfers heat from the valve head toward the valve guide. This significantly reduces the valve head temperature and allows the valve material to be used beyond its conventional temperature limits (see Figure 4) [1]. In addition, valve timing must be adapted for hydrogen operation to ensure efficient gas exchange and optimized cylinder filling.

Apart from the valves, the spark plug must also be adapted. A cold-type spark plug with a low heat rating number is required. The reason is its small heat-absorbing surface area, especially at the insulator foot, which minimizes heat uptake. The short thermal path enables fast heat dissipation. Another significant advantage is the high resistance to pre-ignition and hot corrosion. Furthermore, the use of a blocking diode in the spark plug design prevents the presence of residual electric charge in the system, which could otherwise trigger unintended pre-ignition.

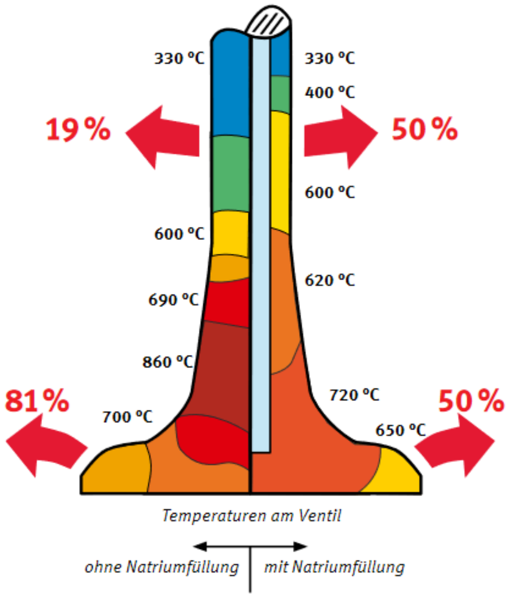


Figure 4 Component temperatures on the valve with/without sodium filling [1]

To counteract the increased thermal loads in the combustion chamber, additional component measures are required, including cooled piston rings and an improved oil cooling system. These help optimize heat removal using oil as a heat transfer medium. In addition to cooling, piston rings must be adapted to the lack of lubrication due to the absence of carbon in both fuel and exhaust gas. As a result, scuffing and piston seizure can be reduced or even prevented by optimizing the ring shape (barrel profile) and surface finish of both the piston rings and piston.

Another key component requirement in hydrogen combustion engines is the use of a hydrogen pressure regulator. This is necessary because hydrogen has very low energy density under ambient conditions. Therefore, hydrogen is typically stored at 300–700 bar to increase the vehicle's energy storage density and driving range. The high-pressure hydrogen is then regulated to the injector supply pressure via a hydrogen pressure reducer, located between the storage tank and the injection system. In the context of turbocharged hydrogen combustion engines, the turbocharging system must also be adapted. The exhaust turbocharger uses the exhaust gas enthalpy to increase boost pressure and thereby improve engine efficiency via turbine expansion and compressor compression.

The available exhaust enthalpy depends on the mass flow, pressure, and temperature of the exhaust gases. Due to hydrogen's low density and rapid combustion, both the exhaust mass flow and temperature are comparatively low. As a result, the exhaust gas enthalpy available from hydrogen combustion is significantly lower than with conventional fuels (seeFigure 5).

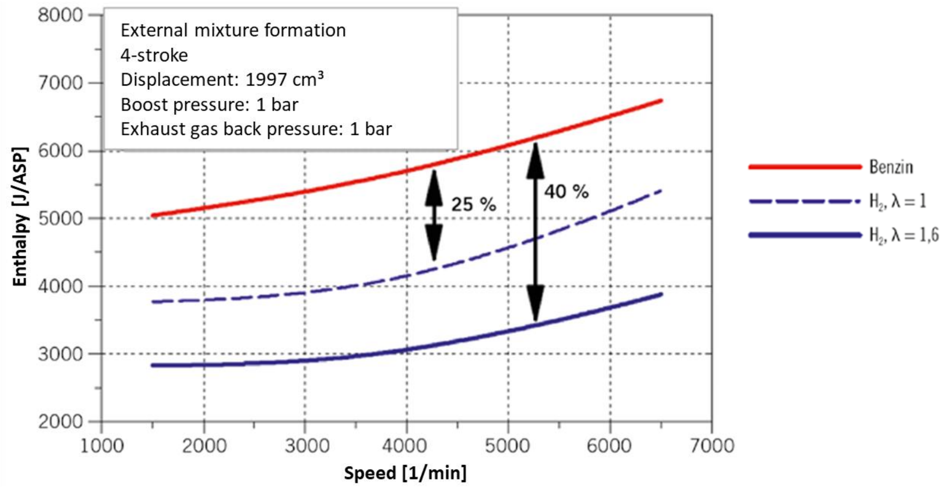


Figure 5 Comparison of the exhaust gas enthalpy of gasoline, stoichiometric and lean H2 combustion [12]

4. Emission Reduction / Decarbonization

To achieve a nearly emission-free hydrogen combustion engine, the entire energy value chain must be considered. This is addressed through the "Well-to-Wheel" analysis, which also includes the production and provision of the propulsion energy. Hydrogen as a fuel is categorized into six different production pathways, commonly referred to as the "Colors of Hydrogen". The most well-known types are grey, blue, and green hydrogen:

Grey hydrogen is produced from fossil fuels such as coal or natural gas, emitting carbon monoxide (CO) and carbon dioxide (CO_2). The most common process is steam methane reforming (SMR). Blue hydrogen is produced in the same way as grey hydrogen, but the CO_2 emissions generated during production are captured and stored, either underground or underwater (carbon capture and storage, CCS). Only green hydrogen is produced without CO_2 emissions, provided that renewable energy sources are used. In this case, hydrogen is produced via electrolysis, using renewable electricity and water.

Figure 6 shows a schematic representation of the hydrogen types mentioned above. In theory, the combustion of hydrogen in an internal combustion engine produces no carbon-containing emissions. The only significant emissions are nitrogen oxides (NO_X), which are formed due to the high combustion temperatures. In real-world engine operation, however, oil combustion contributes to particulate matter and carbon-containing emissions.

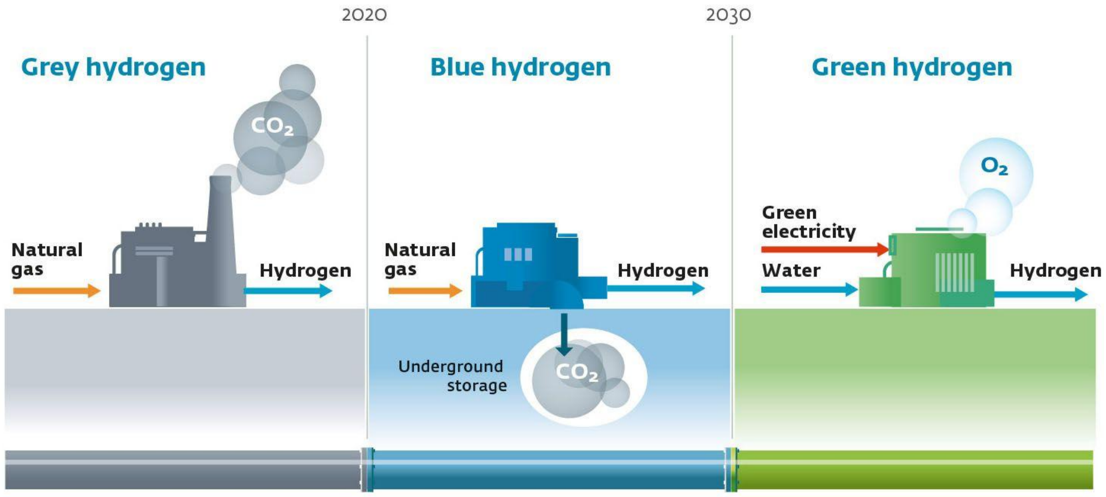


Figure 6 Production and emissions of gray, blue and green hydrogen [13]

Carbon-based emissions in a hydrogen combustion engine can be generated through several mechanisms:

One major contributor is the presence of so-called "blow-by gases". These occur during engine operation when combustion gases leak past the piston rings into the crankcase due to imperfect sealing. In the crankcase, the blow-by gas mixes with the existing oil mist. Using a closed crankcase ventilation system, the gas-oil mixture is filtered and recirculated into the intake air. The challenge lies in filtering out microscopic oil droplets from this mixture. As a result, traces of oil mist enter the intake air and burn in the combustion process, leading to carbon-based emissions. Improving the crankcase ventilation system, specifically the filtration of oil microdroplets, can help reduce these emissions. Additionally, oil leakage from the valve stem seals into the combustion chamber is another source of carbon emissions.

Another source is the oil film on the cylinder wall, which partially burns and emits CO_2. In bivalent vehicles (running on both gasoline and hydrogen), the gasoline tank ventilation system becomes another emissions source. In this system, evaporated gasoline is routed through an activated carbon canister and then into the intake manifold. Even when operating on hydrogen, the lambda control system must adapt to this fuel vapor to maintain stoichiometric combustion. This gasoline vapor becomes particularly relevant when hydrogen is used exclusively, as CO_2 emissions can still be generated due to tank ventilation. In contrast, dedicated hydrogen engines eliminate this source of emissions entirely.

To minimize combustion-generated carbon emissions, the use of low-ash lubricating oil is recommended. This helps reduce both particulate matter and carbon emissions. Additionally, a close-coupled oxidation catalyst can be integrated into the exhaust system to ensure compliance with strict emissions standards such as EU7 and SULEV.

To reduce NO_x emissions, a Selective Catalytic Reduction (SCR) system can be used. This system requires ammonia (NH_3) for the reduction reactions, typically provided by injecting AdBlue (urea solution, CH_4N_2O). To prevent toxic excess NH_3 from being released into the environment, an Ammonia Slip Catalyst (ASC) is installed downstream.

Research is currently exploring an alternative called the H_2-NSCR (Non-Selective Catalytic Reduction with H_2) process, which eliminates the need for ammonia. In this approach, hydrogen is injected directly upstream of the catalyst, creating a reducing atmosphere that allows the catalytic reduction of NO_x in the exhaust gas. Figure 7 illustrates the relationship between raw NO_x emissions and the air-fuel equivalence ratio (λ), along with symbolic representations of the SCR and NSCR exhaust aftertreatment systems.

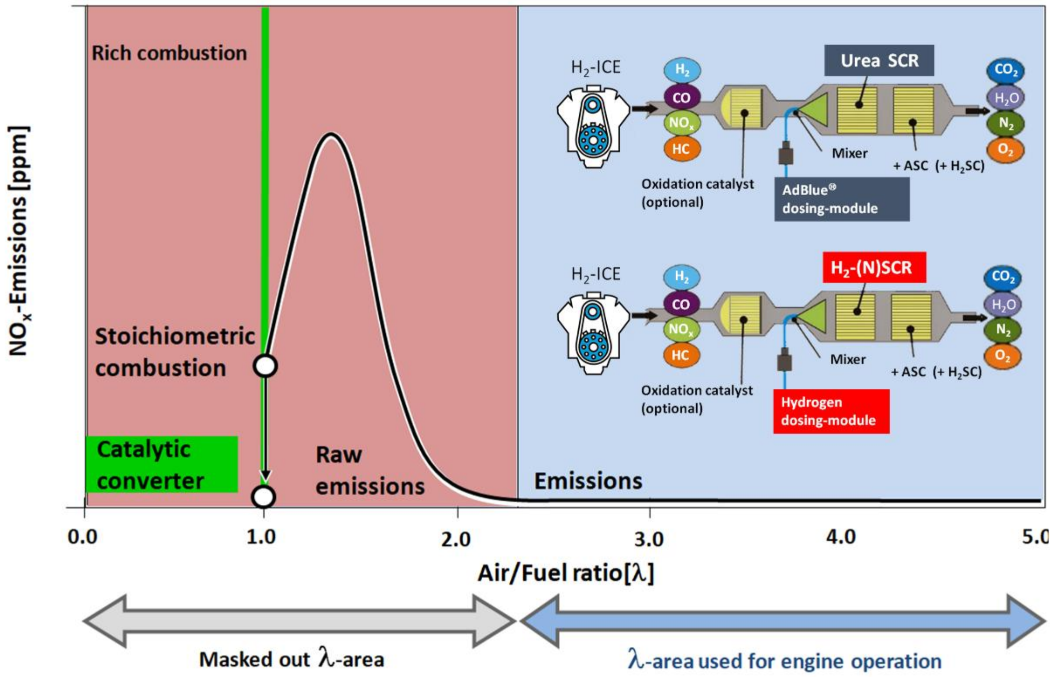


Figure 7 NO_X raw emissions as a function of the fuel-air ratio and the exhaust gas aftertreatment variants

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