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Development of lubricants for hydrogen-fueled large engine power plants

Lubricants

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ABSTRACT

INNIO, the Large Engines Competence Center (LEC) and ExxonMobil have completed a joint R&D program to develop tailored engine oil solutions for future hydrogen-fired large engines and to understand the resulting lubrication needs relating to combustion control.

Hydrogen (H2), as a carbon-free energy carrier and as chemical energy storage, can be a viable addition to natural gas (NG) for hard-to-decarbonize sectors such as power generation. Maintaining stable and steady power generation requires a flexible backup solution, which can be provided by dispatchable large engine power plants currently using NG as fuel and, in the future H2-only or NG/H2 mixtures. Beside their multi-fuel capability, these engines will retain short start-up times, high efficiency over a wider range of power output, high load flexibility, modularity, and the capability of combined heat and power (CHP) production.

Engines have the potential to operate on a broad range of energy sources, from pipeline gas to hydrogen, but based on the different combustion properties (e.g., wider flammability range and lower ignition energy) compared to NG there are also some challenges, which are not yet fully understood. Therefore, among other things tailored lubrication solutions for large engines are required to better control the higher tendency for abnormal combustion of H2 containing gases, material interactions (elastomers and reciprocating components), wear mechanisms, and thermo-oxidative stability. As power plant operators look to reduce operational and maintenance costs and keep their engines in reliable operation for longer, they need robust lubricating oils for H2 engines. In this paper, a single cylinder engine testing procedure is described that was established by INNIO and LEC to investigate oil-induced pre-ignition for pure hydrogen combustion especially under high load (high BMEP) operating conditions.

A high-performance engine oil was developed by ExxonMobil that supports stable hydrogen combustion at the targeted large engine power output and will be the technology platform for current and next-generation INNIO`s Jenbacher H2 and NG high power output engines.

Development of lubricants for hydrogen-fueled large engine power plants

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Hydrogen (H_2), as a carbon-free energy carrier and as chemical energy storage, can be a viable addition to natural gas (NG) for hard-to-decarbonize sectors such as power generation. Maintaining stable and steady power generation requires a flexible backup solution, which can be provided by dispatchable large engine power plants currently using NG as fuel and, in the future H_2-only or NG/H_2 mixtures. Beside their multi-fuel capability, these engines will retain short start-up times, high efficiency over a wider range of power output, high load flexibility, modularity, and the capability of combined heat and power (CHP) production.

Engines have the potential to operate on a broad range of energy sources, from pipeline gas to hydrogen, but based on the different combustion properties (e.g., wider flammability range and lower ignition energy) compared to NG there are also some challenges, which are not yet fully understood. Therefore, among other things tailored lubrication solutions for large engines are required to better control the higher tendency for abnormal combustion of H_2 containing gases, material interactions (elastomers and reciprocating components), wear mechanisms, and thermo-oxidative stability. As power plant operators look to reduce operational and maintenance costs and keep their engines in reliable operation for longer, they need robust lubricating oils for H_2 engines. In this paper, a single cylinder engine testing procedure is described that was established by INNIO and LEC to investigate oil-induced pre-ignition for pure hydrogen combustion especially under high load (high BMEP) operating conditions.

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1 INTRODUCTION

The increase in the world's population and growing prosperity [1] in developing countries will lead to a rise in global energy demand [2] and in case no action is taken, to an increase in carbon intensity related greenhouse gas (GHG) emissions [3] in the coming years. *Energy and human development are tightly linked. Between now and 2050, the world population is expected to grow to almost 9.7 billion from 7.8 billion people, and global gross domestic product (GDP) is expected to more than double (text is from* [4] *page 23)… Efficiency gains and a shift in the energy mix, including increased use of lower-carbon sources, enable an improvement in the carbon emissions intensity of global gross domestic product from 2021 to 2050. Oil and natural gas remain important (text is from* [4]*, page 24).* Internal combustion engines (ICE) convert the energy chemically bound in hydrocarbon-based fuels (e.g., Natural Gas=NG) into usable energy. Carbon dioxide (CO_2), a greenhouse gas, is produced when hydrocarbons are burned (e.g., turbines, engines, boilers).

To lower carbon intensity related GHG emissions from power generation applications, there is a demand for energy systems with low-carbon or carbon-free fuels. Various combustion concepts for hydrogen (H_2) and H_2/NG operation are under development for power generation. Based on the physical properties of H_2, the key features of the potential combustion concepts need to be adjusted compared to NG operation [5].

H_2 combustion concepts can feature central mixing, port fuel injection or direct injection, and ignition can be achieved via spark discharge or pilot injection. While direct injection provides benefits for the achievable power density and the prevention of backfire, it also has higher complexity and requires higher H_2 pressure [6]. A detailed investigation with a quantitative assessment of benefits and drawbacks is required to determine which technology is best suited for a new application. Pure H_2, as well as dual fuel (H_2/NG) combustion engines that are under development can not only reduce CO_2 emissions but also reduce hydrocarbon (HC) emissions while also lowering NO_X emissions. Pure H_2 combustion engines can nearly fully avoid the production of CO_2 emissions

[7]. H_2 combustion systems can be prone to pre-ignition and even backfire [8]. Different uncontrolled combustion scenarios are defined in chapter 2.3. The present paper summarizes findings from the literature that show the main influencing parameters for H_2 pre-ignition in engine operation with pure H_2 fueling and potential means to alleviate them. Lubricants can play a role in H_2 pre-ignition, and the paper briefly describes the processes that are involved.

Literature results suggest that pre-ignition is initiated in the immediate vicinity of glowing solid particles, that originated either from the flaking off of deposits or because of the presence of lubricant droplets within the combustion chamber ejected from the piston crevices after a pre-ignition event [9]. Usage of wide and shallow crevices allowed mitigation of pre-ignition. From these results it was hypothesized that the magnitude of the liquid reservoir consisting of a mixture of fuel and lubricant in the piston ring crevices strongly influences the occurrence of pre-ignition [10].The lubricant properties are found to have an impact on the lubricant-induced pre-ignitions [11]. In particular, aliphatic, paraffinic hydrocarbons featuring high molecular weights (18 to 32 carbon atoms) as well as specific calcium-based detergents were identified as promoting agents for pre-ignition. Furthermore, the viscosity and the surface tension of the lubricant played a role in the likelihood of pre-ignition [10].

A single cylinder test for INNIO`s Jenbacher Type 4 and Type 6 large engine platform was developed by INNIO and the Large Engines Competence Center to investigate oil-induced pre-ignition for pure H_2 combustion especially under high load (high BMEP) operating conditions. This test platform was used for optimization of lubricant technology in that regard. A high-performance engine oil was developed by ExxonMobil that supports stable H_2 combustion at the targeted power output. It will be the technology platform for current and next-generation high power output INNIO`s Jenbacher H_2 and NG engines.

2 HYDROGEN ENGINE FOR POWER GENERATION

2.1 Combustion concepts

The combustion system in an internal combustion engine is defined by numerous design and operating parameters. While many of the operating parameters might be a function of engine speed and/or engine load or might even change with the engine ambient conditions, other parameters do not vary with the operating conditions but must be optimized during engine development. The fuel admission and dosing and the air-fuel mixture formation are especially crucial for hydrogen-fueled engines and describe fundamental concept choices.

Fuel admission and mixture formation

The dosing of the fuel and the mixture of fuel and air for combustion inside an internal combustion engine can be achieved in multiple ways. Figure 1 illustrates potential H_2 fuel admission concepts.

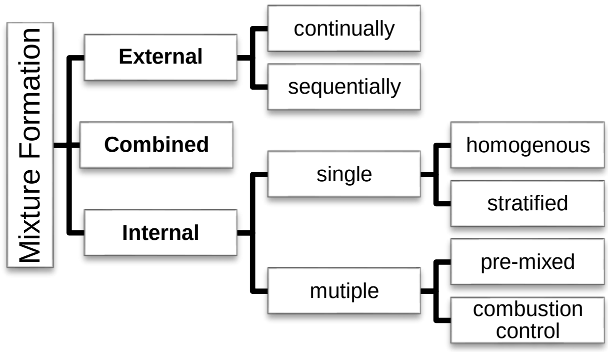

Figure 1. Fuel admission concepts [12]

The main options are central mixture formation (with carburetion, upstream or down-stream of an optional turbocharger), decentral (individually for each individual cylinder), port fuel injection (PFI) and direct injection (DI) into the combustion chamber. The latter can be further divided into high-pressure direct injection for non-premixed combustion or low- to moderate-pressure direct injection for premixed combustion. Decentral mixture formation has a greater number of degrees of freedom for combustion control [6], [13].

Central mixing of H_2 and air has the advantage that only a low-pressure hydrogen supply is required. Furthermore, because a long time is available for the mixing, the best homogeneity can be achieved with central mixing. There are drawbacks, however, with central mixture formation. A combustible mixture is formed far upstream of the engine cylinders and therefore a large volume – the complete intake manifold – is filled with an ignitable and combustible mixture. Due to the wide flammability range of H_2, there is a higher risk of backfiring than with an NG-fueled engine. In addition, the H_2 is displacing air, reducing the air mass that can be delivered to the engine cylinders with a given turbocharger and valve timing configuration. In addition to the points discussed above, H_2 also needs to be compressed in the compressor of the turbocharger.

Port fuel injection of H_2 requires an elevated pressure level ensuring a sufficient pressure difference to the intake manifold pressure, but it has the advantage that during transient operation the fuel injection can be adjusted to meet the demand instantaneously. In addition, it is possible to adjust the fuel injection for each cylinder individually. The port fuel admission can be done continuously or intermittently during the intake stroke. To minimize the risk of backfiring as little fuel as possible should be present in the intake manifold [14]. Therefore, the intermittent fuel admission usually is preferred.

Direct injection of H_2 into the combustion chamber can take place during the intake stroke, early in the compression stroke – allowing some homogenization – or late in the compression stroke. The fuel can be injected with one or more events per cycle, which allows significant differences in the charge composition and distribution, and even stratification to be realized. For all cases where the H_2 injection starts after the closing of the intake valves, backfiring into the intake manifold can be prevented. For different injection times different fuel supply pressures are required. For early injection, fuel supply pressure of 10 bar to approximately 40 bar is sufficient [15]. For injections later in the compression stroke where a critical pressure ratio is still required to ensure that the injection duration is not dependent on the cylinder pressure, supply pressures of at least 50 bar are required [13], [16]. If injection during the combustion process is desired, supply pressures of more than 100 bar are required [16]. Since H_2 compression or liquefaction is very energy-intensive and costly, a thorough assessment is required if the benefits of late injection outweigh the additional effort for high-pressure H_2 supply. Injector technology for H_2 direct injection for large bore engines is in development and not yet commercially available [15].

A combination of the aforementioned fuel admission variants also is feasible. These include the dual-fuel combustion process, in which two different fuels are burned in the combustion chamber at the same time. In that case, H_2 is introduced into the intake manifold or the combustion chamber and ignited in the combustion chamber by a diesel pilot injection. Early concepts for injectors that can supply H_2 as well as diesel fuel at the same time exist (cf. Figure 2) but are not yet commercially available [17]. A comparison of H_2 fuel admission via central mixing and port fuel injection for a high-speed large-bore engine (bore approximately 140 mm) fueled with hydrogen was performed by LEC GmbH in 2020 [17]. Combining the lower heating value of H_2, its density and the stoichiometric air requirements, the maximum theoretical power density for different fuel admission concepts can be calculated [18]. The introduction of H_2 at ambient temperatures into the intake manifold for central mixture formation does not require high supply pressure or additional fuel injection equipment. However, it also has considerable disadvantages. Due to the low density of H_2, the fuel takes up a higher fraction of the volume in the mixture than other fuels and therefore displaces air. This reduces the heating value of the gas mixture on a volume basis. With internal mixture formation, this air displacement effect in the intake manifold can be prevented, and the heating value of the mixture is higher, increasing the achievable power density [18].

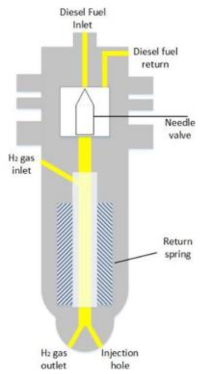

Figure 2. Schematic of DI-H_2 injector [17]

The full load potential of H_2 combustion systems with different fuel admission variants was quantified in [19] relative to a gasoline engine (cf. Figure 3). While H_2 admission into the intake manifold at ambient conditions resulted in a reduction of theoretically achievable power density, direct H_2 injection into the cylinder increased the theoretically achievable power density by approximately 17%.

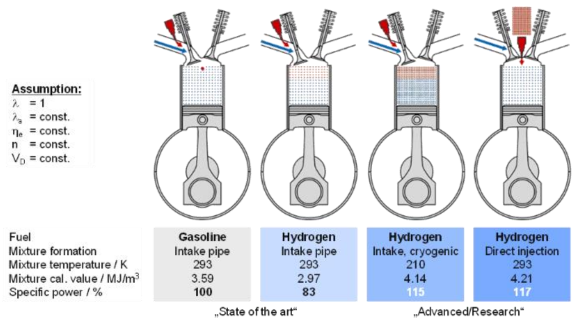

Figure 3. Full load potential of hydrogen combustion systems in comparison to gasoline combustion system [19]

What part of this potential can be realized for direct injection of hydrogen compared to hydrogen central mixture formation and gasoline operation is shown by the results of experimental studies in Figure 4 [18]. The full-load disadvantage of hydrogen port fuel injection for the engine configurations investigated in that study can be converted into an advantage of about 13% for hydrogen direct injection compared to conventional gasoline operation. The disadvantage of central mixture formation or port fuel injection might even be exacerbated by combustion anomalies, (i.e., backfiring into the intake manifold), which tend to increase when approaching stoichiometric mixture conditions.

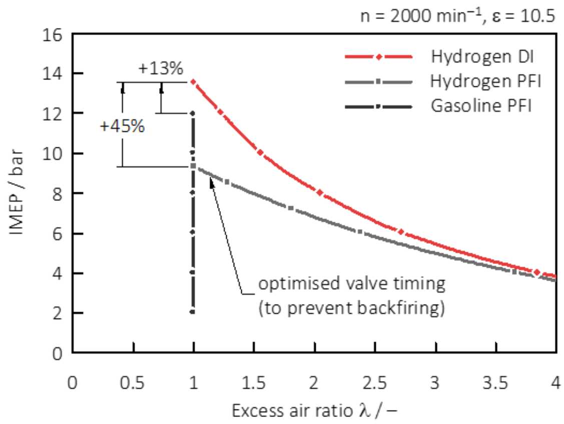

Figure 4. Comparison of achievable mean effective pressure for gasoline and hydrogen operation at various excess air ratio EAR [18]

While combustion anomalies are less frequent for direct fuel admission concepts there are other limitations. Due to the high laminar flame speed of H_2, the injection timing of the direct fuel injection has a significant impact on the heat release rates (HRR) [20]. Figure 5 shows the HRR for early, medium, and late hydrogen injection timing.

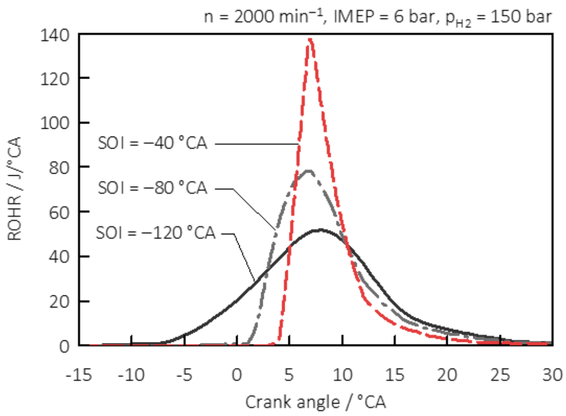

Figure 5. Impact of fuel admission timing on HRR in DI operation [20]

Extremely short, efficient combustion durations can be achieved, especially with late injection timing. However, the engine components are subjected to higher stress due to higher peak cylinder pressures and pressure rise rates, which also lead to higher combustion noise.

2.2 Fuel properties of hydrogen

Some key features of H_2 combustion systems derive from the physical and chemical properties of hydrogen and hydrogen/air mixtures. Table 1 lists some properties of hydrogen (H_2) compared to methane (CH_4) and iso-octane (C_8H_{18}) [8] which are considered as representative for natural gas (NG) and gasoline (C_8H_{18}), respectively.

Table 1. H_2 properties compared to CH_4 and C_8H_{18} properties [8]

Property	Unit	Hydrogen H_2	Methane CH_4	Iso-octane C_8H_{18}
Molecular weight	g/mol	2.02	16.04	114.24
Density (300°K)	kg/m³	0.08	0.65	692
Mass diffusivity in air	cm²/s	0.61	0.16	~0.07
Min.[(a)] ignition delay	mJ	0.02	0.28	0.28
Min[(a)] quenching distance	mm	0.64	2.03	3.5
Flammability limits air	vol%	4-75	5-15	1.1-6
Flammability limits	λ	10-0.14	2-0.6	1.51-0.26
Flammability limits	φ	0.10-7.10	0.50-1.67	0.66-3.85
Lower heating value	MJ/kg	120	50	44.3
Higher heating value	MJ/kg	142.0	55.5	47.8
Stochiometric AFR	kg/kg	34.2	17.1	15.0
Stochiometric AFR	kmol/kmol	2.39	9.55	59.67

(a) At EAR approx. 1.2 – 1.5 at atmospheric conditions

The small and light H_2 molecule leads to a very low density at atmospheric conditions. Its wide flammability limits (excess air ratio (EAR) approximately 0.14 to 10) far exceed the limits of the compared hydrocarbon fuels (methane and iso-octane) and allow coverage of a wide range of the operating window through changes in the EAR. The flammability limits widen with increasing temperature, with the lower flammability limit dropping to approximately EAR = 20 at 300 °C. The minimum ignition energy of a lean hydrogen/air mixture (EAR approx. 1.2–1.5) at atmospheric conditions is about 0.017 mJ an order of magnitude lower than for CH_4/air and C_8H_{18}/air mixtures. Figure 6 illustrates that the minimum ignition energy of H_2/humid air (relative humidity: 90 %) and H_2/dry air mixtures compared to compressed natural gas (CNG) in air approximately in range of rel. AFR from 0.5 to 4 is significantly below 0.1 mJ. This ignition energy can be provided by the energy content of oil droplet sizes below 30 µm or less (based on FEV modeling work from Dipl.-Ing. Aleksandar Boberic, and Dr. Lukas Virnich; $E=1/6\ \rho\ \pi\ D^3\ H_u$). Therefore, it can be assumed that a burning oil droplet can be an ignition source for a hydrogen air mixture in the combustion chamber over a wide relative AFR range. The stoichiometric air-to-fuel mass ratio of H_2 is about twice as high as for CH_4 and C_8H_{18}, while its stoichiometric air-to-fuel mole ratio is significantly smaller. The high H_2 volume fraction in the fuel-air mixture has an impact on the achievable power density. Different autoignition temperatures of hydrogen have been reported by different researchers due to the sensitivity of the measurement to the experimental procedure that is used. A range from 773 K to 858 K has been reported for stoichiometric mixtures and a further increase with higher EAR.

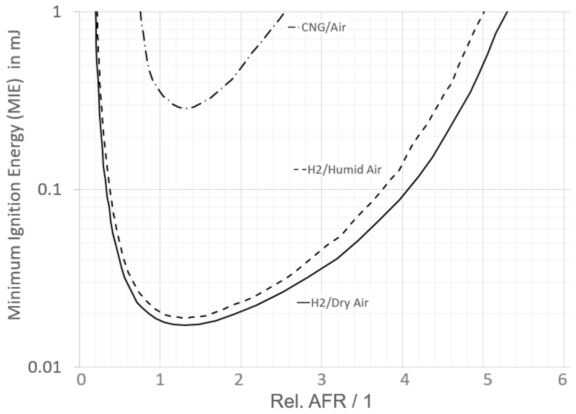

Figure 6. Minimum ignition energy of hydrogen/humid air (relative humidity: 90%) and hydrogen/dry air mixtures; FEV modeling work from Dipl.-Ing. Aleksandar Boberic, and Dr. Lukas Virnich based on [21]

The laminar flame speed of stoichiometric H_2/air mixtures is much higher than that of CH_4 and C_8H_{18} but is significantly reduced for lean mixtures where it achieves similar values as methane/air mixtures for stoichiometric conditions.

2.3 Definition of uncontrolled combustion

To understand abnormal combustion and how it can be alleviated it is important to understand and differentiate the various types of combustion anomalies.

•**Pre-ignition** – combustion events occurring inside the combustion chamber during the compression stroke prior to spark ignition

•**Backfiring** – combustion events occurring during the intake stroke in the cylinder or the intake manifold

•**Knocking** – auto-ignition of the end gas in the later part of the combustion process

Surface ignition normally describes a combustion event induced by a hot spot in the combustion chamber. It can occur at different times in the process. Figure 7 illustrates how the occurrence of combustion anomalies can be observed and differentiated in the measurement data. It shows the in-cylinder pressure traces as well as the crank-angle resolved intake manifold pressures and heat release rates for regular combustion and for different combustion anomalies. The regular combustion (green) exhibits the expected behavior with the cylinder pressure increase and heat release after the ignition timing. Combustion cycles in which pre-ignition occurred are shown in black and red. In both cycles the heat release starts before the ignition timing and both cycles exhibit higher peak cylinder pressures than for regular combustion.

Pre-ignition, is a stochastic event that typically occurs during the compression stroke after the intake valves have been closed, can have numerous causes. Potential triggers or causes include hot spark plugs or spark plug electrodes, hot exhaust valves or other hot spots in the combustion chamber, residual gas from the previous combustion event, hot oil particles or residual charge in the ignition system. Contributing factors are engine operating conditions. Since the minimum ignition energy decreases with EAR, pre-ignition occurs more frequently with lower EAR and particularly with near-stoichiometric conditions. Higher power densities contribute due to increased fluid and component temperatures.

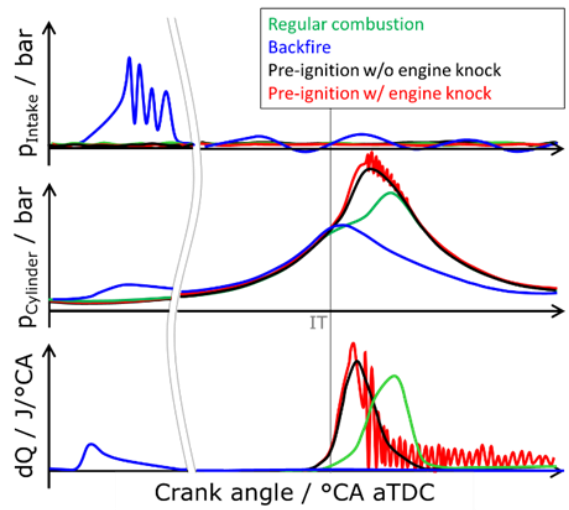

Figure 7. Illustration of typical cylinder and intake manifold pressure (p) traces and heat release rates (dQ) for regular combustion cycles, pre-ignition cycles and backfiring cycles

A backfiring cycle is shown in blue in Figure 7. The main difference between backfiring and pre-ignition is the timing at which the anomaly occurs. Backfire occurs during the intake stroke in the combustion chamber and/or the intake manifold when the fresh H_2/air mixture is ignited at combustion chamber hot spots, hot residual gas or particles, or remaining charge in the ignition system. This results in combustion and pressure rise in the intake manifold, which can severely damage the intake system. The pressure rise also is observed in the cylinder pressure trace. Once the entire fresh charge is burned, the pressure in the intake manifold decreases; the cylinder pressure at intake valve closing is increased compared to the regular trace while the peak cylinder pressure for this backfiring cycle is significantly lower than for the regular combustion cycle. As with pre-ignition, backfire occurs more frequently in near-stoichiometric conditions compared to lean operation. Direct injection of H_2 after the closing of the intake valves can prevent a backfire event from happening. There is some indication that pre-ignition also might be a trigger for backfiring due to its impact on combustion chamber temperature.

2.4 Impact factors for abnormal combustion

Design parameters and component specifications

Design measures to avoid abnormal combustion include spark plug design, design of the ignition system ensuring low residual charges, crankcase ventilation, sodium-filled exhaust valves, and cylinder head and piston cooling concepts that avoid hot spots in the combustion chamber. H_2 direct injection can prevent backfire into the intake manifold and, especially in combination with the injection strategy, also can reduce the risk of pre-ignition. At the same time, H_2 direct injection can reduce the power reduction due to EAR requirements and air displacement by H_2 in the intake manifold. If contact of H_2 or H_2/air mixture with hot spots in the combustion chamber cannot be avoided, limiting the exposure time can reduce the occurrence of abnormal combustion events. Charge air motion with increased flow velocities, e.g., due to increased swirl, can contribute to a reduction of exposure time. To minimize chemical decomposition of lubricating oil in the combustion chamber or in the crevices of the piston top land several design measures could be applied: improved filtering in the crankcase ventilation system, piston, and piston ring design for reduced blow-by from the crank case and reduced lube oil consumption and improved valve guide sealing.

Engine operating parameters

Excess air ratio: Limiting the minimum EAR is an effective measure to reduce the likelihood of abnormal combustion in H_2 operation due to its impact on gas temperature and therefore component temperatures in the combustion chamber and the autoignition temperature. In [12] the minimum EAR was dependent on engine speed and defined as 1.6 @ 1,500 RPM and 2.1 @ 6,000 RPM. Since H_2 has wide flammability limits and high laminar flame speed, lean combustion concepts are feasible and widely used in research applications. To avoid power output reductions due to high EAR, the turbocharger layout must be optimized for hydrogen operation.

Fuel injection strategy: H_2 that is present in the intake ports and if at the time the intake valves are open it can come in contact with hot spots in the combustion chamber (e.g., residual gas, exhaust valves) and can cause pre-ignition and backfire. Therefore, the fuel injection strategy should ensure that no H_2 is maintained in the intake port after the intake valve closing time and that the injection for the next cycle does not start too early. Ideally the H_2 is admitted to the combustion chamber after an initial cooling of hot spots (e.g., exhaust valves or spark plug electrodes) in the combustion chamber by pure air.

2.5 Impact of lubricants on hydrogen combustion anomalies

The impact of lubricants on H_2 combustion anomalies often depends on other impact factors, (e.g., piston ring design, blow-by from the crank case, injection timing, mixture homogenization etc.) In general, all measures that can reduce the entry of lubricating oil into the intake manifold or the combustion chamber can contribute to a reduced risk of combustion anomalies. Overall, it cannot be fully avoided that lubricating oil can enter the combustion chamber through blow-by from the crankcase, through leakage past the valve guides seals, and from the positive crankcase ventilation system. Several mechanisms are associated with lubricant-induced pre-ignition events, but an experimental and quantitative differentiation often is not feasible. Three main factors play a role in the introduction of lubricant into the combustion chamber via the crankcase/piston rings:

- Lube oil evaporation cylinder liner wall film
- Droplet detachment piston top land and crevices
- Deposit detachment

Near-wall processes contributing to lubricant evaporation are described in the literature [22] and [23] (cf. Figure 8).

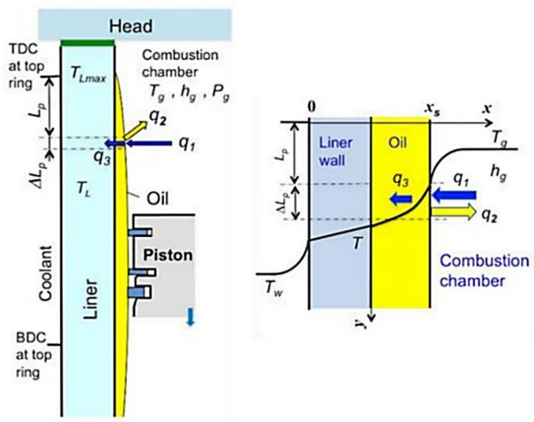

Figure 8. Near-wall processes contributing to lubricant evaporation [23]

Evaporation of volatile components of the lubricant in conjunction with incomplete mixing with the surrounding gases generates mixture pockets close to the liner that exhibit different autoignition properties than the rest of the mixture. The postulated mechanism behind the transportation of lubricants into the combustion chamber are depicted in Figure 9 for a combustion concept with direct injection of a liquid fuel into the combustion chamber. For H_2 combustion concepts, the fuel injection will be in gaseous form such that the fuel spray impact depicted will not occur. While dilution of the lubricant with fuel is not expected to be an issue, the higher water content in the combustion products compared to NG combustion also might impact the lubricant dilution and the composition of the mixture in the piston top land crevices.

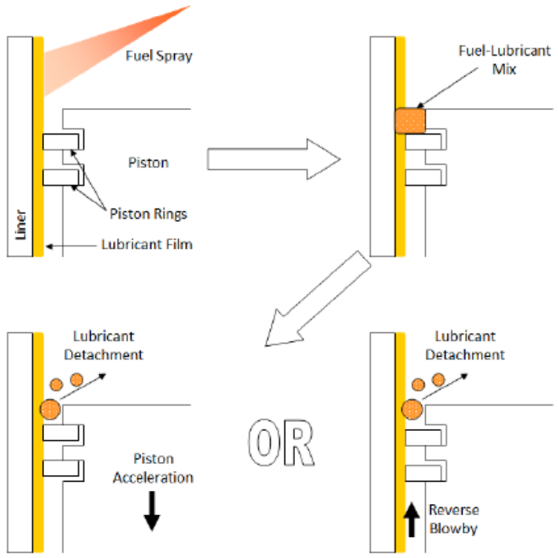

Figure 9. Postulated mechanisms behind the transportation of lubricants into the combustion chamber [24]

The main mechanism for lubricant transportation is represented by droplets ejected out of the wall film or the piston top land and crevice. The lubricant evaporates, and a mixing layer develops around these droplets in which autoignition can be accelerated. This is due to evaporation of lubricant components that might exhibit a different ignitability compared to the fuel/air mixture surrounding it. If the lubricant evaporation is followed by oxidation, the temperature increase can trigger the autoignition of the surrounding gaseous fuel/air mixture. Larger lubricant droplets can survive the combustion, and the non-volatile constituents can remain in the combustion chamber at the end of the exhaust stroke. The inorganic contents of the lubricant also may act as precursors for formation of larger particulates. Furthermore, lubricant is known to contribute to the formation of combustion chamber deposits (e.g., from lubricants that enter the combustion chamber and collect in liquid form on the piston top), followed by partial evaporation. Deposit flaking and subsequent breakup of the larger fragments also leads to the formation of large particulates.

Magar et al. [10] used different experimental techniques and in-cylinder visualization to investigate the root causes of pre-ignition in gasoline engines. They found that during the compression stroke of cycles exhibiting pre-ignition, luminous spots occurred in the combustion chamber that were not present in regular cycles without pre-ignition. The location of the onset of pre-ignition generally was correlated to the position of the luminous spots, leading to the conclusion that the luminous spots were the sources of the pre-ignition. The tracking of the luminous spots showed several collisions of these objects with the combustion chamber walls. Since no significant change in size and shape of the spots occurred, it was concluded that these spots are solid particles. This finding is corroborated by other authors [25], who detected glowing particles during the compression stroke of the cycles exhibiting pre-ignition. Sudden changes of the direction of particle trajectories probably resulting from the rebound of the particles at the cylinder walls lead to the conclusion that these particles likely are solid.

The optical investigations in [10] further revealed a strong contamination of the combustion chamber with droplets and particles following a pre-ignition event. The authors suggest that the contaminants are released from a reservoir that is depleted gradually and that the reservoir is the mixture of lubricant and fuel trapped inside the top land crevice. Magar et al. [10] conclude that pre-ignition is induced by hot solid particles originating from the flaking of deposits or being formed from lubricant droplet precursors ejected from the liquid reservoir in the top land gap. The lubricant properties determine which of the described effects is the dominant one. The viscosity and the cylinder wall wetting characteristics are of particular importance since these parameters are expected to exhibit an immediate impact on the magnitude of the liquid reservoir in the ring crevice. The surface tension governs the droplet forming tendency of a liquid. A decrease of the surface tension potentially can mitigate droplet scattering from the piston ring crevice. An increase of the viscosity of the oil is expected to reduce the amount of oil transported through the piston rings to the top land crevice.

Furthermore, a higher viscosity of the lubricant or molecular structure should inhibit droplet release from the ring crevice. The viscosity also is expected to have an impact on the droplet once it is released. With increasing viscosity, the evaporation usually is reduced, and the lubricant molecules remain more compact in the oil droplet. This results in an inhomogeneous distribution in the combustion chamber and potentially a high local energy density. One hypothesis states that this trend reverses with very high viscosities when the combustion chamber temperatures are insufficient for the evaporation of lubricant components and generation of higher reactivity mixture pockets.

Kieberger [26] listed several hypotheses for pre-ignition in gasoline engine operation that the author subsequently tried to substantiate or dismiss. One of the hypotheses is that with decreasing surface tension, it becomes more likely that lubricant that is scraped off the liner by the piston motion is forming droplets that detach from the wall film and become airborne in the combustion chamber. This hypothesis was neither confirmed nor dismissed in the study.

Only few systematic studies researching the root causes and the impact of specific lubricant parameters on pre-ignition in H_2-fueled internal combustion engines are available in the literature. It can be hypothesized that the general mechanisms leading to pre-ignition are similar for different fuels and that at least the trends observed for non-hydrogen fueling, e.g., in gasoline operation, can be transferred to hydrogen operation. Especially since the operating conditions where pre-ignition often occurs in gasoline engines, engine speed in the range of 1,000–1,500 RPM and high mean effective pressures, also represent the typical operating conditions of large-bore engines.

Splitter et al. [11] investigated the impact of various operating parameters, fuel and lubricant specifications, and fuel/lubricant interactions on stochastic pre-ignition in gasoline engines. They found that lubricant effects on stochastic pre-ignition was minimal at low load conditions but significant at high load conditions, suggesting that there is a thermal component to lubricant detergent effects on pre-ignition. For gasoline spark-ignition operation, the following lubricant properties were found to have a significant impact on the pre-ignition tendency in the literature: viscosity, surface tension, base oil and additives (e.g., detergents).

With a commercial lubricant featuring an SAE viscosity of 0W-40 as a baseline the physical properties of the lubricant were modified with different additives in an experimental study reported in [10] where the pre-ignition rate was assessed for the different lubricants used. Modification of the surface tension of the oil resulted in an improved wetting behavior of the lubricant and was expected to reduce the droplet scattering from the piston ring crevice. The results of this study are displayed in Figure 10. As expected, the number of pre-ignition events decreased significantly with the reduced surface tension of the lubricant. This figure also shows results from another test sequence where a viscosity improver was added to the lubricant and the number of pre-ignition events significantly reduced.

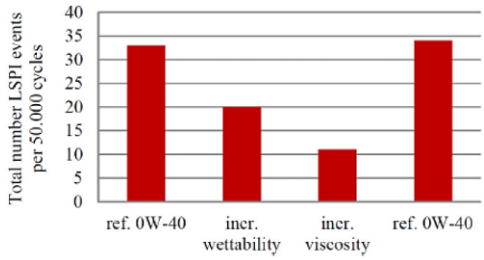

Figure 10. Effect of the addition of viscosity improver and an increased wettability on LSPI occurrence [10]

A similar finding was reported by He et al. [27] who investigated the autoignition behavior of different lubricant compositions in a controllable active thermo-atmosphere. It was found that the ignition delay of the lubricant increased with an increase of the kinematic viscosity. Welling et al. [28] found a correlation between the physical properties of the lubricant (density and kinematic viscosity) and the propensity to autoignition. An increase in the density as well as in the kinematic viscosity of the lubricant caused a substantial drop in the observed ignition frequencies. Luef found a critical viscosity range of 8–23 cSt @ 40 °C where increased pre-ignition occurred [29]. Luef [29] additionally ranked the most influential lubricant properties for pre-ignition events with the most important being: molecular structure of the base oil, molecular mass (C_xH_y), kinematic viscosity of the base oil and boiling curve.

The worst performance regarding pre-ignition was shown by aliphatic (non-aromatic) and, in particular, paraffinic hydrocarbons (C18-C32) in the mineral base oils. These also included higher n- and iso-alkanes ($C_{14}H_{30}$), which generally exhibit a high tendency to spontaneous combustion. Higher alkanes (>$C_{16}H_{34}$) were found to trigger pre-ignition. The opposite was demonstrated with purely synthetic base oils of API group V (e.g., ester oils, naphthenes, aromatics), which exhibited predominantly inconspicuous behavior with regard to pre-ignition due to their molecular structure. Typical lubricant additives had only secondary effects, e.g., on aging and deposit formation due to ash-forming and metal organic components. Welling et al. [28] determined the propensity to autoignition of various lubricant base oils and mixtures of base oils with different additives in a production engine and also pointed out the impact of the base oil and showed strong disparities in the ignition tendency between base oils from the different API classes and even between the different representatives within a specific API category.

Park et al. [30] studied the effects of various types of lubricants and additives on pre-ignition in a turbocharged gasoline engine. They hypothesize that oil particles get released from the piston top land and crevices to the combustion chamber, evaporate and oxidize and due to their temperature increase are the trigger for the auto-ignition of the fuel-air mixture. They showed that the lubricant composition had an impact on the pre-ignition and concluded that Ca, Zn, and Mo content in the lubricant additive had an impact on pre-ignition frequency. While detailed investigations of the impact of lubricant properties on pre-ignition in hydrogen combustion engines are still outstanding, some observations and recommendations for lubricants in hydrogen combustion engines can be found in the literature.

Swain et al. [31] attributed the impact of lubricants on pre-ignition to oil vapor and oil deposits on the piston. During low load operation lubricant that enters the combustion chamber can collect in liquid form on the piston top. During transition to high load operation this liquid oil can be evaporated and create an ignitable hydrogen-air-oil-vapor that can lead to pre-ignition. Any substantial deposits that remain on the piston after evaporation can heat up and cause surface ignitions. Swain therefore recommends lubricants with high vaporization temperature and oil control via optimization of valves, valve guide and valve stem seal, and piston rings and piston. Furthermore, a reasonably large skirt thickness and cast pistons were recommended to improve heat transfer from the piston head/ring land area to the piston skirt. The coolant water temperature was reduced to lower the cylinder-block temperature and lube oil temperature. Verhelst et al. [32] summarized root causes for abnormal combustion and cited other authors [33], [34] naming deposits and particulate matter originating from the (partial) combustion of lubricating oil as hot spots in the combustion chamber that are contributing to pre-ignition. Liu et al. [35] named hot oil ash in the combustion chamber as a source of hydrogen pre-ignition. Das [36] identified particulate matter resulting from the pyrolysis of lubricating oil vapors as a potential cause of hot spot-induced backfire and cited other authors recommending synthetic ashless lubricants for hydrogen engine applications. The oil could be suspended in the combustion chamber or in the crevices just above the top piston ring. Fayaz et al. [37] named remaining hot oil particles from previous combustion events as one source for pre-ignition. In addition, any carbon-based deposits from the engine's lubricating oil on the top of the piston, in piston ring grooves or in the cylinder's squish areas are potential hot spots and therefore potential sources of pre-ignition. For an automotive application, Kiesgen et al. [38] also named oil carbon deposits as one root cause of pre-ignition and highlighted the importance of oil composition to fulfill the tribological requirements but also considered the special conditions related to hydrogen operation. They selected a low-ash oil of grade 0W-30 to guarantee sufficient viscosity in the piston ring area and avoid oil carbon formation.

3 TEST METHOD DEVELOPMENT TYPE 4 AND TYPE 6 SINGLE CYLINDER ENGINE

3.1 Test procedure Type 4 SCE and Type 6 SCE

In this chapter the selected single cylinder test rig, hardware configuration, test procedure, and mechanical sensitivity are described. Before beginning lubricant screening tests, it is important to ensure that mechanical influences that impact test-to-test variation of abnormal combustion events are reduced to a minimum. Often new mechanical hardware/seals require some operating hours to reduce variability in oil leakage and improve sealing capabilities. Two contributors to mechanical oil consumption are from the piston ring and liner system, along with cylinder head valve sealing components. Oil protector caps were tried on intake valves to reduce the leakage rate into the combustion chamber, allowing an increase in engine load with a reduction in abnormal combustion events. Too high of a leakage rate from intake valves, or fluctuating oil consumption from the piston ring design may have a stronger effect on influencing combustion stability compared to the influence of different lubricant formulations.

3.1.1 Engine configuration Jenbacher Type 4 SCE and Type 6 SCE

The experimental investigations were carried out on two different high-speed four-stroke single cylinder research engines (SCE) derived from the INNIO`s Jenbacher Type 4 and Type 6 series. For the investigation of the impact of lubricant formulations on H_2 combustion, the Jenbacher Type 4 engine was used in an open chamber configuration with a centrally mounted spark plug. The compression ratio (CR) was chosen in the range of typical large engine applications with low flashpoint fuels and a camshaft with late intake valve closing (IVC) after bottom dead center was selected. The Jenbacher Type 6 engine was configured with a gas scavenged precombustion chamber and a camshaft with early IVC before bottom dead center and a similar compression ratio chosen for the Type 4 engine. Table 2 shows further information about the engines. The H_2 was admitted to the engine via port fuel injection. For this, a modified serial production gas dosage valve and an injection nozzle design – investigated with 3D CFD prior to the engine testing – were used.

Table 2. Type 4 and Type 6 SCE technical data

	Type 4	Type 6
Rated speed	1,500 rpm	1,500 rpm
Displacement	≈ 3 dm^3	≈ 6 dm^3
Valve timing	Atkinson valve timing	Miller valve timing
Number of intake and exhaust valves	2/2	2/2
Swirl/tumble	Swirl ports	≈ 0/0
Charge air	Provided by external compressors with up to 10 bar boost pressure	Provided by external compressors with up to 10 bar boost pressure
Hydrogen supply	Port fuel injection, up to 10 bar	Port fuel injection, up to 10 bar
Ignition system	Modified high-voltage capacitor discharge	Modified high-voltage capacitor discharge

Figure 11 shows a computer-aided design (CAD) model of the assembly of the nozzle and the intake ports in the Type 6 engine.

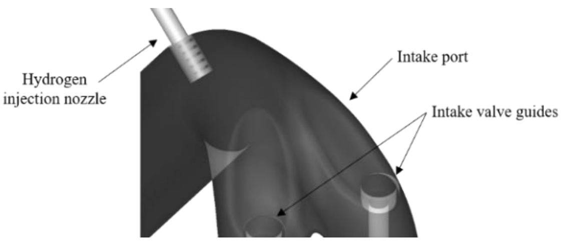

Figure 11. Position of the H_2 injection nozzle in the intake port (Type 6)

All engine fluids including cooling water, lubricating oil, fuel gas and charge air are controlled to ensure well-defined and reproducible testing conditions. Additional measuring instruments are shown in Table 3.

Table 3. SCE measurement instrumentation

Quantity	Instrumentation
Air mass flow	Emerson Micro Motion CMF300
Hydrogen mass flow	Emerson Micro Motion CMF050
Charge air humidity	Vaisala HMT 338
Charge air temperature	Resistance temperature sensor PT100
Charge air pressure	Piezoresistive pressure sensor
Cylinder pressure	AVL piezoelectric transducer QC34C
Exhaust gas temperature	Thermocouple type K
Exhaust gas pressure	Piezoresistive pressure sensor
Exhaust gas emissions	V&F HSense, IAG FTIR, AVL AMA i60

Instead of a turbocharger, an air compressor upstream of the engine and a flap in the engine exhaust system are used to adjust intake and exhaust manifold pressures. A flush mounted piezoelectric cylinder pressure transducer enables real-time calculation of the IMEP of each cycle.

3.1.2 Experimental evaluation approach of oil induced pre-ignition

Combustion anomalies and H_2 pre-ignition, are stochastic events that are difficult to quantitively assess. Therefore, the Large Engine Competence Center developed a methodology for the assessment of the impact of lubricant induced combustion anomalies based on engine load ramps and statistical analysis of the measurement results. To ensure repeatability of the load ramps, the operating procedure was automated. For each load level, setpoints for all four engine control parameters that determine the operating point were defined: injector current duration (ALPHA), intake manifold pressure, exhaust manifold pressure, and ignition timing. Figure 12 depicts the setpoints used for the investigations on the Type 4 SCE where the load ramp was divided into seven distinct load levels. The setpoints for the Type 6 SCE were prescribed in a similar manner.

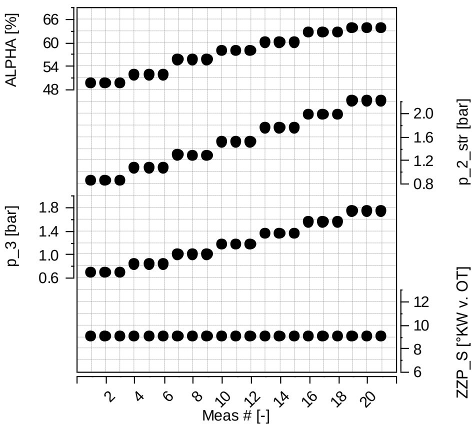

Figure 12. Engine control parameter setpoints for automated load ramps

The time sequence for the load ramps is illustrated in Figure 13.

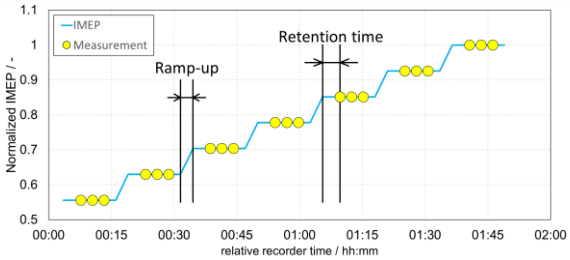

Figure 13. Time sequence for automated load ramps

For the load increase from one load level to the next higher one a ramp-up time of 5 minutes was prescribed. Once the target load was reached a retention time of 3 minutes allowed the engine operation and all significant parameters to stabilize. After a stable condition was achieved, three measurements were acquired with 1,100 cycles in each measurement. Each load ramp was repeated up to three times.

The post-processing of the measurement results used time-resolved data with a sampling frequency of 10 Hz as well as crank angle resolved data with 0.1 crank angle degree (CAD) resolution. The parameters used for the evaluation were the crank angle position when 5 % of the fuel energy was released (MFB05%) to quantify the intensity of the combustion anomaly and the peak cylinder pressure (PFP) to distinguish pre-ignition events from fast combustion events. Furthermore, the total number of combustion anomalies were counted. The statistical analysis used individual threshold definitions for the MFB05% and PFP for each measurement point. The threshold for the MFB05% was defined by the mean value and standard deviation as shown in Equation 1.

$$Threshold_{MFB05\%} = MeanValue_{MFB05\%} - 2.0 * Std.Dev._{MFB05\%} \qquad (1)$$

The threshold for the peak cylinder pressure used the mean value and a fixed multiplication factor (Equation 2).

$$Threshold_{PFP} = MeanValue_{PFP} * 1.2 \qquad (2)$$

A glow ignition was defined as a combustion cycle in which the MFB05% was smaller than the threshold while the PFP exceeded the PFP threshold (Equation 3).

$$MFB05\%_{Current\ cycle} < Threshold_{MFB05\%} \ \& \ PFP_{Current\ cycle} > Threshold_{PFP} \qquad (3)$$

A fast combustion was defined as a cycle in which the MFB05% was still larger than the threshold but the PFP still exceeded the PFP threshold (Equation 4).

$$MFB05\%_{Current\ cycle} > Threshold_{MFB05\%} \ \& \ PFP_{Current\ cycle} > Threshold_{PFP} \qquad (4)$$

The total number of combustion anomalies was defined as the sum of fast combustion cycles and glow ignition cycles (Equation 5).

$$\#\,Anomalies = \#Fast\ combustion + \#Glow\ ignition \qquad (5)$$

Figure 14 shows an illustration of the combustion anomaly detection. The cylinder pressure traces and accumulated heat release rates are shown for 1,100 consecutive engine cycles. The mean values are shown in (cyan), while the threshold values for this operating point are shown in (blue). A glow ignition cycle (yellow) and a fast combustion cycle (red) are highlighted individually. To capture the impact of the frequency and the intensity of the combustion anomalies the comparison of different lubricants was performed based on the integral value of the filtered MFB05% for the complete load ramp consisting of different load levels (cf. Figure 15).

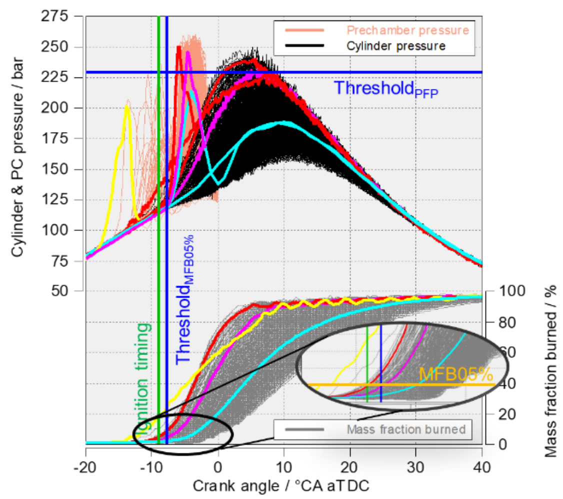

Figure 14. Illustration of combustion anomalies detection

With increasing load, the frequency of combustion cycles with MFB05% values exceeding the threshold increases significantly, resulting in steep increases of the integral value.

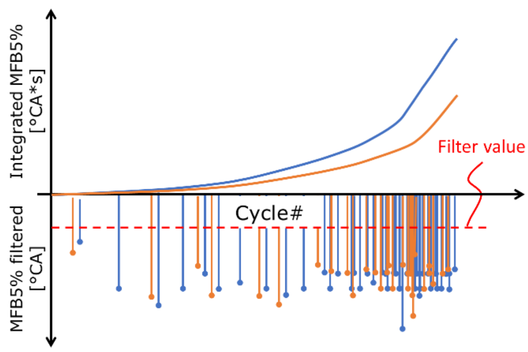

Figure 15. Analysis of combustion anomalies

The calculation of the integral indicator followed a multi-stop approach:

Standardization of measurement data

- $MFB05\%_{Standardized} = \frac{MFB05\% - MFB05\%_{AVG}}{MFB05\%_{STD}}$

Filtering of the measurement data

- $MFB05\%_{Standardized} > Threshold_{Integral}$
- $MFB05\%_{Filtered} \begin{cases} MFB05\%_{Standardized}, & MFB05\%_{Standardized} > Threshold_{Intgeral} \\ 0, & MFB05\%_{Standardized} < Threshold_{Intgeral} \end{cases}$

Integration (with incremental calculation of mean values and standard deviations)

- $MFB05\%_{Integral} = \int_{t_{Recorder\ start}}^{t_{Recorder\ end}} MFB05\%_{Filtered}\ dt$

3.2 Engine oil test matrix

Different experimental engine oils have been developed by ExxonMobil to investigate the effect of various base oil and additive combinations in relation to oil-induced pre-ignition for pure hydrogen combustion especially under high load (high BMEP) operating conditions. Table 4 summarizes the investigated engine oils which were tested on Type 4 and Type 6 single cylinder engine test platforms.

Table 4. Engine oil test matrix

Engine Oil	Developed for fuel type	SAE Class	Main Base oil Group	Additiv Comb-ination
GEO A	NG	40	II	A
GEO B	NG	40	II	B
GEO C-I	H_2	30	V	C
GEO D	NG	40	V	D
GEO E	H_2	30	V	C
GEO C-II	H_2	40	V	C

4 TEST RESULTS TYPE 4 SCE AND 6 SCE

Results Type 4 SCE

The cumulative analysis of the combustion anomalies in the load ramps with different lubricants is shown in Figure 16 and allows a statistically significant comparison of the lubricant performance in that regard.

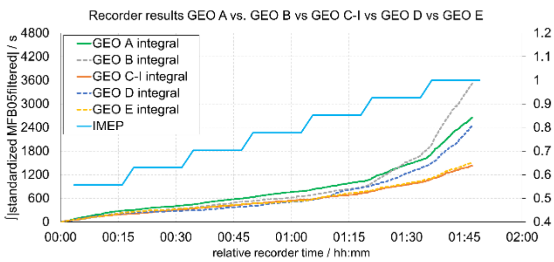

Figure 16. Cumulative analysis of selected oil types on Type 4 SCE

The single cycle analysis for three different lubricants (GEO A, GEO B, GEO C) used in the Type 4 SCE are shown in Figure 17, Figure 18, and Figure 19. The histograms depict for each measurement point the number of regular combustion cycles (green), the total number of combustion cycles with combustion abnormalities (red), and the differentiation of these combustion anomalies into glow ignitions (orange) and fast combustion cycles (yellow). For all lubricants the number of combustion anomalies at low load conditions is very low. The differences between the lubricants become apparent in the second part of the load ramp. The GEO A oil exhibits total combustion abnormalities in less than 1.5 % of the combustion cycles at the highest load condition (Figure 17). The GEO B in Figure 18 shows a higher number of total combustion abnormalities with nearly 2.5 % at the highest load condition.

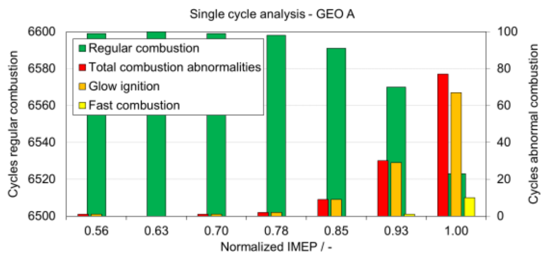

Figure 17. Single cycle analysis - GEO A

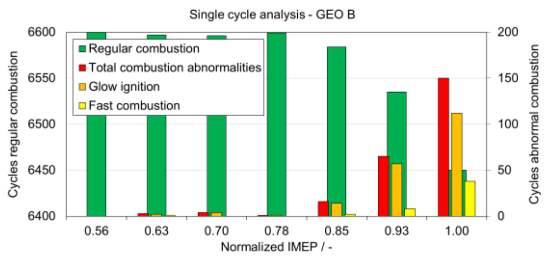

Figure 18. Single cycle analysis – GEO B

The benefit of the lubricant formulation that was specifically developed for H_2 applications can be observed in Figure 19 where, compared to GEO A and GEO B a smaller number of cycles with total combustion abnormalities were encountered and even at the highest load condition fewer than 0.5 % of the combustion cycles showed irregular behavior.

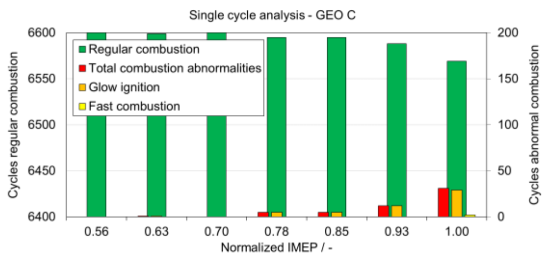

Figure 19. Single cycle analysis – GEO C-I

Results Type 6 SCE LEC

Based on the results on the Type 4 SCE an additional lubricant formulation (GEO C-II) was investigated on the Type 6 SCE. The cumulative analysis of the combustion anomalies is shown in Figure 20 in comparison to the reference oil GEO A. With the Type 6 SCE test procedure developed by the Large Engines Competence Center INNIO could confirm that the new SAE 40 formulation of the Group V oil maintained the excellent performance shown on the Type 4 SCE. In addition, it helped to reduce abnormal combustion events on Type 6 SCE more than 90 % compared to GEO A, and more than 80 % compared to other synthetic lubricants (not shown here) at the tested load ramp.

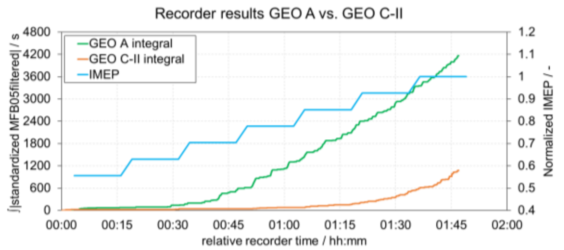

Figure 20. Cumulative analysis of selected oil types on Type 6 (GEO A vs GEO C-II)

Summary hydrogen SCE test program

- The Large Engines Competence Center developed a test procedure that was shown to be suitable and reproducible to assess the impact of lubricant formulations on the occurrence of combustion anomalies in hydrogen-fueled engines.
- Hydrogen-fired engines can be more sensitive to lubricant-induced pre-ignition compared to those run-on NG at higher loads.
- As part of the strategic collaboration agreement between INNIO & ExxonMobil, a new lubricant has been developed to enable more controlled combustion at higher loads while operating with hydrogen.
- The new lubricant technology has been through a dedicated lubricant screener test on several of Jenbacher single cylinder & multi cylinder engines and was validated by INNIO the original equipment manufacturer (OEM) with excellent performance of the investigated engine oils. Using the INNIO rating the Group V based lubricants with either SAE 30 or SAE 40 (GEO C-I, GEO C-II) viscosity grades showed a significant performance advantage compared to Group II based lubricants (GEO A and GEO B), both in terms of the total number of cycles with combustion anomalies as well as the integral value of filtered MFB05%.

5 FIELD EXPERIENCE WITH OPTIMIZED OIL TECHNOLOGY FOR JENBACHER HYDROGEN ENGINES

INNIO`s hydrogen/natural gas Jenbacher engine portfolio is summarized in Figure 21.

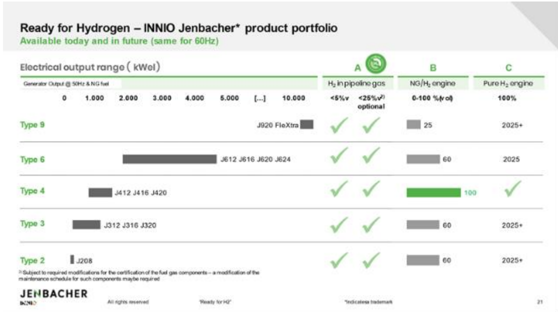

Figure 21. INNIO Jenbacher H_2 engine portfolio [39]

The experimental engine oil candidate GEO C has been field tested with regard to achievable oil drain intervals and overall piston cleanliness on the Jenbacher Type 4 engine platform. The Type 4 engine platform is H_2-Ready, considered fuel-flex capable, and optimized for alternative fuel gas types such as hydrogen, mixtures of hydrogen and methane, and pure methane. Table 5 lists field test engine parameters for Jenbacher Type 4 engines.

Table 5. Field test engine parameters for Jenbacher Type 4 engines (wood gas)

Engine	**JEN412** (wood gas)
Bore (mm)	145
Stroke(mm)	185
Engine RPM (50Hz)	1500
Average piston speed (m/s)	9.25
P_{elec} (kW)	500
P_{Therm} (kW)	513
$BMEP_{AVG}$ (bar)	~11.3
Engine Oil	GEO C-I

Figure 22 shows exemplary used oil analysis results for a field trial on Jenbacher Type 4 engines operated on wood gas. More than 6,000 hours of oil drain interval have been achieved and no condemning criteria have been reached. The OEM INNIO oil drain interval criteria also have been exceeded for the other gas qualities (NG, H_2; not shown here) tested.

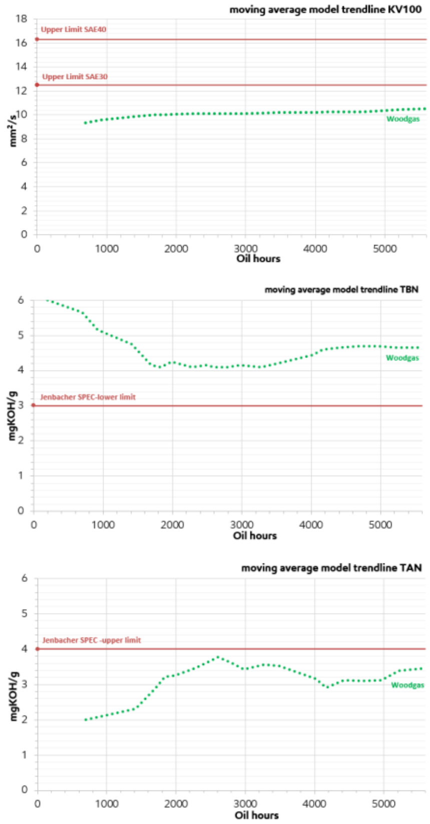

Figure 22. Field trial used oil analysis results (wood gas)

Besides achievable oil drain intervals, a key development target was to demonstrate excellent piston cleanliness. Figure 23 shows a photo documentation of Power Unit 2 of a Jenbacher Type 412 engine operated 6,000 h on wood gas.

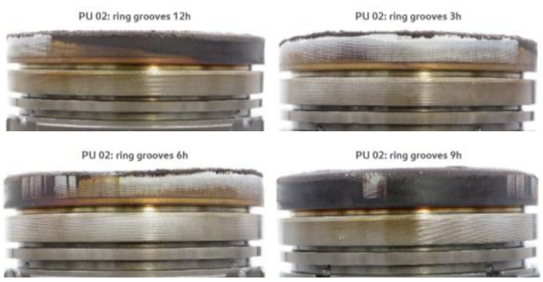

Figure 23: Field inspection results (wood gas, example Power Unit 2): piston cleanliness ring area

The investigated experimental oil candidate GEOC-I achieved an outstanding overall piston cleanliness with a rating of 90 out of 100 maximum points (average of six inspected power units). A key observation was that the first ring grove in particular was nearly free of oil coke formation, and overall combustion chamber ash formation was l on a low level in regard to INNIO Jenbacher rating.

Summary Type 4 wood gas field test program

- The new ExxonMobil development (GEO C) is the only lubricant to have successfully passed all INNIO`s screener testing, and a successful 6,000 hour field trial in a hydrogen-enriched wood gas engine.
- Piston cleanliness was excellent according to INNIO`s Jenbacher rating scale.
- No combustion anomalies were reported.
- GEO C-I also successfully passed field trials in pure NG and H_2.

6 CONCLUSIONS

The Large Engines Competence Center and INNIO Jenbacher developed a test procedure that was shown to be suitable and reproducible to assess the impact of lubricant formulations on the occurrence of combustion anomalies in hydrogen-fired engines. It was determined that hydrogen-fired engines can be sensitive to lubricant-induced pre-ignition, and lubricant selection can influence pre-ignition frequency, especially under high load conditions. As part of the strategic collaboration agreement between INNIO & ExxonMobil, ExxonMobil developed a new lubricant to enable more controlled combustion at higher loads while operating with hydrogen. The new lubricant technology has been through a dedicated lubricant screener test on several of INNIO`s Jenbacher single cylinder and multi-cylinder engines and excellent performance was measured by INNIO Jenbacher. The experimental lubricant (GEO C technology platform) with either SAE30 or SAE40 viscosity grades showed a significant performance advantage compared to other GPII based lubricants (GEO A, GEO B) tested, both in terms of the total number of cycles with combustion anomalies as well as the integral value of filtered MFB05%. The new development (GEO C) is the only lubricant to have successfully passed all INNIO screener testing for Jenbacher engines, and a successful 6,000 hour field trial in a hydrogen enriched wood gas (~20% H_2) Type 4 engine. INNIO documented excellent piston cleanliness, and no combustion anomalies have been reported. The GEO C technology platform also successfully passed field trials in pure NG and H_2. Therefore, it was selected to be the technology platform for the commercialization of an engine oil for future fuel and NG operation.

7 DEFINITIONS, ACRONYMS, ABBREVIATIONS

AFR	Air Fuel Ratio
BMEP	Brake Mean Effective Pressure

CAD	Crank Angle Degree and Computer Aided Design
CNG	Compressed Natural Gas
CR	Compression Ratio
DI	Direct Injection
EAR	Excess Air Ratio
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing
KV100	Kinematic Viscosity at 100°C
LSPI	Low Speed Pre-Ignition
MIE	Minimum Ignition Energy
NG	Natural Gas
PFI	Port Fuel Injection
PFP	Peak Cylinder Pressure
ROHR	Rate of Heat Release
SCE	Single Cylinder Engine
TAN	Total Acid Number in mgKOH/g
TBN	Total Base Number in mgKOH/g

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