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Electricity Powering Combustion: Hydrogen Engines

Sebastian Verhelst, Thomas Wallner, Helmut Eichlseder, Kaname Naganuma, Falk Gerbig, Brad Boyer and Shiro Tanno

***Abstract*—Hydrogen is a means to chemically store energy. It can be used to buffer energy in a society increasingly relying on renewable but intermittent energy or as an energy vector for sustainable transportation. It is also attractive for its potential to power vehicles with (near-) zero tailpipe emissions. The use of hydrogen as an energy carrier for transport applications is mostly associated with fuel cells. However, hydrogen can also be used in an internal combustion engine (ICE). When converted to or designed for hydrogen operation, an ICE can attain high power output, high efficiency and ultra low emissions. Also, because of the possibility of bi-fuel operation, the hydrogen engine can act as an accelerator for building up a hydrogen infrastructure. The properties of hydrogen are quite different from the presently used hydrocarbon fuels, which is reflected in the design and operation of a hydrogen fueled ICE (H_2ICE). These characteristics also result in more flexibility in engine control strategies and thus more routes for engine optimization. This article describes the most characteristic features of H_2ICEs, the current state of H_2ICE research and demonstration, and the future prospects.**

***Index Terms*—hydrogen, internal combustion engines, vehicles, (near-) zero emissions, transportation, sustainable, renewable, clean, gaseous, energy, fuel**

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I. Introduction

THIS Special Issue focuses on the intermittency challenge, for reasons outlined in the preface article, and seeks to offer an overview of the possibilities for massive scale energy storage. One of the options for converting intermittent renewable energy to storable energy, at times when energy supply exceeds demand, is to electrolyze water to hydrogen. The reasons for the interest in hydrogen as a medium for storing energy [1], and the present state and future expectations for electrolyzer technology [2], are discussed elsewhere in this Special Issue. Here, we focus on the use of hydrogen ICEs for transportation.

Producing hydrogen from excess energy, distributing it and storing it aboard vehicles also offers massive scale energy buffering, because of the very large number of vehicles worldwide. Moreover, when converting hydrogen to motive power, the main reaction product is water vapor, leading to a reversible cycle without any carbon emissions along the fuel production to fuel use path.

However, hydrogen's potential as an energy storage medium for buffering renewable energy was not the main reason for the start of research on hydrogen vehicles. Work on hydrogen as an energy carrier for transport started for three other reasons. One is the reversible cycle just mentioned, second is the diversity of energy sources that can be used to produce one and the same energy carrier, hydrogen (it goes without saying that renewable energy sources are the most attractive ones as they are sustainable and do not result in carbon dioxide [CO_2] emissions). Third is the possibility of achieving (near-) zero tailpipe emissions from hydrogen-fueled vehicles, eliminating or strongly reducing noxious emissions. Worldwide, increasingly stringent emission regulations aim at lowering emissions from vehicles such as carbon monoxide (CO), nitrogen oxides (NO and NO_2, collectively termed NO_X), unburned hydrocarbons (HC) and particulate matter (PM), to lower their harmful effects on human health and environmental degradation. However, the positive effects of these stricter regulations are partly negated by the increasing number of vehicles and vehicle miles driven. This leads to local pollution hot spots which governments are trying to address by stimulating, among other things, (near-) zero emission vehicles. One might argue that many of the reasons stated above also make a case for electric vehicles, which are actually (much) more attractive from a well-to-

wheel energy use point of view [3], [4]. Hydrogen does offer a number of advantages compared to chemical storage of electricity in batteries. First are practical aspects, hydrogen has a far superior energy storage density than batteries, thus enabling much larger driving ranges, and refueling times are quite comparable to what we are used to today. A second advantage lies in the "chemical sustainability", namely that hydrogen vehicles do not rely on the (large scale sustainable) availability of chemical elements, as opposed to batteries [5].

Much of the above and many other hydrogen-related topics are discussed at length in a previous Special Issue on the Hydrogen Economy [6]. One application that is frequently left out of the hydrogen "story" is the main subject of the present paper: the hydrogen-fueled *internal combustion engine*. Verhelst and Wallner [7] have recently reviewed the characteristics of H_2ICEs in detail. However, the paper was targeted at ICE researchers. Here, we attempt to provide a general understanding of what makes hydrogen engines attractive, what makes them different to hydrocarbon fueled engines and what the current state and future prospects are for H_2ICEs.

Hydrogen is mostly associated with fuel cells, which convert hydrogen to electricity and have the potential for high efficiencies [8]. However, when subjected to the test of chemical sustainability, fuel cells have the same disadvantage as batteries, albeit that the chemicals that limit the number of fuel cell vehicles are different from those that limit battery electric vehicles [5] (for instance platinum in the case of the proton exchange membrane fuel cell and lithium in the case of the lithium-ion battery). In contrast, the internal combustion engine on which we rely today for the bulk of vehicles worldwide, is made of abundantly available and easily recyclable materials, namely common metals. Also, being a mature technology implies low costs. Hydrogen vehicles using ICEs are less costly than either fuel cell or battery electric vehicles, in the case of the latter because of the high battery costs. Battery electric vehicles might be cheaper to run, due to their unrivalled efficiency, but for passenger cars the vehicle purchase cost is more important than the fuel (or energy carrier) cost.

Next to this, ICEs have the interesting advantage of being fuel-flexible: they can be set up to allow the combustion of different fuels within the same engine. Inevitably this leads to compromises in the engine design, but having the option of switching to another fuel is a distinct benefit, especially for introducing a new fuel such as hydrogen into the market. It is also advantageous as this enables the continued use of the vast engine manufacturing infrastructure in place today.

Finally, in the following we also hope to demonstrate the high efficiency potential of H_2ICEs, and dispel the misconception that fuel cell vehicles are several times more efficient than H_2ICE vehicles, as sometimes claimed in popular literature [9].

II. ICE BASICS

A. Working principle of the four stroke spark ignition and compression ignition engine

For a good understanding of the remainder of this article, we briefly explain the working principles of the typical gasoline and diesel engines currently used, as well as some common terminology. For the majority of road transportation, these types of engines work according to the four stroke cycle, in which four piston strokes complete an engine cycle: intake – compression – combustion&expansion (the power stroke) – exhaust, as depicted and further explained in Fig. 1.

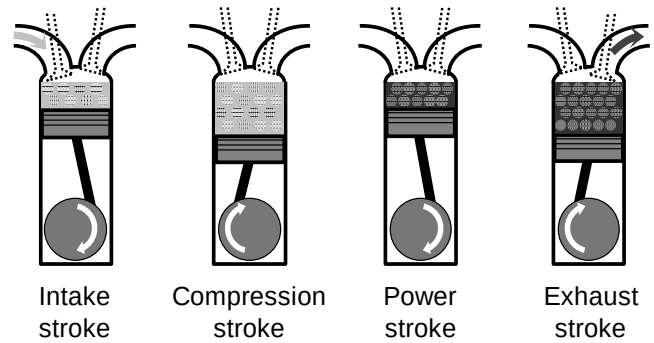


Fig. 1. 4-stroke cycle engine schematic. During the intake stroke, the intake valve (left) is open and the piston descends, drawing in fresh air (diesel engine) or air-fuel mixture (gasoline engine). When the intake valve closes, the compression stroke begins, with the piston ascending and compressing the working fluid. Combustion is initiated either with an electrical spark (gasoline) or by high pressure injection of fuel that auto-ignites in the hot compressed air (diesel). This forces the piston down and delivers work to the crankshaft through expansion of the hot gases. When the exhaust valve opens, the piston ascends again for the exhaust stroke, expelling the burned gases.

The "compression ratio" of an engine is defined by the ratio of maximum to minimum cylinder volume – i.e. the ratio of the cylinder volume when the piston has completely descended, to the cylinder volume when it has reached the end of its upward motion (see Fig. 1). These two piston positions are termed bottom and top dead center (BDC and TDC), respectively. Thermodynamic analysis suggests that the theoretical cycle efficiency (the ratio of engine work output to fuel chemical energy input) increases with compression ratio. Another important factor influencing the theoretically obtainable efficiency is the "working fluid". The properties thereof depend on its constituents: air, fuel and residuals from combustion, depending on the engine type.

When the amount of air that is admitted to the engine is such that it is just enough to fully burn the injected fuel, the engine operation is termed "stoichiometric". The actual fuel to air ratio divided by the stoichiometric fuel to air ratio is called the (fuel/air) equivalence ratio ϕ. Thus, $\phi=1$ denotes stoichiometric operation. If $\phi < 1$, more air is aspirated by the engine than is necessary for complete combustion. This is termed lean operation. The reverse, $\phi > 1$, is termed rich operation.

B. *Gasoline engine basics*

In a traditional gasoline engine, fuel is injected in the intake manifold, evaporates and mixes with air, so that a homogeneous fuel-air mixture is drawn into the cylinder during the intake stroke. Towards the end of compression, this mixture is ignited with an electrical spark. This initiates combustion, leading to a flame propagating through the combustion chamber, with burned mixture behind the flame and unburned mixture ahead of it.

Reducing the power output, for part load operation, is done by controlling the amount of (air+fuel) mixture per cycle. This amount is regulated by using a throttle valve in the intake, thus controlling the quantity of mixture allowed into the cylinders ("quantitative control"). Using a throttle valve to accomplish this results in poor part load efficiency, as the partly closed valve represents a resistance for the flow, and leads to pressure losses ("pumping losses"). Another important factor limiting the engine efficiency, is the necessity of avoiding auto-ignition of the mixture. Simply put, when the temperature in the unburned mixture exceeds the mixture's auto-ignition temperature, it will auto-ignite, leading to a very rapid rate of pressure rise and very high temperatures. In the best case, this will result in pressure oscillations, and thus vibrations, manifested as a knocking or pinging sound, hence the term "knocking" combustion. In the worst case it will severely damage the engine. Auto-ignition must thus be avoided. This effectively limits the compression ratio of the engine. As the theoretical cycle efficiency increases with compression ratio, this thus limits the efficiency.

For a gasoline engine to meet emission legislation, a three way catalyst (TWC) is placed in the exhaust, that oxidizes most of the CO and HC, and reduces NO_X (reduces in the chemical sense, i.e. converts it back to N_2 by removing oxygen). For maximum conversion efficiency of the TWC, a gasoline engine is fitted with a closed loop control system monitoring the exhaust oxygen content and adjusting the fuel to air ratio such that stoichiometric operation is maintained (ϕ=1). The conversion efficiency of a TWC when operated closely around ϕ=1 can be very high (>99%), resulting in low tailpipe emissions.

C. *Diesel engine basics*

In a diesel engine, only air is drawn into the cylinders during the intake stroke. Towards the end of compression, diesel fuel is injected directly in the combustion chamber which will auto-ignite due to the high temperature of the compressed air. Here, the combustion progress depends on the rate of injection, the rate of evaporation of the injected fuel and the rate of mixing of the fuel vapor with the air to form a locally combustible mixture. The short time available for fuel-air mixing necessitates excess air ($\phi < 1$).

The power output is now controlled by the amount of fuel that is injected (as opposed to the amount of fuel-air mixture for a gasoline engine) , which is termed "qualitative control". Thus, part load efficiency is much better than for a gasoline engine as reducing the power output does not rely on increasing the intake flow resistance with a throttle valve. Peak efficiency is also higher, because of the higher compression ratio of diesel engines (in this case actually needed to obtain sufficiently high compression temperatures to allow auto-ignition) as well as the overall lean combustion ($\phi < 1$) with which a diesel engine is operated, leading to better thermodynamic properties of the working mixture. Automotive diesel engines are now almost exclusively equipped with a turbocharging system. This consists of a turbine wheel in the exhaust manifold, which is powered by the energy still contained in the exhaust gases, itself driving a compressor wheel in the intake. Thus, the intake air is compressed leading to a higher specific power output of the engine as more fuel can then be injected per cycle.

The non-homogeneous nature of the diesel cycle, with large variations in local fuel-air ratios, leads to quite high engine-out emissions of PM. Engine-out NO_X emissions are comparable to gasoline engines. Unfortunately, because of the overall lean operation and relatively low exhaust temperatures, effective aftertreatment is difficult to achieve. The overall lean operation leads to exhaust gases containing oxygen, which makes it difficult to reduce the NO_X to N_2. The relatively low exhaust temperatures lead to low catalyst activity. Thus, tailpipe emissions are generally higher than for gasoline engines.

Due to the better part load efficiency, the CO_2 emissions are around 20 % lower and the fuel economy is typically 25-30 % better than for gasoline engines. For diesel engines, a peak efficiency of 44% is currently the state of the art, with 36% achieved for gasoline engines. The specific power output of diesel engines is increased by turbocharging and reaches a level up to 75 kW per liter engine displacement (e.g. passenger car: 3.0 L, 225 kW).

D. *Present and future developments*

From the previous section it can be concluded that gasoline engines score well in terms of emissions (related to local air quality) but have a relatively poor efficiency (related to CO_2 emissions, thus global warming), whereas diesel engines are more efficient but generally more polluting. The more advanced engines currently in the market, and new engine technology now being developed, try to improve the weaker points while keeping the assets. For instance, gasoline engine development is targeted primarily at improving part load efficiency, mostly through minimizing throttle losses. Measures taken include variable valve timing and lift, exhaust gas recirculation (EGR) and "downsizing" of the engine, using a smaller engine while keeping the peak power. For a given load, the smaller engine effectively needs to work harder, asking for larger throttle openings resulting in a better efficiency. Keeping the same peak power as the larger engine is enabled through turbocharging (i.e. increasing the pressure of the intake air, resulting in more mass per cycle, see above). This will inevitably increase in-cylinder temperatures, which increases the danger for auto-ignition (knock). This can be counteracted by decreasing the compression ratio of the

engine, but as mentioned above decreases the engine efficiency. Modern engine technology allows the compression ratio to be kept quite high, through either direct injection, variable valve timing, or a combination thereof. These measures can be used to lower in-cylinder temperatures. Thus, there is indeed an efficiency benefit.

Diesel engine development is targeted mainly towards reducing emissions of NO_X and PM by promoting better mixing of fuel and air (very high injection pressures, multiple injections enabled by common rail systems, high charging pressures increasing the amount of excess air), reducing peak temperatures (e.g. by advanced exhaust gas recirculation) and new aftertreatment systems.

In the next years we will see a specific power of close to 100 kW/L, both for gasoline and diesel passenger car engines, which will be used for "downsized" applications. This will, in combination with "electrification" (electrically supported ICE) in hybrid vehicles allow for a 30-40 % improvement in fuel economy at the same vehicle performance.

Gasoline engines will, through continued effort, achieve extremely low pollutant emissions levels. Whether diesel engines will achieve this goal is still an open question.

III. Hydrogen as an ICE fuel

A. *Fuel properties and resulting mixture formation and load control strategies*

The properties of hydrogen are drastically different from those of conventional engine fuels such as gasoline or diesel, see Table I. Unlike those fuels hydrogen does not contain any carbon and therefore combustion of hydrogen does not create carbon-containing emissions such as CO_2, CO, HC or PM.

Also, the mass-specific lower heating value of hydrogen is almost 3 times as high as that of gasoline or diesel. However, hydrogen is gaseous at ambient conditions with a density that is several orders of magnitude lower than that of liquid fuels. Increasing the energy storage density is a topic of continuing research but outside of the scope of this paper. Currently used storage systems are either pressurized (commonly used storage pressures are 20 and 35 MPa, with 70 MPa already demonstrated) or liquefied (requiring cooling to -253 °C). Other possibilities to store hydrogen are metal hydrides or liquid organic hydrogen carriers [10].

TABLE I
TYPICAL PROPERTIES OF GASOLINE, DIESEL AND HYDROGEN

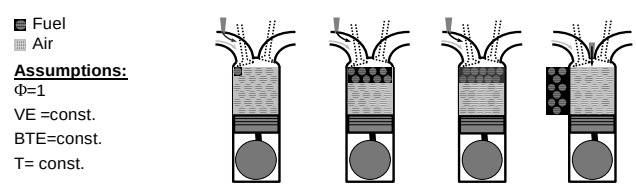
Parameter	Unit	Gasoline	Diesel	Hydrogen
Chemical formula		C4 - C12	C8 – C25	H2
Composition (C, H, O)	Mass-%	86, 14, 0	87, 13, 0	0, 100, 0
Lower heating value	MJ/kg	42.7	42.78	120
Density	kg/m^3	720 - 780	848	0.089
Stoichiometric air/fuel ratio	kg/kg	14.7	14.5	34.3
Flammability limits	Vol-% ϕ	1.0-7.6 0.7-2.5	1.0-6 0.7-2.1	4-75 0.1-5
Laminar flame speed	cm/s	35-50	na	210
Auto-ignition temperature	°C	approx. 350	approx. 265	585

Properties given at standard temperature (273.15 K) and pressure (101325 Pa)

The flammability limits of hydrogen are extremely wide, spanning from 4 to 75 Vol-% (of hydrogen in air), or in equivalence ratio terms: $0.1 < \phi < 5$, thus making hydrogen ideally suited for lean burn operation ($\phi < 1$). Also, the laminar flame speed, a physicochemical mixture property affecting the fuel burn rate, of a hydrogen-air mixture at ambient and stoichiometric conditions is by a factor of 4 to 6 higher than that of gasoline. At lean conditions around $\phi \sim 0.5$ the laminar flame speed of hydrogen is approximately equivalent to that of a stoichiometric gasoline-air mixture [7].

As outlined earlier, diesel engines rely on the temperatures at the end of the compression stroke to be high enough to ignite the injected fuel. Since the auto-ignition temperature of hydrogen is significantly higher than diesel fuel, it is not easily reached with conventional compression ratios. Therefore hydrogen is better suited for spark ignition operation rather than compression ignition, although the latter is not impossible and can have its advantages, as explained in Section V-C.

The low density of hydrogen has important implications on the attainable power density with various mixture formation concepts. Fig. 2 shows hydrogen mixture formation strategies compared to conventional gasoline operation that is shown on the left hand side of the figure. The schematic outlines how fuel and air are introduced into the combustion chamber for different mixture formation strategies. When comparing engines with external mixture formation (port fuel injection - PFI), the lower density of hydrogen results in a lower power output compared to liquid fuels, because the large volume of hydrogen gas displaces air in the engine intake. This is somewhat compensated by the stoichiometric air to fuel ratio of hydrogen, which is about twice that of conventional fuels. Thus, less fuel mass is required for stoichiometric combustion of a given amount of air. Also, the lower heating value of hydrogen is roughly thrice that of conventional fuels. Nonetheless, the theoretical power density (kilowatts of power per unit engine displacement) of a hydrogen engine with port fuel injection is approx. 18% lower than that of a comparable gasoline engine [11].



Fuel	Gasoline	H_2	H_2	H_2
Mixture formation	*external*	*external*	*external cryogenic*	*internal*
Mixture temperature [K]	293	293	210	293
Mixture calorific value [MJ/m³]	3.6	3.0	4.15	4.22
Power potential [%] (compared to gasoline)	100	82	115	117

Fig. 2. Influence of mixture formation concept on full load potential. The figures show where the fuel is introduced (external of the cylinder or internal), the dark grey indicates fuel, the light grey indicates air. The number of "spheres" represents the density. The assumptions for this comparison include stoichiometric operation (Φ=1), constant volumetric efficiency (VE) meaning that the same volume of fresh charge is aspirated, constant engine efficiency (BTE) and constant ambient temperature (T).

Because of its relative simplicity and availability of injection equipment, the strategy is widely used for hydrogen engines (see Section IV). Both, cryogenic external mixture formation of hydrogen from liquid fuel storage as well as hydrogen direct injection (DI) allow for increased theoretical power density compared to gasoline, as seen in Fig. 2. However, both concepts require complex injection equipment that either has to operate reliably at extremely low temperatures or increased injection pressures. Both mixture formation concepts are currently at a research stage using prototype equipment.

The most common load control strategy for hydrogen internal combustion engines is quantitative load control (Fig. 3 left), i.e. regulating the amount of mixture using an intake throttle, while keeping the fuel to air ratio constant. If operated at stoichiometric conditions, this strategy is identical to what is used in conventional gasoline engines with the same advantages, such as simplicity and effective aftertreatment options, as well as disadvantages including lower part load engine efficiencies. This load control strategy can be applied for stoichiometric operation (ϕ=1) or lean operation (fixed ϕ lower than one) due to the wide flammability limits of hydrogen. In both cases a throttle is used and the relative air fuel ratio remains constant over the entire engine operating regime. Lean burn concepts with quantitative load control show advantages in part load engine efficiencies, however, aftertreatment options are limited (depending on the equivalence ratio aftertreatment may not be necessary, see further below) and the power output is typically significantly lower than a comparable gasoline engine.

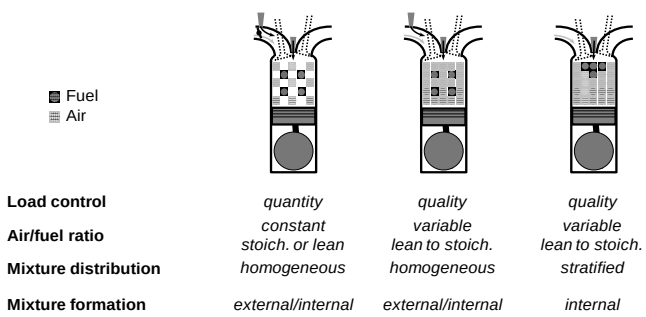


Fig. 3. Load control strategies in hydrogen operation. The number of "spheres" represents the density. Thus, in the left figure the density of mixture entering the cylinder is lower, because of the throttle resistance. The dark grey areas denote fuel, the light grey areas air. In the two figures on the left, fuel and air are evenly distributed, while in the right hand figure fuel is concentrated around the injector (stratified operation).

In contrast to that, qualitative load control takes full advantage of the lean burn ability of hydrogen and avoids the intake throttle losses [12]. The relative air fuel ratio is adjusted depending on engine load while the intake air flow is unthrottled, similar to a diesel engine [13]. This can either be done with using homogeneous (Fig. 3 center) or inhomogeneous – "stratified" (Fig. 3 right and Fig. 4) mixture conditions. Both strategies show advantages due to increased part load efficiencies. However, lean burn operation also limits the ability to employ a simple aftertreatment system.

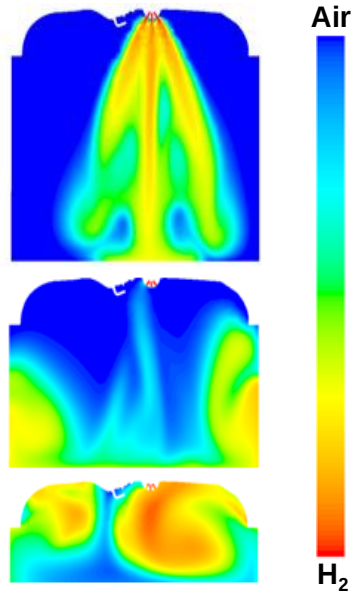


Fig. 4. Development of a stratified mixture with direct injection, from injection (top), through compression (middle) to spark timing (bottom). The hydrogen molar fraction calculated using 3-D CFD (Computational Fluid Dynamics) is shown with red areas depicting zones with high hydrogen content and blue zones representing areas with mainly air.

B. Emissions from hydrogen engines

The critical emissions component for hydrogen internal combustion engines are oxides of nitrogen (NO_X). Unlike hydrocarbon (HC) and carbon monoxide (CO) emissions which are the result of incomplete combustion with conventional fuels, NO_X emissions are a result of using air as the oxidizer (containing nitrogen, besides oxygen) and high combustion temperatures. Fig. 5 shows the well-established trend [14] - [16] for NO_X emissions as a function of fuel to air equivalence ratio, with homogeneous hydrogen operation.

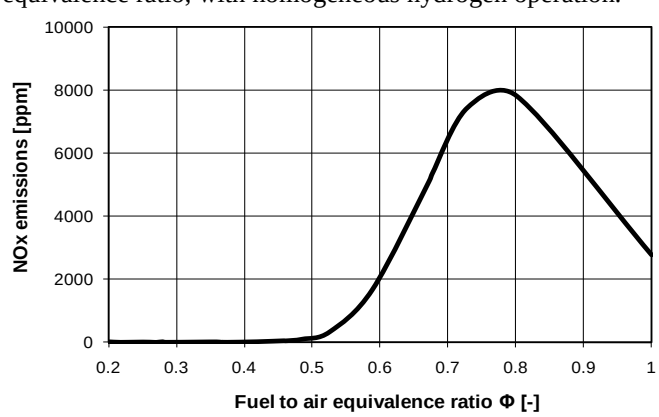


Fig. 5. NO_X emissions in ppm with homogeneous hydrogen operation

At low fuel to air equivalence ratios the NO_X emissions levels are negligible. With increasing ϕ, the combustion temperatures increase, resulting in NO_X formation. At a threshold around ϕ ~ 0.5 the NO_X emissions increase exponentially and reach their peak around ϕ ~ 0.8. Further

increasing ϕ results in a decrease in NO_X emissions due to reduced availability of excess oxygen. This trend is applicable with engines using homogeneous mixture formation, employing qualitative load control. With increasing load the fuel to air equivalence ratio increases and NO_X emissions follow the trend of Fig. 5. This has implications for the load strategies used in hydrogen vehicles, as will be explained in Section IV.

C. *Combustion anomalies*

The wide flammability limits and high flame speeds listed in Table I, combined with a low required ignition energy, are beneficial for the engine efficiency, as explained above and further illustrated below. However, these properties can also result in undesirable combustion phenomena, generally termed combustion anomalies.

Much of the early work on H_2ICEs was targeted at trying to avoid these anomalies. The most frequently cited anomaly is called 'backfire', which happens when the hydrogen-air mixture already ignites during the intake stroke. This results in a combustion event in the intake. In the best case, this is manifested by a loud 'bang', in the worst case it results in damage of the intake. Care must thus be taken to prevent unwanted ignition sources such as hot spark plug electrodes, exhaust valves etc., which implies some hardware modifications to the engine. Also, avoiding the presence of ignitable mixtures when there might be an ignition source is a means to prevent backfire. Thus, variable valve timing and a carefully controlled injection timing have been used to avoid backfire occurrence [17], [18] by allowing a cooling phase with fresh air while limiting the presence of hydrogen during the early intake phase. Directly injecting hydrogen in the combustion chamber, after the intake valves have closed, avoids backfire completely and is one of the advantages of DI engines.

Unwanted ignition can also occur when the intake valves are already closed. When this happens before the spark ignition, this is termed 'pre-ignition'. As the valves are closed, and the cylinder charge is being compressed, this can lead to very high pressures. The potential damage is thus much more severe.

Finally, as for any spark-ignition engine, auto-ignition of the unburned mixture (mostly termed engine 'knock') must be avoided, as explained in Section II-B. Because of the high auto-ignition temperature of hydrogen (see Table I), 'knock' is less likely to occur. However, if it occurs it can be severe, due to the high burning velocity of hydrogen mixtures.

IV. H_2ICE VEHICLES

Most H_2ICE prototype and demonstration vehicles employ hydrogen port fuel injection for mixture formation. The resulting air fuel mixtures are homogeneous and the NO_X emissions follow the trend shown in Fig. 5. The engines used in these demonstration vehicles are typically converted or derived from gasoline engines and in some cases dual-fuel operation is maintained.

Some of the earliest modern H_2ICE vehicles were built by Musashi Institute of Technology (now Tokyo City University) starting with Musashi 1 in 1974 employing a 4-stroke engine and 7 Nm^3 (cubic meter at standard temperature and pressure) compressed hydrogen storage. This vehicle was followed by several other prototype vehicles employing 2-stroke as well as 4-stroke engines and compressed and cryogenic storage [19], [20].

In 2001 Ford Motor Company revealed their Ford P2000, the first North American production viable hydrogen prototype vehicle with 1.5 kg compressed hydrogen storage and a 2.0 L inline 4 cylinder engine. The prototype vehicle was tested with different calibrations including a constant air fuel ratio lean burn strategy (quantitative control) at $\phi \sim 0.55$, an operating strategy with full load enrichment up to $\phi \sim 0.7$ as well as a largely unthrottled calibration that was only demonstrated through engine dynamometer testing [21]. Starting in 2004 Ford offered a Hydrogen Shuttle Bus for leasing, based on the E-450 bus, see Fig. 6. The 8-12 seater shuttle bus is powered by a 6.8 L V-10 Triton engine using supercharging and a constant equivalence ratio strategy. The fully engineered demonstration vehicle features a compressed hydrogen tank system operating at 35 MPa with a total capacity of 29.6 kg allowing for a vehicle range of 240-320 km [22] - [25].

Fig. 6. Photograph of a Ford shuttle bus powered by a 6.8 L V-10 hydrogen engine

With approximately 100 units built the BMW Hydrogen 7, see Fig. 7, is likely the hydrogen vehicle with the highest number of vehicles produced. The BMW Hydrogen 7 features a 6.0 L V-12 engine with hydrogen port fuel injection that is based on BMW's production 12-cylinder engine for the BMW 760 sedan. The car employs a variable air fuel ratio lean burn strategy (qualitative control) at low and medium engine loads and switches to stoichiometric operation (quantitative control) at high engine loads. This strategy avoids the NO_X critical operating regime with air fuel ratios $0.5 < \phi < 1$ (see Fig. 5) and a catalyst is used to reduce NO_X emissions in stoichiometric operation. Approximately 8 kg of hydrogen

(equivalent to 8 gallons of gasoline in terms of energy content) are stored on-board the vehicle in a cryogenic tank that is located in the trunk allowing for a range of 200 km on hydrogen. The vehicle is equipped with a dual-fuel system allowing for operation on hydrogen or gasoline with an additional range of 480 km on the conventional fuel [26]. Emissions tests of a dedicated mono-fuel version of the Hydrogen 7 vehicles showed the low emissions potential of hydrogen powered vehicles. With a dedicated aftertreatment design featuring two catalysts (one for stoichiometric operation and one for reducing NO_X peaks that occur when switching from lean to stoichiometric operation) the vehicle achieved drive-cycle NO_X emissions that were approximately 0.0008 g/mi, which is equal to 3.9% of the Super Ultra Low Emissions Vehicle (SULEV) limit [27].

Fig. 7. Photograph of a BMW 7 series sedan powered by a 6.0 L V-12 hydrogen engine

Mazda has pursued a different route of combining hydrogen and combustion engines over the last 20 years presenting their first hydrogen rotary engine vehicle prototype, called Mazda HR-X, in 1991 [28]. A rotary engine, originally designed and developed by Dr. Felix Wankel, and thus sometimes called a Wankel engine, differs from a reciprocating engine. It uses a triangular rotor that moves on eccentrics in a specifically designed housing instead of using a piston and crank mechanism. Due to the rotor and housing design and the eccentric motion the four strokes known from a reciprocating engine (see Fig. 1) occur in separate areas. While it takes a reciprocating 4-stroke engine 2 engine revolutions to complete the intake, compression, power and exhaust stroke, a rotary engine completes three power strokes per rotor rotation. Typically, an additional gear ratio of 3:1 is used between the rotor and the input shaft of the transmission, so that one rotor of a rotary engine has one power stroke per rotation. While the specific power of rotary engines is therefore typically higher than that of comparable reciprocating engines, challenges facing the rotary engine include durability of the face seals between rotor and housing as well as disadvantages due to non-ideal shape of the combustion chamber. The unique properties of hydrogen, namely high flame speeds and short flame quenching distance, help mitigate the challenges due to the combustion chamber shape. In addition, there are intrinsic advantages of rotary engines in combination with hydrogen such as the local separation of intake, compression, power and exhaust stroke which help reduce occurrence of combustion anomalies as well as the less stringent demands for injectors when using direct injection. Due to these compelling reasons Mazda has presented several generations of hydrogen rotary engines. The most recent generation of hydrogen rotary engine from Mazda is called RENESIS Hydrogen RE, provides a maximum power of 80kW and also sports a dual-fuel system. The engine has most recently been integrated in the Mazda Hydrogen Rotary RX-8 allowing for a vehicle range of approx. 100 km storing 105 L of hydrogen on-board at 35 MPa as well as the Mazda Premacy Hydrogen RE Hybrid, a 5 passenger vehicle with a peak power of 110 kW and a range of 200 km using a 150 L 35 MPa tank [28], [29].

From the description of demonstration vehicles above, it can be seen that the current driving ranges are still rather limited. This is partly due to the current hydrogen storage methods [10], but is also due to the use (primarily) of (ambient) hydrogen port fuel injection, which does not lend itself to exploiting the properties of hydrogen for maximizing engine efficiency. The present H_2ICE research is mostly focused on increasing efficiency, as explained in the following section.

V. H_2ICE RESEARCH

A. *Advanced port fuel injection research*

As outlined earlier, port fuel injection is the predominant mixture formation strategy for hydrogen prototype and demonstration vehicles. Both, cryogenic hydrogen injection and hydrogen direct injection have demonstrated promising results in laboratory experiments and research settings. Their application in demonstration vehicles has so far been limited by the lack of suitable, production-ready injection equipment.

Challenges for cryogenic injection are related to the low boiling temperature of liquid hydrogen (-253 °C) and the resulting extremely low temperatures of even the gaseous phase fuel. Care must thus be taken to avoid the occurrence of injector icing. Research has been performed in an attempt to take advantage of the related cooling effects. As outlined in Fig. 2, injection of cryogenic, gaseous hydrogen into the intake manifold significantly lowers the temperature, thus increasing the density of the air/fuel mixture and therefore results in higher power output. Calculations based on a verified model suggest that the trapped air mass per cycle can be increased by up to 16 % with cryogenic hydrogen port injection (injection temperature around 90 K) compared to ambient hydrogen injection [30]. Besides an increase in attainable power output, cryogenic injection also leads to a higher efficiency in the medium and upper part-load regions, see Fig. 8 [31].

B. *Hydrogen direct injection*

Hydrogen direct injection on the other hand poses challenges

due to injector leakage and durability. Avoiding leakage of hydrogen at pressures in the range of 10 MPa with the injector nozzle exposed to combustion temperatures and pressures is everything but trivial. It is estimated that the injector tip of a hydrogen injector can easily reach 300 - 400 °C. Furthermore, hydrogen injectors for late direct injection and multiple injection applications (see below) are required to have a control range of up to 1:16 (ratio of minimum to maximum amount of fuel that can be delivered by a single injection) at minimum injection durations around 0.1 ms while operating at a complete absence of lubrication [32]. Life times of 200 hours are reported for current injector prototypes with a short term target to extend the durability to 1,000 hours which would allow for multi-cylinder demonstrations. A longer term goal of 20,000 hours with a production-oriented design has also been targeted [33].

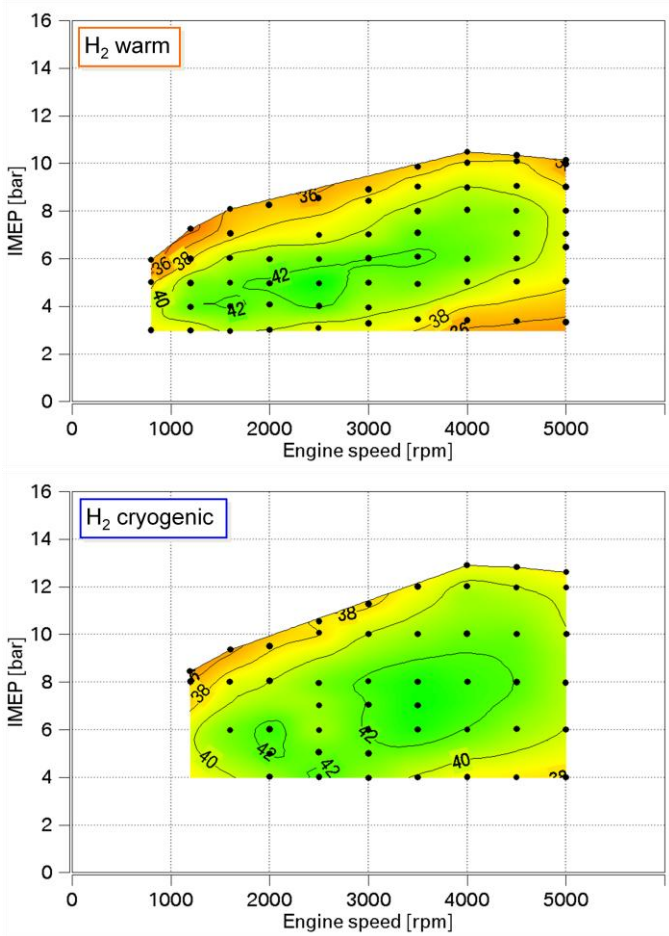


Fig. 8. "Maps" of indicated efficiency, shown by contour lines of iso-efficiency (with green indicating higher efficiency numbers, while red stands for lower numbers). The indicated efficiency is the ratio of work delivered to the piston to the fuel energy input. The actual engine efficiency will be lower because of friction losses. However, we want to demonstrate the qualitative differences between ambient temperature ("warm", top figure) hydrogen injection and cryogenic injection (bottom figure). The maps show the operational range of an engine, by plotting indicated mean effective pressure (IMEP), a measure of the engine load, against engine speed. Note how the peak efficiencies are the same for both concepts, but that the peak efficiency can be obtained in a larger region of the map in the case of cryogenic injection. Also note the higher peak loads that are achievable with cryogenic injection (higher IMEPs).

Hydrogen direct injection mixture formation systems can be classified based by injection strategy, injection pressure and number of injection pulses. Fig. 9 illustrates the most common hydrogen direct injection strategies graphically.

Typically hydrogen direct injection is performed after the intake valves are closed to avoid displacement of intake air (reduced power) and backfiring. Early injection refers to strategies with start of injection (SOI) shortly after intake valve closing typically resulting in fairly homogeneous mixtures at spark timing. The required injection pressures for early injection are in the range of 0.5 - 2 MPa. On the other hand, late injection against higher in-cylinder pressure at the end of the compression stroke requires higher injection pressures of up to 10 MPa and typically results in rather stratified mixtures. Finally, multiple injection strategies with pulses during the compression as well as the combustion phase require injection pressures between 10 and 30 MPa.

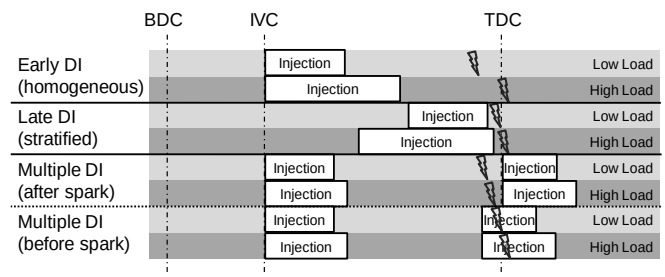


Fig. 9. Schematic of H_2 DI strategies. Typically, the intake valve closing (IVC) is after bottom dead center (BDC). Injecting after IVC avoids the occurrence of backfire and allows a maximum amount of air to be aspirated into the engine.

The mixture stratification that results from a late injection strategy can be exploited to improve the typical trade-off between efficiency on the one hand and NO_X emissions on the other hand. As shown in Fig. 5, the fuel to air equivalence ratio is the dominating factor for NO_X emissions in homogeneous operation. Apart from the overall equivalence ratio, mixture homogeneity was found to also strongly influence NO_X emissions levels [34]. A homogeneous mixture at an equivalence ratio ϕ~0.8 results in peak NO_X emissions (see Fig. 5), whereas a stratified mixture with the same global air fuel ratio will result in lower NO_X emissions as the local equivalence ratio will be either left or right of the peak. Thus, the stratification of the fuel-air mixture that is possible with DI allows both lowering NO_X emissions and increasing engine efficiency, compared to PFI. The increase in efficiency is also due to a reduction of heat losses to the cylinder walls. Finally, when combining DI combustion with EGR, even higher efficiencies and lower NO_X emissions are possible [35].

Recent research activities with hydrogen direct injection suggest that peak engine efficiencies around 45 % are achievable. Experimental results on a single-cylinder research engine with simulated turbocharged operation at an engine speed of 3000 RPM and full load resulted in an estimated brake thermal efficiency of almost 44 % at very low NO_X emissions levels [36].

C. Diesel-like hydrogen operation

As explained above, the compression ratio of spark ignition

engines is limited to avoid knocking and this implies a thermodynamic limit for efficiency. Looking for means past this limit, concepts for enabling diesel-like combustion with hydrogen were investigated. If combustion with very late direct injection of hydrogen was possible, the risk of knock could be removed and thus the compression ratio raised. However, in order to benefit from an increase in efficiency, heat losses of the flame to the cylinder walls must be minimized at the same time, as otherwise they would more than compensate for the benefit of the raised compression and could even lead to a loss of efficiency [37]. It is possible to counter this risk by means of appropriate combustion chamber and injector geometry design. Figure 10 shows a H_2 single cylinder research engine, which was developed for a Diesel-like combustion system for a hydrogen ICE.

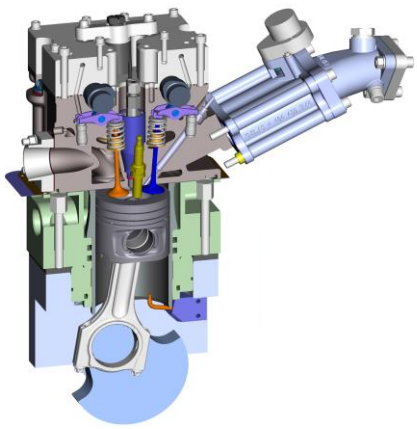


Fig. 10. H_2 single cylinder research engine with Diesel configuration [38] with characteristic flat cylinder head and piston bowl.

The wide flammability limits of hydrogen in air suggest that this fuel is suitable for a diesel combustion process. However, the main challenge for realizing compression ignition arises from the high spontaneous ignition temperature of hydrogen, 585°C (Table I). Assuming direct injection at ambient temperature, the temperature at the end of compression would need to be around 1100 K in order to guarantee sufficient heat input [38]. Simply raising the compression ratio does not allow such temperatures to be achieved. Additional measures such as inlet air preheating would be necessary, which are ruled out for practical applications [39]. Overall, compression ignition operation has limited use in passenger cars even with high levels of inlet air preheating, because it is insufficiently controllable and limited to low engine loads [39].

The passenger car application requires an operating strategy that retains the concept of non-premixed combustion and yet, in terms of ignition, moves away from the conventional diesel cycle combustion process. One such possibility is the use of surface ignition by means of a glow plug. Here, following late direct injection, the outer boundary of the cloud of injected hydrogen, which is already mixed with air, is ignited by contact with the surface of a hot glow plug [37]. The resulting combustion proceeds in a way comparable to a conventional diesel engine in the cold start phase.

An H_2 combustion process such as this is very robust. It is possible to avoid measures such as inlet air pre-heating, and combustion anomalies are practically excluded. In addition, the combustion process is compatible with almost unlimited supercharging and can cover the entire operating range relevant to passenger car applications. The documented efficiency based on measurements on the engine shown in Fig. 10, assuming friction losses of a modern passenger car diesel engine and a turbocharger optimized for the operating point, turns out to be at the same level as the most efficient passenger car turbo-diesel engines, approximately 42% [37].

Another approach to obtain diesel-like hydrogen operation is to use multiple injections. First, a limited amount of hydrogen is injected relatively early resulting in a lean but flammable mixture, and is spark ignited. Then, during the combustion of the initially injected hydrogen, a second injection delivers hydrogen that will burn in a diffusive (non-premixed), diesel-like way (see Fig. 9, “multiple DI (after spark)”). A study on a multi-cylinder engine modified for single-cylinder hydrogen operation concluded that stratified and diffusive combustion showed a 3 % improvement in engine efficiency compared to homogeneous combustion with conventional flame propagation. The same study also suggests that engine efficiencies beyond 45 % are achievable with optimized injection equipment, charging strategy and valve timing [35]. Since no carbon is present in hydrogen the risk of particulate formation due to the non-premixed combustion is eliminated.

VI. Conclusion

The three main reasons for a hydrogen economy to be so desirable consider that (a) a reversible cycle can be established, with hydrogen made from water and turning back into water when used; (b) a variety of (renewable) sources can be used to make hydrogen and (c) the conversion of hydrogen into other forms of energy results in (near-)zero emissions. Further advantages for the case of hydrogen powering an internal combustion engine include combustion engines to be an extremely mature technology and their producibility from recyclable materials which both reduce cost. In addition the internal combustion engine route opens up a multi-fuel option where hydrogen is used when available and another fuel is used when it is not. Finally, this article clearly makes a case for the high efficiency potential of hydrogen internal combustion engines.

The unique properties of hydrogen create certain challenges, such as combustion anomalies, but also open up the potential to increase the power output by more than 15% compared to conventional gasoline engines. Further, the wide flammability limits and high flame speeds even under lean conditions enable the use of highly efficient lean-burn engine load control strategies. At this point, prototype and demonstration vehicles, such as the Ford Hydrogen Shuttle bus or the BMW Hydrogen 7 take some advantage of the unique properties of hydrogen while demonstrating the (near-) zero emissions potential. In a research environment, advanced mixture formation concepts such as cryogenic injection or direct injection, that more fully exploit the possibilities due to

the specific properties of hydrogen, have been proven to achieve engine efficiencies in excess of 45% which exceed those currently achieved with gasoline or diesel.

Further development for hydrogen internal combustion engines is needed in several areas including development of injection equipment that meets automotive durability requirements, assessment of efficiency potential of advanced mixture formation concepts on multi-cylinder engines, as well as demonstration of real-world fuel economy and tailpipe emissions of advanced prototype vehicles. In addition to those areas related directly to engine research and development, critical improvements in on-board hydrogen storage density have to be achieved. Finally, the dual-fuel capability of hydrogen internal combustion engine vehicles might help to overcome the chicken-and-egg problem regarding the timing of introduction of infrastructure versus vehicle deployment.

APPENDIX

BDC	bottom dead center
BTDC	before top dead center
CA	crank angle
CFD	computational fluid dynamics
CO	carbon monoxide
CO_2	carbon dioxide
DI	direct injection
EGR	exhaust gas recirculation
ϕ	fuel/air equivalence ratio
HC	hydrocarbons
H_2ICE	hydrogen-fueled internal combustion engine
ICE	internal combustion engine
IMEP	indicated mean effective pressure
IVC	intake valve closure
NO_X	oxides of nitrogen
PFI	port fuel injection
PM	particulate matter
RPM	revolutions per minute
TDC	top dead center
TWC	three way catalyst

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