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A Layout for the Hydrogen Engine

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ABSTRACT

Hydrogen combustion leaves no carbon footprint and holds high promise for energy production. It can be used in a fuel cell, H_2-FC, or an internal combustion engine (ICE). The net lifetime cost of the H_2-ICE is less than that of the H_2-FC, making it attractive for off-road and heavy-duty on-road applications. Physical and chemical properties such as diffusivity, flame speed, ignition energy and ignition temperature are favourable for high-speed operation and high compression ratio. Low density makes storage and transport of the fuel cumbersome, and high flame temperature at stoichiometric composition results in excessive NOx compared to gasoline. The popular three-way catalyst used in gasoline engines is not applicable to quell NOx. The other option is to run the fuel very lean at an air excess ratio, $\lambda = 2.2$. To produce respectable power, the lean burn engine must be turbocharged and intercooled. Direct injection is preferred due to the high diffusivity of hydrogen. A prechamber with the spark plug and fuel injector mounted inside is beneficial in creating high turbulence in the main chamber by the high-velocity hot jets and providing a multitude of ignition points. It achieves leaning to λ=2.2. It could further help in reducing backfire as the raw fuel may not stay very close to the walls or small gaps of the main combustion chamber.

1 Introduction

The exhaust from internal combustion engines (ICE) burning hydrocarbon fuels is responsible for 21% of the total carbon dioxide emissions contributing toglobal warming [1].Trucks and buses probably are responsible for 30% of the total CO_2 emissions by ICEs [2]. Therefore, hydrogen as a fuel in an internal combustion engine, H_2-ICE or a fuel cell, H_2-FC is attractive because of nil carbon dioxide emissions. The maximum achievable efficiency of fuel cells could be as high as 60% against 45% for an engine, Figure 1a. Interestingly, while the efficiency of an H_2-ICE increases with load like all other ICEs, it drops for H_2-FC[3, 4].

Heavy-duty truck engines are more powered per tonnage in Europe and the USA than in India and hence run at a relatively lower average load of 31%. In the context of India, the average load could be higher than 60% [5] since the vehicle tonnage to rated-power ratio is designed higher for better fuel economy and a maximum speed of 80 km/h. The difference in efficiencies of H_2-ICE and H_2-FC narrows down at such high loads, Figure 1a. In addition, a fuel cell vehicle in conjunction with an electric motor, battery and accompanying electronics is more expensive and heavier than an ICE truck. Also, it asks for very pure hydrogen which is expensive,unlike an ICE.The ICE technology is already well-developed for applying liquid and gaseous hydrocarbon fuels; therefore, the new hydrogen engines can be produced at a fraction of the cost of H_2-FC. The difference in costs of fuel and the original equipment makes a diesel truck [6] more economical than an H_2-ICE even after five years of operation, assuming the costs of fuelsare pegged at 2022 levels, in the European context (Figure 1b). The total ownership cost (TOC) of H_2-ICE falls between the expensive H_2-FC and more economic diesel trucks. The TOC of diesel trucks for five years of operation is expected to match that of H_2-ICE in the near future,with the possible increase in the price of diesel from 1.7 euro/litre to 2.5 and a decrease in the price of hydrogen.

2 Basics of H_2Combustion

2.1 Chemistry

2.1.1 Flame Velocity, Flame Temperature and NOx (Gasoline, Natural Gas, Hydrogen)

To sustain a flat flame static in space, the unburned mixture of fuel and air at given pressure and temperature must rush steadily at a certain velocity towards the flame, known as the flame velocity. For laminar flows, the behaviour of flame velocity ofiso-octane with respect to fuel equivalence ratio, φ at different pressures is shown in Figure 2a.With the increase in pressure, the velocity decreases. The maximum for any given pressure is when the air-fuel mixture is nearly stoichiometric and drops off when the mixture is leaner or richer.

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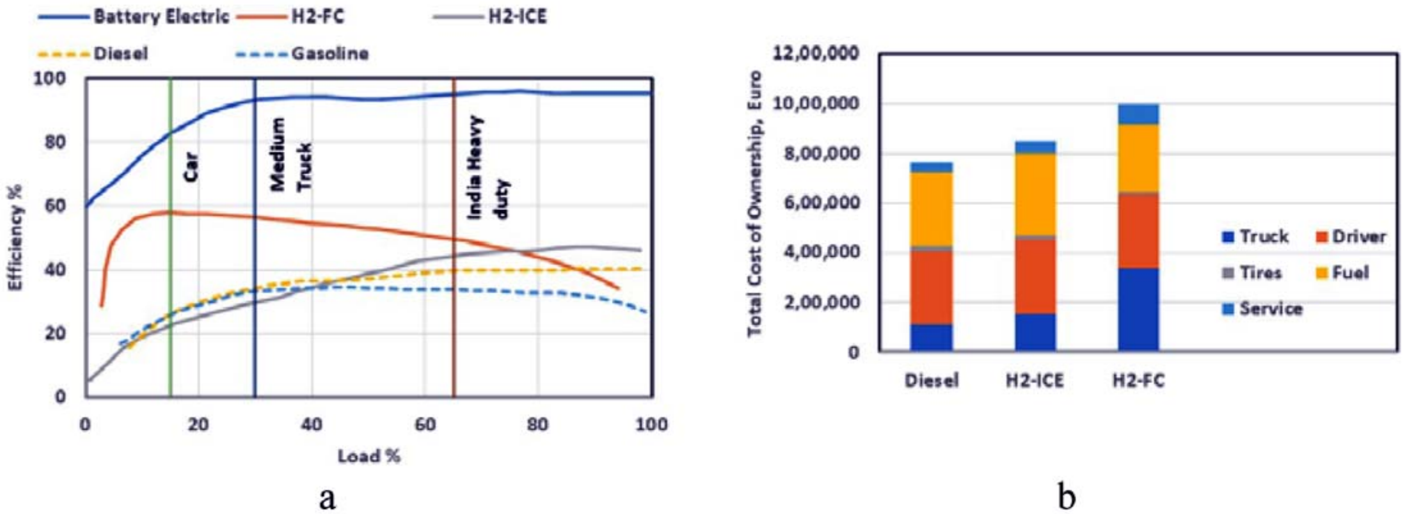


Figure 1 Economics of different power sources of heavy-duty engines: (a) Brake Thermal Efficiency of vehicles as a function of load: battery electric, fuel cell, H_2-ICE, diesel and gasoline [3, 4], (b) Total cost of ownership of Diesel, H_2-ICE and H_2-FC over five years of operation [6]

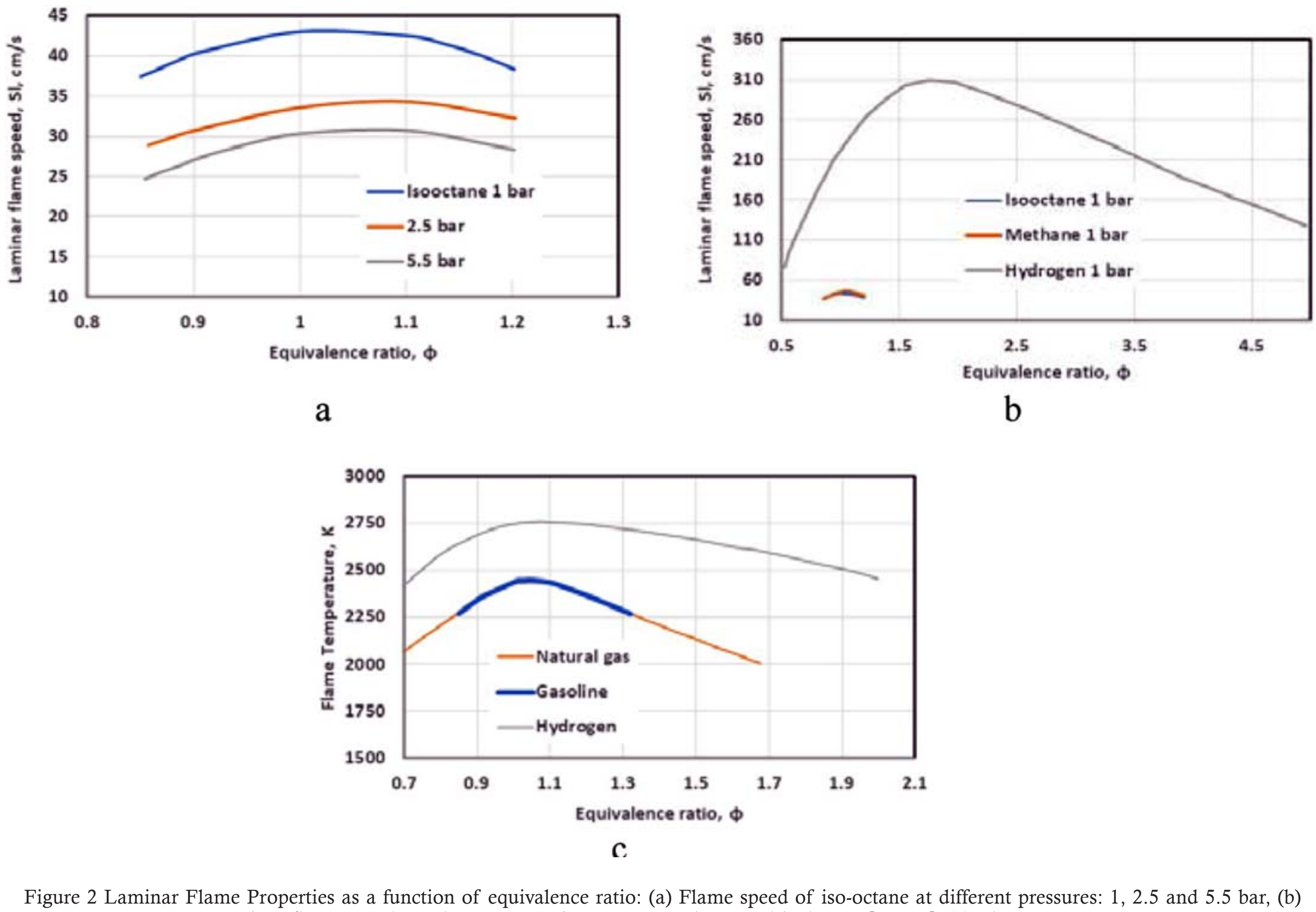


Figure 2 Laminar Flame Properties as a function of equivalence ratio: (a) Flame speed of iso-octane at different pressures: 1, 2.5 and 5.5 bar, (b) Laminar flame speed at 1 bar pressure: iso-octane, methane and hydrogen [10, 11], (c) Flame temperature

When the flow is laminar, the flame is very thin with a thickness, δ_L on the molecular scale, Figure 3a. The gaseous products of combustion are at a high temperature and hence have a lower density than the unburned gases (ρ_f).To maintain the continuity of flow, the hot products of combustion travel away from the flame at a higher speed, $||u||$ which is inversely proportional to their density, ρ_p. The vector of the mass flow, is perpendicular to the local flame surface, as indicated inthe figure. The velocity of the unburned fuel-air mixture in laminar flow i. e., the laminar flame velocity, S_L is the property of the fuel-air mixture at given pressure and temperature.The non-dimensional concentration of burned products, increases from 0 to 1 as the flow crosses the flame surface.

The laminar flame velocity is usually listed at some reference pressure and temperature. It could be evaluated at any other conditions by the following equation [7, 8, 9]

$$S_L = S_{L0}\left(\frac{\rho_0}{\rho_u}\right)^{\alpha} exp\,\frac{E_A}{2R}\left(\frac{1}{T_{b0}}-\frac{1}{T_b}\right) \tag{1}$$

where:

E_A = apparent activation energy

α = exponent determined by order of combustion reaction

S_{L0} = Laminar flame speed for refcondition given by $T_b/T_{b0} = \rho_u/\rho_0 = 1$

For isenthalpic combustion, the flame temperature or the burned gas temperature in K is given by

$$T_b = \left(h_{fu} - h_{fb} + c_{pu}T_u\right)/c_{pb} \tag{2}$$

where c_{pb} and c_{pu} are the effective specific heats for the burned and unburned gases at constant pressure.

The parameters in Eq. 1 may be determined from data on laminar flame speeds summarized by NASA [9]. Eq. (1) is graphically shown in Figure 2b for gasoline, compressed natural gas and hydrogen [10, 11] at 1 bar.

Since the heat of combustion is high for hydrogen the flame velocity is also far higher than for other fuels at any equivalence ratio, ϕ. The laminar burning velocity peaks when the fuel-air mixture is nearly stoichiometric and drops off when it is rich or lean. At higher pressures, the burning velocity is reduced, as already shown in Figure 2a. The flame temperature behaves similarly, Figure 2c and is higher for hydrogen than any other fuel.

The Nitric Oxides (NOx) form at high temperatures. Therefore, they are higher for hydrogen than for fuels like gasoline or natural gas, Figure 4a. This problem must be reckoned with when hydrogen is used in an engine, by running it lean and hence at low flame temperatures. The excess air ratio can be practically stretched to 2.2 (ϕ= 0.45) without breaching the lean flammability limit of (4% fuel-air ratio by volume corresponding to λ=10, Table 1). This aspect is discussed in detail in paragraphs 2.1.2 and 3. However, for the stability of the flame, higher turbulence is required than what is usually provided by the flow shear induced by the flow during the intake stroke.

Table 1 Minimum and maximum flammability limits of hydrogen, natural gas, gasoline and diesel fuels

Flammability limit	Fraction of fuel in the air (vol %)			
	Diesel	Gasoline	Natural Gas	Hydrogen
min	0.6%	1%	5.3%	4%
max	5.5%	7.6%	15%	75%

The laminar flow becomes unstable and trips to a turbulent regime at higher Reynolds numbers. The turbulent flame front is no more a simple sheet. It is highly wrinkled (Figure 3b) with an effective area which is increased manifold with a commensurate increase in observable thickness, δ_T. The flame velocity itself is enhanced corresponding to the increase in the true area of the wrinkled flame front.

Turbulent flame velocity, S_T = *ratio of turbulent and laminar flame areas* x *laminar flame speed* (3)

The time for combustion is limited to about 25 crank degrees after TDC in an internal combustion engine. The flow inside engines is invariably turbulent. The turbulence enhances the flame speed and enables complete combustion in a short time. It can be easily inferred that the higher the turbulent flame velocity, the higher the capability of the engine to run at higher speeds.

2.1.2 Flammability Limits

The laminar flame velocity drops abruptly when the mixture is either too lean or too rich, Figure 2b. Depending upon the time available for initiating combustion the rich and lean limits of combustion can be decided. Practical limits for different mixtures are given in Table 1. Hydrogen by its high flame velocity and heat of combustion has a far wider range of flammability than other fuels like natural gas or gasoline, enabling very lean combustion in an engine and as a result, producing very low nitric oxides.

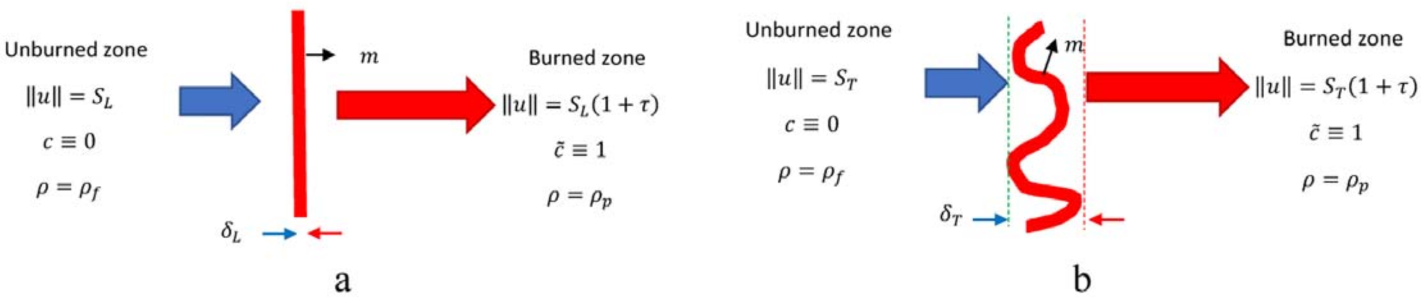


Figure 3 Flame fronts in (a) laminar flow and (b) turbulent flow

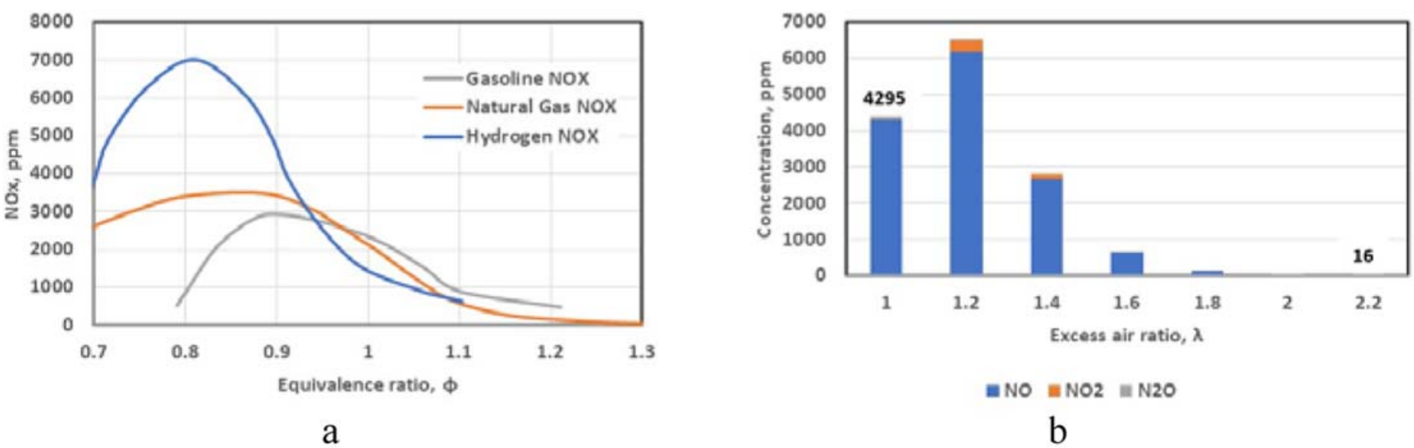


Figure 4 Nitric oxides as a function of equivalence ratio and excess air ratio (reciprocal of equivalence ratio): (a) comparison of gasoline, natural gas and hydrogen, (b) NO, NO_2 and N_2O as a function of air excess ratio for hydrogen

2.1.3 Ignition Energy

The energy needed to initiate a stable combustion in a fuel-air mixture is termed the ignition energy. In the context of a spark ignition engine, it is the minimum energy of the spark needed to trigger stable combustion. The ignition energy of hydrogen is substantially lower than that of natural gas or gasoline, Figure 5a. In other words, a conventional and cost-effective spark ignition system used in a gasoline engine is sufficient to start the flame in a hydrogen engine even with a lean mixture. The disadvantages of low ignition energy, however, are the difficulty in controlling the ignition timing and the possibility of firing due to hot spots in the hydrogen engine. In addition, how low ignition energy is responsible for backfire is discussed later in the paper.

2.1.3.1 Storing hydrogen

Also, while storing hydrogen, care must be taken by using only compatible materials, to avoid unintentional mixing with hydrogen and by considering design and safety factors to prevent leaks. The user must anticipate and control ignition mechanisms that can trigger the combustion of hydrogen-air mixtures. Finally, best practices must be practised to implement safety with inspection, assembly, operation, maintenance and safety systems.

2.1.4 Ignition Temperature

While the ignition energy of the hydrogen-air mixture is very low, the temperature at which combustion starts is higher than for either gasoline or natural gas, Figure 5b.Translating to the language of an engine designer, the knock resistance of hydrogen is far superior to gasoline [12]. The research octane number (RON) is 120 against 91-95 for gasoline. Thus, a compression ratio higher than 17and advanced spark timing are enabled to achieve higher thermodynamic efficiency.

2.1.5 Quench Distance

The heat loss at the walls of a combustion chamber is sufficiently high to bring down the temperature of the unburned fuel-air mixture, locally near the walls. The reciprocal of the temperature of the unburned gases appears in the exponent in the equation for laminar flame speed (Eq. 1). Therefore, at low temperatures, the flame velocity drops to too low a value to sustain steady combustion. The unburnt fuel at the walls is emitted in the exhaust and consequently can be serious if it is a hydrocarbon fuel. The distance from the wall at which the flame stops is defined as the quench distance. This distance is strongly correlated with the ignition energy, Figure 5c. Thus, the hydrogen flame with its low ignition energy quenches closer to the wall than a gasoline flame. In other words, the flame ends up so close to the wall that it can sneak through narrow gaps in the combustion chamber. The gap between the valves and valve seats and the "closed" gap of piston rings are some of the typical zones through which flame can propagate to the inlet manifold and crankcase inducing deflagration which is also known as backfire, Figure 5d. In this connection, the port fuel injection (PFI) of hydrogen can be a serious disadvantage. Also, sufficient care must be taken to ventilate the crankcase to avoid any build-up of hydrogen-air mixture within flammable limits that can explode. Since the flame can burn close to the walls, the oil on the cylinder surface must be correctly formulated to avoid any serious thermally induced degeneration.

Usually, the engine layout is checked during the design using Computational Fluid Dynamics (CFD) for any chance of backfire [13]. The backfire is sustained when the surface temperature of the hot spots near the quench gap is above 950 K.Therefore, the temperature of the spark plug must be maintained below 900 K to control backfire as well as autonomous ignition. The location of the hot spot does not influence backfire except for the timing of its origin.

2.2 Physics

2.2.1 Diffusivity

Hydrogen diffuses faster in the air than gasoline vapour, creating a uniform mixture which is conducive to complete combustion and improved efficiency. Higher diffusivity also helps a hydrogen engine's high-speed capability and hence develops higher power than a gasoline engine of comparable size. What is more, any leak of hydrogen from the vehicle is fast diffused in the atmosphere to less than the lean flammability limit avoiding danger.

2.2.2 Lower Density of Hydrogen

The density of hydrogen is extremely lower than that of gasoline. This calls for a large fuel tank which occupies a lot of space. The fuel must be compressed to higher pressures (700 bar) than even natural gas (240 bar) to store in somewhat economical volumes. Further, the stoichiometric mixture of air and hydrogen is 2.4 by volume and 34.3 by mass. See Table 2. The corresponding value by volume for gasoline is 83.3 and for natural gasis 9.7. In other words, hydrogen in its stoichiometric mixture occupies 34% of the cylinder volume compared to 1.2% by gasoline vapour. Put differently, if the fuel is mixed with air in stoichiometric proportion in the inlet manifold (e.g., port fuel injection, PFI) then, the air taken in is only 66% of the capacity of the cylinder in the case of hydrogen and 98.8% in the case of gasoline. If the fuel can be injected directly into the cylinder (direct injection, DI) the stoichiometric hydrogen engine can produce (100%/66%= 1.5) 50% more power. Therefore, a DI engine is preferred when hydrogen is used as fuel. A similar situation arises for natural gas as fuel but relatively to a lower extent.

3 Design Challenges

Table 3 summarises the properties and the design challenges when hydrogen is used in a spark ignition engine. The high octane number of hydrogen enables a higher compression ratio and consequently higher thermodynamic efficiency. Good diffusivity allows the engine to run at higher speeds and hence increase the power performance of the engine. In contrast, running at stoichiometric composition like most gasoline engines results in high NOx. Whereas in the case of gasoline,the NOx can be reduced in a three-way catalyst (TWC) with the help of hydrocarbons and carbon monoxide in the exhaust, the carbon-free hydrogen engine needs an expensive selective catalytic reduction (SCR) technique to reduce NOx like in a diesel engine. Therefore, the cost-effective solution to the NOx problem lies in running the engine lean at excess air ratio

$$(\lambda = \frac{1}{\phi} \text{ where } \phi = \text{equivalence ratio})$$

of 2.2. Figure 4b shows the reduction in NOx emissions from 4390 ppm to 14 ppm when the excess air ratio is increased from 1 to 2.2 due to a severe drop in the flame temperature. However, this scheme requests pumping in excess air resulting in parasitic losses and higher cylinder pressure. Yet, this appears to be a better choice than having an SCR aftertreatment of NOx from a stoichiometric engine. Low quenching distance of the engine results in a backfire deflagration which can be annoying and may result in some damage to the engine. The design must take this into account by keeping all the gaps at the valve seat and piston at a temperature sufficiently lower than 900 K. This can be achieved by refining the design layout by using CFD simulation as mentioned already.

Lower density is probably the biggest problem while storing and handling hydrogen in a vehicle. Compressing to unusually high pressures is the only practical solution. However, the penalty in terms of cylinder weight is very high. In some automotive applications, when the cylinder is full, the weight of hydrogen is only 5.7% of the total weight.The cylinders occupy most of the boot space in a car; it is possible to accommodate them in a truck behind the driver's cabin.

Of course, using hydrogen in a liquid state would reduce the volume occupied by the fuel substantially; however, the associated disadvantages like boiling off of liquid during the dormancy of the vehicle and the complex design of the storage cylinders make the technology not attractive in automotive applications.

The method of cryo-compressed hydrogen appears to have the good attributes of cryogenic liquid hydrogen technology and compressed gas technology at the same time with their disadvantages diminished to a large extent. A note on these storage technologies is given in the Appendix.

Table 2 Density, heating values and stoichiometric air-fuel ratios of hydrogen, natural as and gasoline

Type	Fuel	Density	Higher Heating Value		Lower Heating Value		Air/fuel	
		[kg/m^3]	[MJ/kg]	[MJ/m^3]	[MJ/kg]	[MJ/m^3]	mass	vol
Gaseous	Hydrogen	0.090	141.7	12.7	120.0	10.8	34.3	2.4
	Natural gas	0.777	52.2	40.6	47.1	36.6	17.2	9.7
Liquid	Gasoline (petrol)	737	46.4	34200	43.4	32000	14.7	83.3

Table 3 Summary of properties and challenges

Property		Design		Comments
Octane number	130	Compression ratio	> 17, Higher Efficiency	desired
Flame temperature, at λ=1	2800°C		High NOx	Not desired
Flame temperature, at λ=2.2	Low	NOx	Low	desired
Density	Low	Fuel tank size	Large	Not desired
		Engine power	Low	
Flame speed	Very high	Engine speed	High	desired
Quenching Distance	Very low	• All the engine's components must meet particularly stringent leak-tightness requirements. • This is because the molecules of hydrogen are very small. • That's why they can diffuse more easily.	• Purely conventional and solvable method • A specific solution has to be found for this	Not desired
Diffusivity	Very high			desired

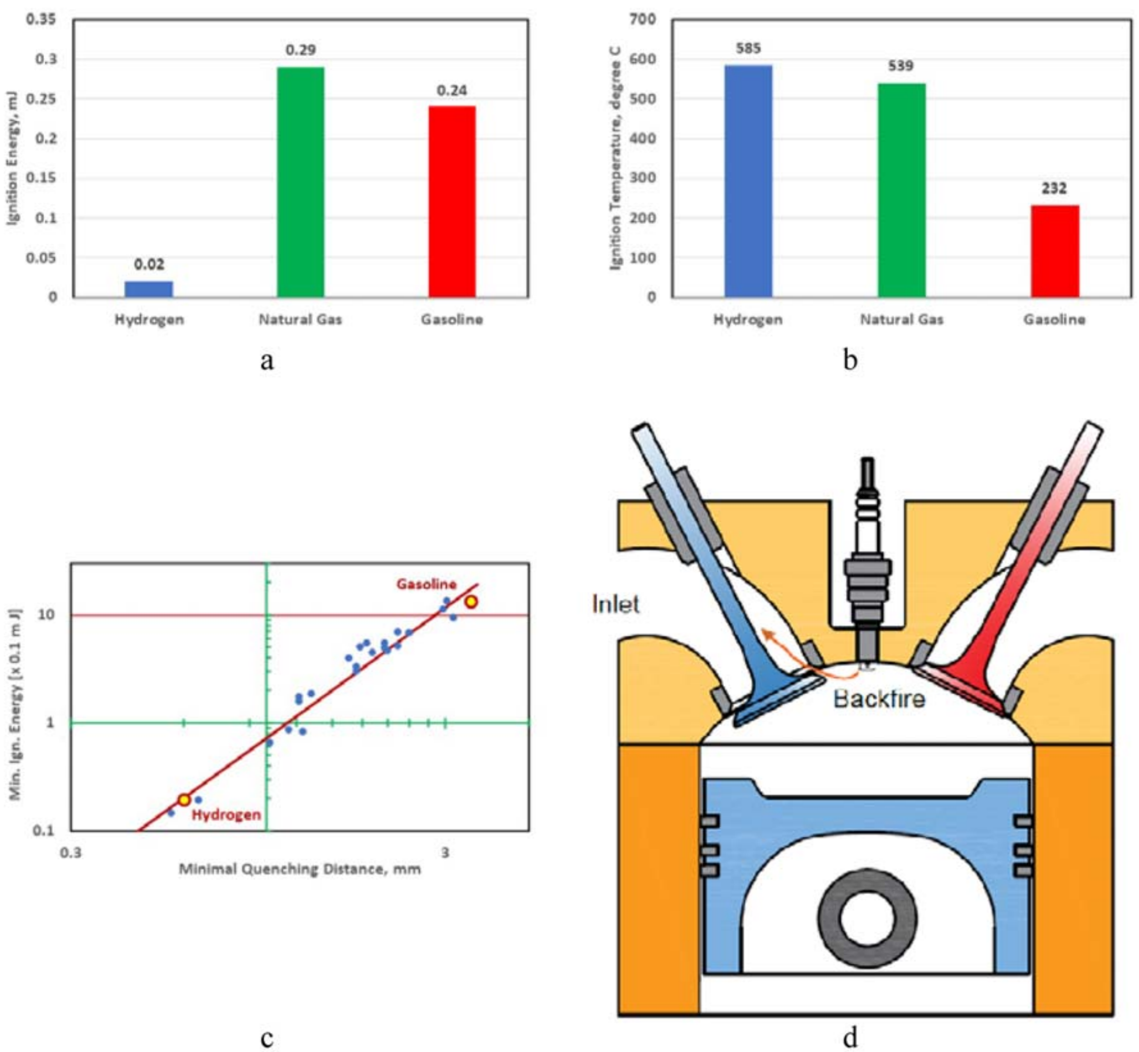


Figure 5 Consequences of Ignition Energy: (a) Ignition energy of hydrogen, natural gas and gasoline, (b) Ignition temperatures of hydrogen, natural gas and gasoline, (c) A strong correlation between quenching distance and ignition energy, (d) Because the quench distance of hydrogen flame is puny, the flame goes past the narrow gaps of valve seats triggering a backfire

3.1 Ignition System

While the ignition system can be borrowed from conventional gasoline engines because of the low ignition energy of hydrogen, three important points must be taken care of. (1) The spark plug should have a cold rating. A cold-rated plug transfers heat from the plug tip to the coolant in the cylinder head quicker than a hot-rated spark plug. Thus, it maintains the temperature below 900 K and avoids any chances of the spark plug tip igniting the charge. (2) The electrode of the spark plug should not be of platinum since it is a good catalyst and initiates uncontrolled ignition even at low temperatures. Incidentally, platinum and hot rating are prevalent in gasoline engines to burn away any carbon deposit which does not arise when carbon-free hydrogen fuel is used. (3) For gasoline engines, waste spark systems work well and are less expensive than other systems. In these systems, the spark is energised each time the piston is at the top dead centre whether or not the piston is in the compression or the exhaust stroke. For hydrogen engines, the waste sparks are a source of pre-ignition.

3.2 Crankcase ventilation

The unburnt fuel seeps by the piston ring gaps and enters the crankcase. Since hydrogen has a lower ignition energy than gasoline, any unburnt fuel entering the crankcase has a greater chance of getting ignited. On ignition within the crankcase, a startling noise is produced due to a sudden rise in pressure or an engine fire results. Therefore, it should be prevented from accumulating by proper ventilation. In case there is some ignition, the pressure rise must be relieved by a pressure relief valve installed on the valve cover. During combustion and exhaust strokes, exhaust gases containing water vapour seep into the crankcase. Water condenses on the crankcase surfaces when proper ventilation is not provided. The mixing of water into the crankcase oil reduces its lubrication ability, resulting in a higher degree of engine wear.

3.3 Materials to withstand Hydrogen Embrittlement

The combustion chamber and other parts exposed to hydrogen must be made of materials that withstand hydrogen embrittlement. These are discussed in Ref. [14]. For example, the recommended material for valves and prechamber is stainless steel 316/L. The valve seat resting faces must be hard.

4 Lean Burn Concept

The discussions so far irrefutably point towards the lean burn concept of a hydrogen engine with direct injection. In reference [15] the heat release rate in a stoichiometric combustion engine (λ=1) and a lean burn hydrogen engine (λ=2.2) are compared for an engine of a certain size, Figure 6a. The heat release in the former case is explosive and concentrated near the TDC whereas in the latter the heat release is steady and gradual over 65 crank degrees. Also, the pumping losses reduce by 5% from 9.6% when the mixture is leaned with the corresponding improvement in brake thermal efficiency by 7.5% from 26%.

4.1 Fuel Injection

Port fuel injection (PFI) suffers from the possibility of backfiring since the fuel-air mixture fills the nook and corner of the cylinder. Also, the effective volumetric efficiency is poor since 34% of the air is replaced by hydrogen in the cylinder for stoichiometric combustion and 16% for a λ=2.2. This loss results in substantially reduced torque compared to the performance with direct injection (DI) of fuel in the cylinder. Ref. [16] demonstrates this aspect clearly as shown in Figure 6b for an engine of a certain size and cylinders. Therefore, to maximise the power produced for a given engine size, direct injection seems to be the right choice.

4.1.1 Two Possibilities of Direct Injection

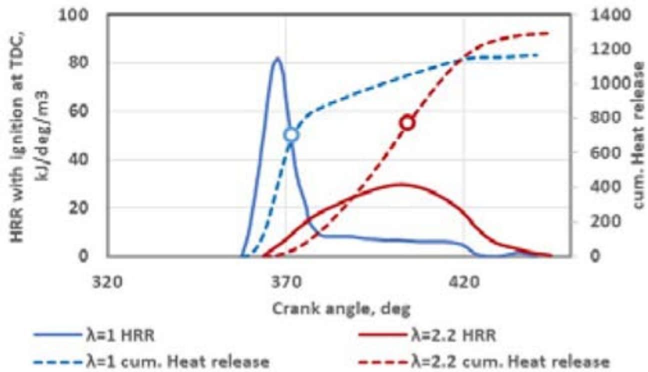
Direct Injection (DI)above the piston, along the lines of a conventional GDI (Gasoline Direct Injection) engine, is an option. Designing the cylinder head to accept the injector and the spark plug is simple to execute. Thanks to the turbulence created by the airflow past the valves and left over after the decay during the compression stroke, the flame speed is enhanced. In the case of less engines using less diffusive gasoline, complex piston tops enable the flow of fuel spray to be guided either by the piston crown or the airflow. However, the desire to burn very lean mixtures of hydrogen may not be successfully met due to the lack of stratification of fuel since hydrogen diffuses fast and insufficient turbulence level for the stable combustion of an ultra-lean mixture of hydrogen and air. In addition, the fast-diffusing hydrogen in the cylinder can fill in all the gaps through which the flame can propagate because of low ignition energy/quenching distance resulting in a backfire or crankcase explosion. Also, the flame may be close to the walls causing the deterioration of oil on the cylinder walls. Just like in the case of PFI, the whole combustion chamber is in contact with hydrogen with simple direct injection and therefore, all the walls have to be proofed against hydrogen embrittlement, by choosing expensive materials. In summary, simple direct injection of hydrogen above the piston is not very attractive.

4.1.1.1 Prechamber Direct Injection

Passive prechamber

Alternatively, there is a concept of prechamber direct injection used in some gasoline engines, Figure 7a. Here the spark plug is fitted in the upper part of the prechamber which is integrated into the cylinder head and the existing cooling system. The lower part is a cap with several holes directed towards the wall of the bowl in the piston [17]. This system as shown in Figure 7a, where the injector is mounted in the main chamber above the piston is known as the passive prechamber arrangement [18], and it is less costly than the active prechamber system to be discussed next section.

In the case of a passive system, the mixture strength in the prechamber is the *same* as in the main chamber.The fuel-air mixture trapped in the prechamber is ignited by the spark and the partially burned gas exits ferociously through the holes into the main chamber. This leads to multiple places of ignition in the main combustion chamber. Further, the higher turbulence by the jets enhances flame speed and helps in the stable combustion of lean mixtures up to λ = 1.4, Figure 8 and not more. The fuel economy is up to 4% better than the plain DI case. The spark plug has multiple electrodes to assure ignition at lean limits. It may be noted that as in the case of plain DI or PFI, in passive prechamber cases, the entire zone above the piston must be strengthened against hydrogen embrittlement since hydrogen can make contact with the walls.



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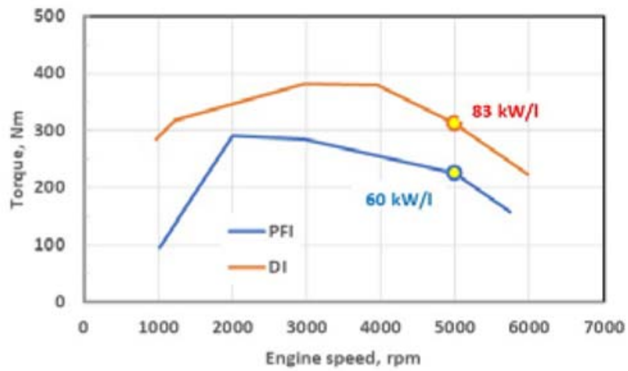

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Figure 6 Comparison of port fuel injection and direct injection engines: (a) Heat release rates and cumulative heat release of stoichiometric and lean combustion engines [15], (b) A Comparison of the torque-speed performance curves of hydrogen engines using port fuel injection (PFI) and direct injection (DI) systems [16]

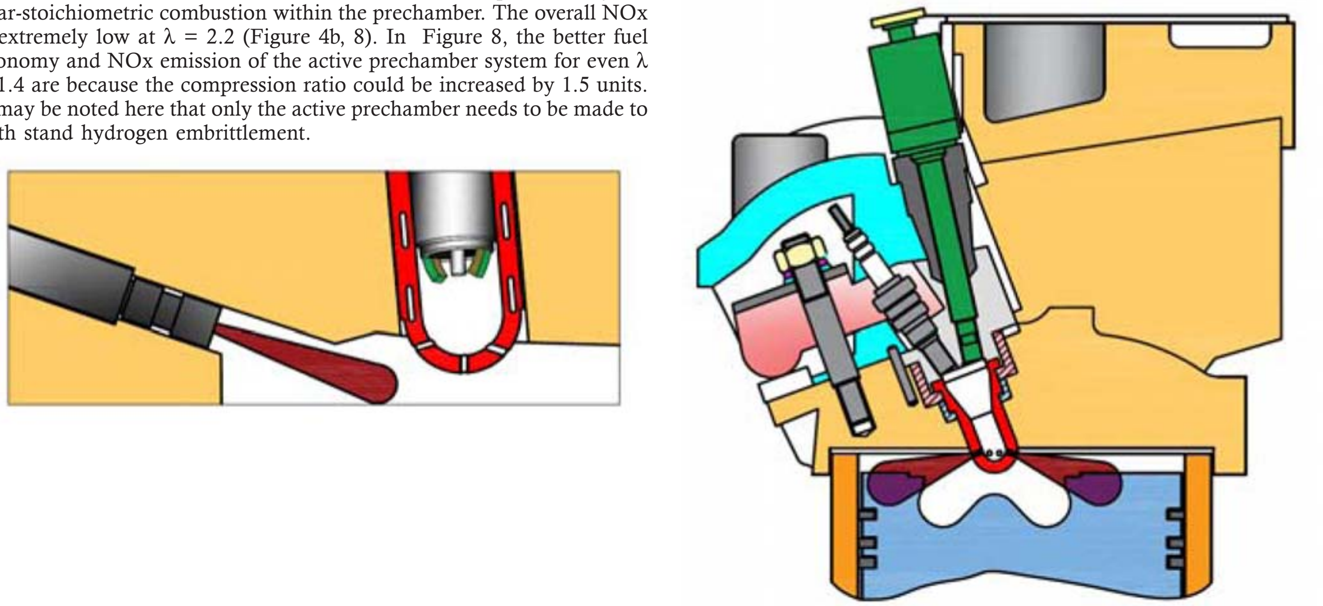
Active Prechamber

The prechamber system is termed active if the injector is also mounted within the prechamber, Figure 7b [19]. It provides stable combustion of a very lean mixture of $\lambda = 2.2$ (Table 4) because the mixture in the prechamber is nearly stoichiometric during the early part of the combustion in the prechamber aiding stable combustion and the jets above the piston provide multiple sources of ignition and enhanced turbulence. The system offers fuel economy improvement by up to 10% over DI since the efficiency improves with leaner combustion.

By using CFD, it is possible to exploit the residual exhaust gases in the prechamber to work as internal EGR and control NOx produced by the near-stoichiometric combustion within the prechamber. The overall NOx is extremely low at $\lambda = 2.2$ (Figure 4b, 8). In Figure 8, the better fuel economy and NOx emission of the active prechamber system for even $\lambda < 1.4$ are because the compression ratio could be increased by 1.5 units. It may be noted here that only the active prechamber needs to be made to with stand hydrogen embrittlement.

4.2 *Design of Bowl in Piston of an Engine with Prechamber*

A gasoline DI engine has complicated shapes for the piston crown arrived at using CFD and transparent engines running in laboratories. The shapes are designed for guidingthe fuel sprays by air or crown shape as per the designer's choice. The guides allow the stratification of fuel within the combustion chamber. The distribution of fuel is such that the walls are avoided completely by the fuel to minimise hydrocarbon emission and there is a flammable mixturenear the spark plug at the time of ignition. In the case of hydrogen, such stratification is difficult because of the high diffusivity of the fuel. Therefore, as already discussed, this makes



a b

Figure 7 Prechamber arrangements (a) Spark plug in the passive prechamber [17], (b) Active prechamber with the injector and the spark plug; the main chamber with a diesel-like bowl in the piston [19]

Table 4 Guide to the selection of Fuel Injection System

Detail	Property		Design	
Port Fuel Injection	NOx high at $\lambda = 1$		• If the three-way catalyst is used, run rich at $\lambda = 0.95$	• Poor efficiency • Satisfactory power
	NOx low at $\lambda = 1.4$	The volume occupied by hydrogen high	• Naturally aspirated • Catalyst for NOx can be avoided	• Low power • Higher efficiency • Low NOx
			• Turbocharge and run at $\lambda = 2.2$ • Catalyst for NOx can be avoided	• Satisfactory power • Higher efficiency • Low NOx
PassivePrechamber	Overall $\lambda = 1.4$	Fuel injected in the main chamber above the piston	• Naturally aspirated • Can be turbocharged • Catalyst for NOx can be avoided	• Low power • Higher efficiency • Low NOx
Active Prechamber	Overall $\lambda = 2.2$	Fuel injected in prechamber	• Naturally aspirated • Can be turbocharged • Catalyst for NOx can be avoided	• Satisfactory power • Higher efficiency • Low NOx • Suitable for heavy-duty application

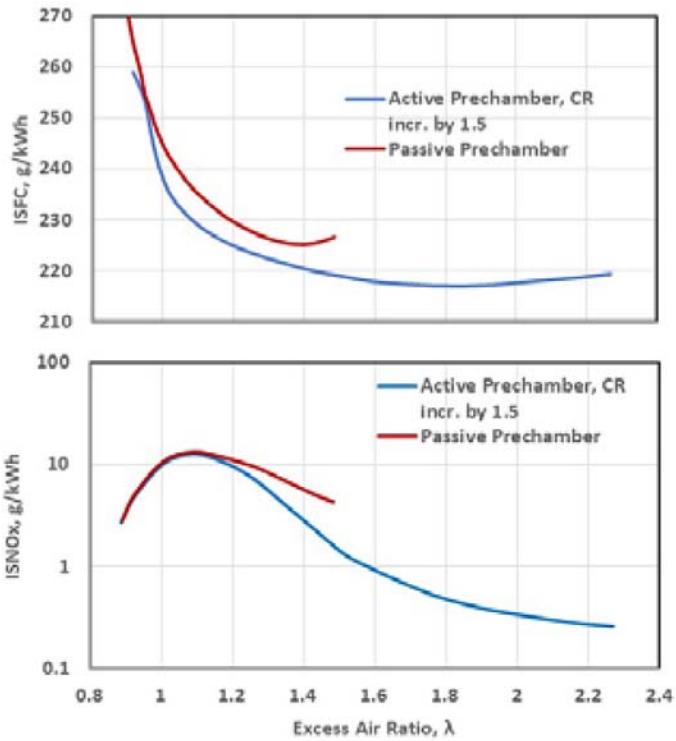


Figure 8 Indicated specific fuel consumption (ISFC) and NOx as functions of excess air ratio, λ: Red - Passive prechamber, Blue - Active prechamber with compression ratio increased by 1.5 [20]

prechamber combustion more suitable than a plain direct injection system popularly used in gasoline engines. The hot jets emanating through the holes of the prechamber provide multiple sources of ignition and create high turbulence to burn mixtures as lean as λ=2.2 with an active system. These jets appear similar to the hot fuel sprays in a DI diesel engine. Hence, the combustion bowl in the piston of a hydrogen actuve prechamber engine is designed based on similar principles used to design the DI diesel combustion chamber [20].

4.3 Air mass flow requirements and Turbocharging a Lean Burn Engine

For lean combustion hydrogenengines (λ=2.2), the turbochargers have to provide more than two times the mass asin equivalent naturally aspirated stoichiometric PFI gasoline engines. Hence, if the performance of a naturally aspirated PFI or DI gasoline of similar size is to be achieved, a conventional single-stage turbocharger developing a boost of 1.8-2 would be sufficient, when the hydrogen injection is direct type. However, for the PFI Lean burn concept of a hydrogen engine, the air has to be admitted at a boost still higher by 20% or more.

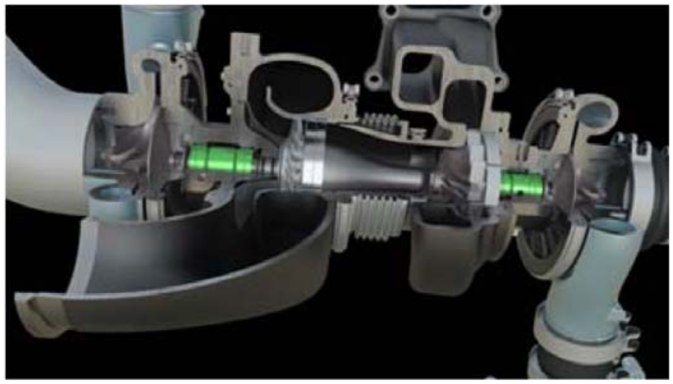


Figure 9 A two-stage turbocharger mounted on a single shaft [21]: from left (1) high-pressure compressor, (2) low-pressure turbine, (3) high-pressure turbine and (4) low-pressure compressor

For achieving the performance of a contemporary turbocharged and intercooled gasoline engine (e.g., bmep 20 bar/1500 rpmor 15 bar/5000 rpm) a two-stage turbocharger delivering a higher boost of 3-4 would be required even when hydrogen is admitted by direct injection in an engine of similar size. A two-stage turbocharger running at relatively lower speeds is suitable for this.

One such turbocharger [21] with a special feature of coaxial arrangement of two compressor stages, which are driven by the turbines or the supporting electric motor using a common shaft is shown in Figure 9. In the first stage compressor, the pressure is raised to 2 to 2.5 times the atmospheric pressure. Then the air is cooled in an intercooler. The pressure is raised another 2 to 2.5 times in the second-stage compressor. Then the air is cooled again. The resulting charge air at the intake manifold of the engine is typically 4 to 5 times that of atmospheric pressure and at about 20°C higher than ambient temperature. Both compressors operate at peak efficiency by splitting the compression between two turbo-compressors. Also, the lower pressure ratios for each stage mean lower rotating speeds, which results in improved reliability for the bearing systems, compressor wheels and turbine wheels

5 Summary

The various possibilities to develop a highly efficient and cost-effective hydrogen engine for heavy-duty application on- and off-road are discussed. NOx emissions per kWh from stoichiometric hydrogen combustion are 2.5 times higher than gasoline combustion. Low NOx lean combustion is preferred to stoichiometric because an SCR increases the cost. A single-stage turbocharger for the lean burn engine may be satisfactory to achieve the performance of an equivalent naturally aspirated gasoline or diesel engine, but a two-stage intercooled turbocharger is recommended for matching an equivalent turbocharged gasoline engine. Direct injection is the obvious choice to improve volumetric efficiency, but there are two options: with or without a prechamber. The case without a chamber is easy to execute but is less efficient and may result in backfire with λ<1.4. Placing the injector inside the prechamber enables the mixture near the spark plug richer than the average mixture strength and permits stable combustion even when the average strength is as lean as λ=2.2. The route to the selection of technologies for the most optimum hydrogen engine is shown in green in Figure 10.

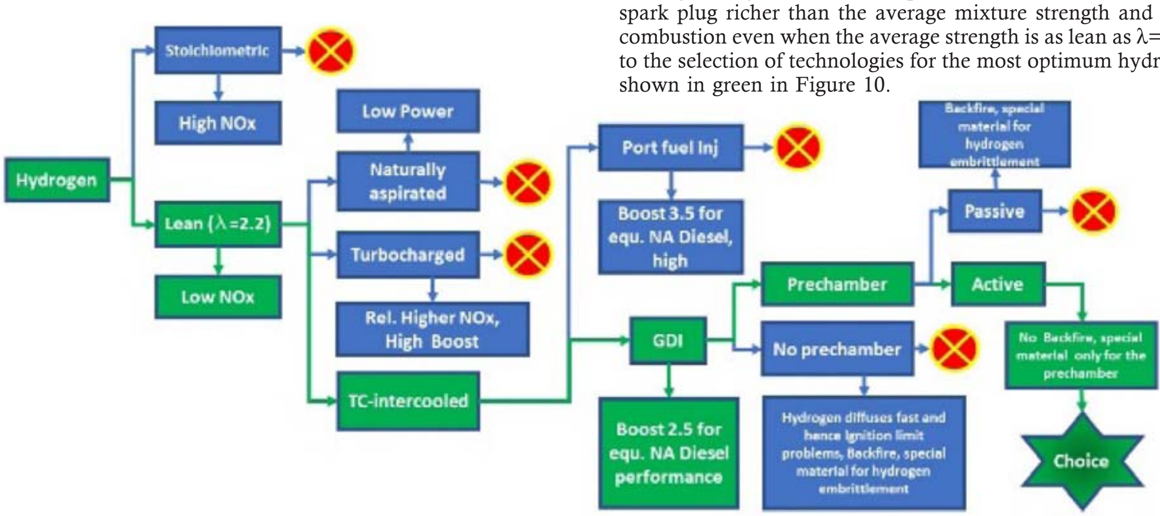


Figure 10 Technology options and the optimum choice for a hydrogen engine

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Appendix: Storing Hydrogen for Automotive Application

Conventional cryogenic hydrogen storage vessels (LH_2) with increased volumetric density in a specially insulated tank to avoid heat intrusion, are limited to the storage of low-pressure liquid hydrogen (LH_2) at about 20 K and 1 atm [A1]. Interestingly, liquid hydrogen occupies three times more volume than gasoline of equivalent energy. The LH_2 technology is not favoured for the following reasons in automotive applications:

1) about 35% of the fuel energy is used to liquefy which is three times the energy needed to pressurise the gas to compressed gas hydrogen (CGH_2);
2) the fuel boils off, even in highly insulated tanks;
3) the pressure quickly increases as heat is absorbed from the environment which must be vented every 3–5 days;
4) during the running of the vehicle, the tank is pressurised by heat absorption.

Due to these issues, studies of storage have shifted to cryo-compressed H_2 (CcH_2) which combines the favourable attributes of compressed and cryogenic hydrogen storage [A2]. At the same time, it serves to limit the challenges of

a) CGH_2 requiring large volumes and high pressures, and
b) the inevitable boil-off losses in the case of LH_2 storage.

The technology uses an insulated vessel that can operate at both cryogenic temperatures as low as 20 K and pressures up to 350 bar. Advantages are

1) 2–3 times higher storage-system energy density than CGH_2 at room temperature thus reducing the intrusion of the cylinders into the cargo space as well as extending the vehicle range by about 30%;
2) elimination of evaporative losses in repetitive use;
3) improved thermal endurance by 5–10 times more than for the case of LH_2
4) no over-pressurization during hydrogen fill;
5) reduction in the capital cost by up to 25% of CGH_2 vessels.

The optimum operating regime for CcH_2 vessels is 200 K, 480 bar for safe and reliable performance, though less than 300 bar is typical. Some automotive companies have committed to the development of this technology because the gravimetric and volumetric hydrogen capacities improved by 91% and 175% respectively, and the loss during vehicle dormancy for up to seven days is neligible, under specific conditions of minimum daily driving. Here, the insulated vessel used to store hydrogen is capable of withstanding cryogenic temperatures and moderately high pressures. The tank has an exterior stainless steel jacket and an inner aluminium cylinder wrapped with carbon fibre composite. The inner cylinder is surrounded by a vacuum space with highly reflective surfaces to limit heat transfer from the environment [A3].

This helps to improve the volumetric hydrogen storage capacity and safety over that of CGH_2 or cryogenic LH_2. In this regard, by compressing liquefied hydrogen at 20 K and 240 bar, the volumetric hydrogen storage capacity is increased by 24% from 70 g/litre at 1 bar. The capability of the insulated vessel to hold high pressures permits a higher rise in pressure within the tank than in the case of cryogenic storage, and extends the dormancy period, resulting in increased storage density and decreased boil-off losses. Furthermore, the lower pressures employed in CcH_2 storage relative to CGH_2 (700 bar) reduce the requirement for costlier carbon fibre composites. However, the outstanding issues are problems with the LH_2 pump performance, vacuum stability, and manufacturability.

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