Performance of Hydrogen Direct Injection Engine

Abdul Rashid Abdul Aziz, Muhammad Adlan Abdullah, Firmansyah and Ezrann Zharif Zainal Abidin

Hydrogen's high flammability, low ignition energy, clean burning, and high flame speed are attractive advantages of using internal combustion engines to attain high performance and efficiency as well as achieving "real" clean emission with only water and carbon dioxide as exhaust. To overcome volumetric efficiency loss, high NO_x emissions and abnormal combustion, the influence of air–fuel ratio is studied. Optimization of the air–fuel ratio effect and understanding of the combustion processes are actively being studied in the automotive industry and research institutions.

Introduction

Hydrogen economy is the most accepted and talked about hypothetical energy economy in the world today. Efforts to realize the hydrogen economy have led to research initiatives across the globe [1]. The hydrogen production from renewable resources supporting this initiative by introducing various production methods such as biomass gasification, biological production, biomass pyrolysis, and supercritical water hydrogen production [2]. Technology development of these production methods has reached commercialization stage with production efficiency of up to 70%, and thus improving the availability of hydrogen in the market and lowering the production costs while utilizing waste materials.

In the use of hydrogen for transportation fuel, there is a widely accepted argument that direct combustion of hydrogen in internal combustion engine could serve as a pathway to hydrogen economy before fuel cells technologies mature and become

A. R. Abdul Aziz (✉) · M. Adlan Abdullah · Firmansyah · E. Z. Z. Abidin
Universiti Teknologi PETRONAS, Perak, Malaysia
e-mail: rashid@utp.edu.my

M. Adlan Abdullah
PETRONAS Research Sdn. Bhd., Bangi, Malaysia

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cost effective [3]. Current automotive technologies and manufacturing processes can be used, and the experience of operating and handling gaseous fuels such as natural gas can prove beneficial. This would also facilitate proliferation of the hydrogen production and refueling infrastructures. As hydrogen has unique and sometimes contradicting characteristics in relation to its use as internal combustion fuel [4], it poses new challenges and opportunities. On the one hand, hydrogen's wide flammability range gives an opportunity for throttle-less operation (or power control by fuel quantity controlled means similar to diesel engines) with significant volumetric efficiency improvements potential. The high flame propagation speed gives rise to near constant volume combustion, yielding a potential thermal efficiency improvement. A further thermal efficiency improvement is also possible due to the inherent hydrogen's high auto-ignition temperature (a higher compression ratio engine is possible).

On the other hand, hydrogen's very low density and viscosity, low ignition energy, and low radiant heat push for new sets of thinking and approach to using it as a fuel for internal combustion engines. For example, while high compression ratio is desired considering the high octane of hydrogen, its low ignition energy may result in preignition due to glow ignition. In this case, optimized compression ratio or combustion chamber design may need to be determined. Even though the combustible limits of hydrogen are wide, its low density and viscosity may require optimization of injection parameters in order to reduce NO_x and the occurrence of preignition. Most of the previous works adopted lean burn strategies for hydrogen operation to avoid abnormal combustion. Even for direct injection engines, only compression ratios similar to gasoline engine were used.

The immediate development in direct hydrogen implementation as fuel is its utilization in a direct injection, spark ignition engine. In view of hydrogen favorable properties, it is used to overcome performance deficit of DI CNG engines, especially at low-end speeds (<2500 rpm), whereby other gaseous fuels such as CNG have low combustion efficiency resulting in lower torque and power [4, 5].

In light of the hydrogen economy, hydrogen-fueled internal combustion engines are said to be the pathway or bridging technology before fuel cells technology becomes mature and widely available. This helps proliferate the hydrogen refueling infrastructure while gaining the much-needed experience and reputation. The experience of NGV industry can be proven valuable as the operation of the compressed gaseous fuel is not much different from hydrogen, especially in the aspects of refueling and storage [6]. With the increasing concern over global warming and the push for a hydrogen economy, the interest in hydrogen internal combustion engine (H2ICE) has recently been renewed. Some research groups in the automotive industry are looking into H2ICE, and some prototypes were launched.

Hydrogen internal combustion engine started as early as in the 1800 s when Francois Isaac de Rivaz of Switzerland used a mixture of hydrogen and oxygen as fuel [4]. The interest in hydrogen engine subsided as gasoline and diesel fuels, which have better versatility in terms of storage and energy density became widely available. With the advent of environmental and energy security issues coupled with the availability of the state-of-the-art engine controls, hydrogen internal combustion engines have recently regained much interest. This is compounded by the fact that

H2ICE is regarded as the transition technology before fuel cells technology is mature and ready for wide adoption.

The main driving force for H2ICE is the ability to operate the engine at significantly leaner air–fuel ratio so that higher efficiency in comparison to gasoline [7–9] can be achieved. This is due to the low lean limit and ignition energy of hydrogen that allows the engine to be run with stable combustion with little or no throttling. As such, most of the research works involving hydrogen internal combustion engines were aimed at increasing the lean limit of the spark ignition engines [10–12].

Hydrogen has stoichiometric heating value that is 17% higher than that of natural gas. Thus, it follows that for hydrogen, higher power output is expected for the same engine capacity. However, the specific output is generally low since hydrogen at stoichiometric occupies 30% volume in comparison to only 10% volume for natural gas. For comparison, gasoline occupies only 1.76% volume [14]. This is true in the case of carburetor or port fuel injection. For direct injection, the intake charge can be maximized, and hence, the full load performance can be increased [15].

Hydrogen's very wide flammability limit allows the engine to operate at ultra-lean combustion leading to high efficiency and low NO_x emissions. The engine can be run much, if not all the time, unthrottled. Its high octane number can be exploited to increase efficiency. Hydrogen combustion has a high flame velocity that leads to almost constant volume combustion, and thus resulting in a higher thermodynamic efficiency.

On the other hand, hydrogen's ignition energy is about one order of magnitude lower. Thus, it is susceptible to preignition from hot spots such as spark plug electrodes, combustion chamber deposits, oil contaminants, combustion in crevice volumes and residuals energy in ignition systems [16, 17]. This abnormal combustion could also lead to backfire as the combustion "flashback" to the intake system. As shown in Fig. 1 on the ignition energy versus equivalence ratio, operating the engine at close to stoichiometric air–fuel ratio increases the risk of the abnormal combustion [17–19]. Optimized fuel injection timing [20], water injection [21, 22], improved scavenging by variable valve timings, use of liquid hydrogen, ultra-lean combustion strategy, and direct injection [9, 20, 23] have been shown to be effective in reducing the preignition. Direct injection strategy also eliminates the possibility of backfire.

In addition, hydrogen combustion has higher flame temperature and lower quenching distance that leads to narrow thermal boundary layers. This results in more heat loss to the combustion chamber walls. Charge stratification strategies by employing direct injection technique was shown to be able to reduce the heat loss.

As hydrogen has no carbon content, its combustion results in zero fuel-derived carbon emissions. As expected, extremely low THC, CO, and CO_2 emissions are produced. The source of the minimal carbon emissions is from the engine lubricants. NO_x emissions can be an issue in hydrogen engine since it has high peak cylinder temperatures. It has been demonstrated that this can be effectively controlled by several means. Ultra-lean operation, EGR, water injection, use of liquid hydrogen, use of catalytic converter, and direct injection strategies are the various methods used to minimize the emission of NO_x.

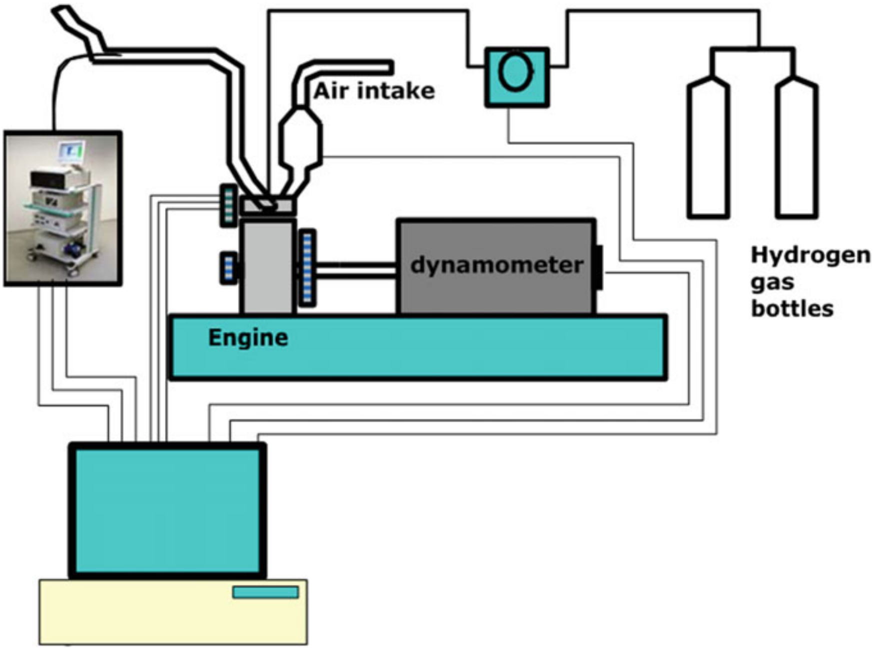


Fig. 1 Schematic of the experimental setup

In short, running an engine on hydrogen requires different strategies and design features than traditional fuels. Direct injection is generally preferred while the load control via mixture quality control means is desired for improved efficiency and fuel economy.

In an internal combustion engine, the energy introduced into the cylinder is closely related to the stoichiometric air–fuel ratio of the fuel in the air. The theoretical air–fuel ratio for hydrogen is 34 with volume percentage in the mixture of 29.6%.

For internal combustion engine, the energy introduced into the cylinder is the parameter that determines the mean effective pressure and power produced by the engine. For port/external mixture preparation, this depends on the mixture volume. In this case, since hydrogen occupies almost 30% of the mixture volume, the energy content is significantly diminished. In contrast, for direct injection, the energy introduced into the cylinder depends on the mass of air inducted into the engine.

The mixture calorific values of various fuels for external mixture preparation and for direct injection were shown by White et al. [13]. They reported that hydrogen has the largest difference in the mixture calorific value for direct injection and external mixture preparation as compared to other fuels such as methane. Thus, for hydrogen, direct injection is the solution for best power density. In the case of direct injection, hydrogen is also shown to have the highest mixture calorific value in comparison to other fuels. For example, hydrogen has 20% more mixture calorific value than natural gas (methane). Thus, it follows that for the same direct injection engine capacity, operating with hydrogen would result in 20% more torque and power than with natural gas.

The ignition energy required to start the combustion of hydrogen at various equivalence ratio is lower than that of natural gas. Therefore, it is expected that operating an engine with hydrogen would result in easier ignition in comparison with natural gas. This is also true at engine operating regions where air motion and mixing is poor.

The flame speed of hydrogen is approximately six times faster than natural gas, and it varies with different air–fuel ratio as found by Ilbas et al. [24]. Thus, it is expected that the combustion of hydrogen in an engine will be faster even when mixing is poor as in the case of low engine speed. These two arguments lead to the assumption that operating an engine with hydrogen would result in a more efficient combustion at lower speeds where natural gas is shown to have poor efficiency and lower performance.

Test Procedure and Equipment

Figure 1 shows a schematic of the experimental setup in this work. A single cylinder engine was coupled to a direct current dynamometer that allowed engine braking and motoring while the performance parameters were measured.

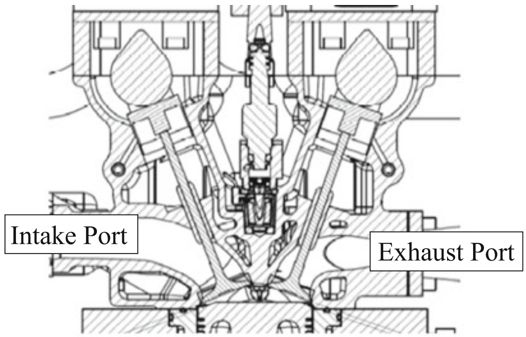
The Test Engine

The single cylinder engine system was set up based on a PROTON CamPro engine with modifications to its cylinder head to enable direct injection of gaseous fuel (Fig. 1). Shown in Fig. 2 is the cutoff view of the engine. Originally, the engine was designed for natural gas application. To demonstrate the practicality and easy adoption of hydrogen, no modification to the engine was made for this study. The specification of the engine is as given in Table 1. It was a four-stroke spark ignition engine with a compression ratio of 14:1 to take advantage of the high octane of natural gas. Since hydrogen auto-ignition temperature is higher than natural gas, this compression ratio was maintained.

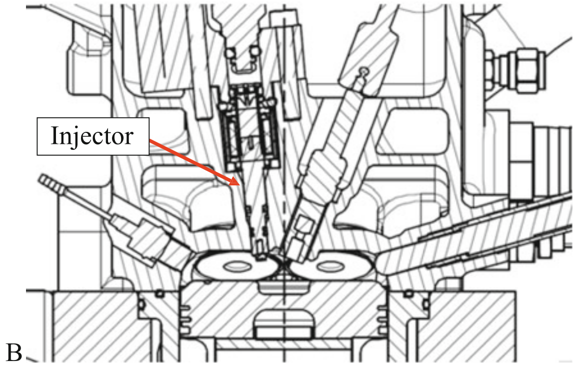
A programmable ECU connected to a computer was used to control the engine. The engine parameters that could be controlled from the computer were the injection timing and duration, the spark timing, and throttle position. Real-time data were available from the engine ECU and could be viewed and recorded accordingly.

The original natural gas direct injector was used for the study without any modification, as shown in Table 2. Due to hydrogen's low density and viscosity, narrow-angle injector (30°) was chosen to allow maximum fuel spray penetration. The spray was executed with 18 bar injection pressure in air at atmospheric conditions. It shows that although the penetration of hydrogen was quite similar to natural gas, the distribution of hydrogen was wider. This suggests that the mixing of hydrogen was better than natural gas.

Fig. 2 Cutoff view of the engine



(a) Intake and exhaust port position



(b) Injector and spark plug position

The Dynamometer

Table 3 shows the specification of the direct current dynamometer. Being a DC dynamometer, it has the capability to motor the engine. This was useful to obtain the unfired cycle data for the cylinder pressure such that the combustion analysis can be carried out. The data such as the torque, speed, engine oil temperature, coolant temperature, and intake air temperature were recorded manually from the dynamometer panel.

The Fuel System

The fuel system used for the study is as shown in Fig. 3. The hydrogen was supplied in gas bottles at a pressure of 200 bar. Pressure regulators were used to regulate the pressure delivered to the engine. A Micro motion™ CMF010 ELITE Series fuel flow meter was used to measure the fuel flow rate.

Table 1 Specifications of the single cylinder engine

Engine specifications	
Displacement volume	399.25 cm^3
Cylinder bore	76 mm
Cylinder stroke	88 mm
Compression ratio	14
Exhaust valve open	ATDC 10°
Exhaust valve closed	BBDC 45°
Inlet valve open	BTDC 12°
Inlet valve closed	ABDC 48°
Injection type	Direct injection, spray guided, central injector, 30° spray angle
ECU	Orbital Inc.
Positive crankcase ventilation	No
Injection pressure	18 bar

Table 2 Specifications of the fuel injector

Manufacturer	Synergist
Part number/designation	37-152 CNG
Nozzle spray angle	30°
Spring	33.0 N
Stroke	0.135 mm
Maximum operating pressure	2.0 MPa (high pressure)
Turn on time	1.05 ms
Turn off time	0.95 ms
Operating voltage	14.0 VDC
Coil resistance	1.3 O

Table 3 Dynamometer specifications

Make and model	David McClure DC30
Type	Direct current
Capacity	30 kW
Maximum speed	5000 rpm

The main limitation of the hydrogen implementation in ICE engine is its susceptibleness to abnormal combustion. Therefore, the discussion in this chapter starts with the abnormal combustion using hydrogen in relation to ignition timing and is followed by the effect of air–fuel ratio to the combustion performance of H2ICE.

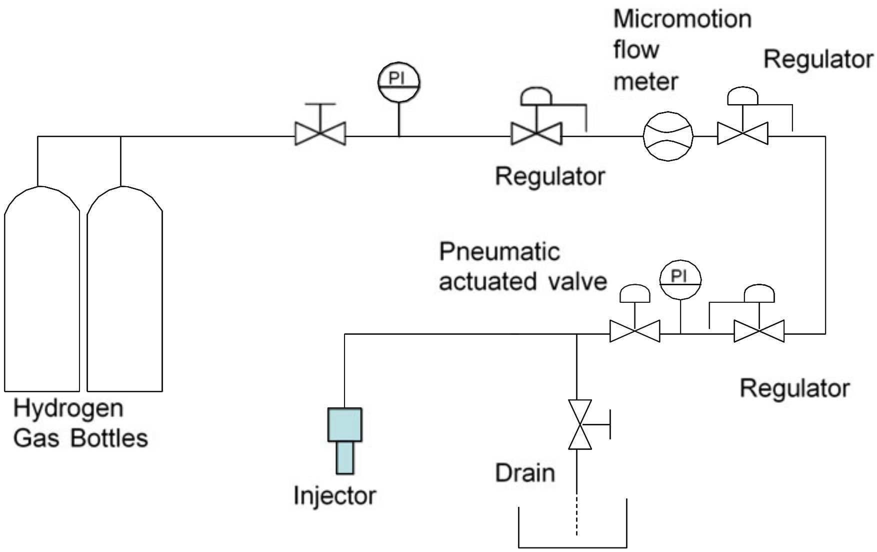


Fig. 3 Fuel supply system

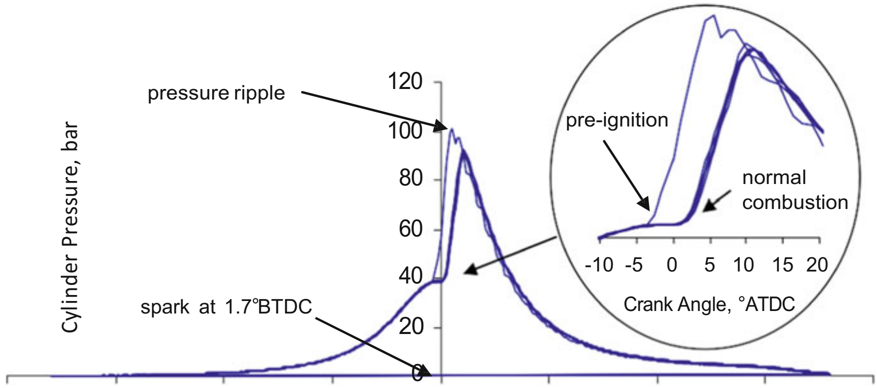


Fig. 4 Engine knocking caused by preignition

Abnormal Combustion in H2ICE

The abnormal combustion is highly affected by the auto-ignition temperature of hydrogen. The auto-ignition temperature of hydrogen is 574 °C, which is higher than that of natural gas (540 °C). However, abnormal combustion in the form of "knocking" was observed during the tests with hydrogen. When the engine was operated under certain conditions, pinging noise was detected accompanied by a sharp rise in the cylinder pressure and the corresponding pressure ripple, as implied in Fig. 4. These were the characteristics of engine knocking.

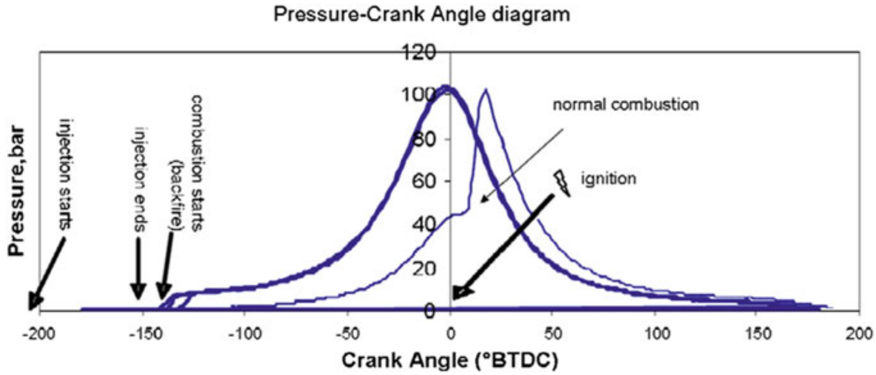


Fig. 5 Engine backfire due to early preignition

As shown in Fig. 4, combustion started prior to the spark. Thus, the knocking observed was thought to be due to preignition caused by hot spots in the engine. As the engine was cleaned prior to the test program, and the engine was inspected to be clean after the tests, the existence of hot spots from engine deposits was ruled out. Other possible sources were spark plug tip, sharp edges in the combustion chamber, or as a result of pyrolysis of engine oil.

At early injection timing and air–fuel ratio close to stoichiometric, there was a tendency for the preignition to start before the intake valve was closed, giving rise to the “backfire” condition. This is shown in Fig. 5. If the engine continues to operate with this abnormal combustion, damage to the engine intake systems will ensue.

These abnormal combustions led to the need to retard the ignition timing during the tests, especially at air–fuel ratios close to stoichiometric. Figure 6 shows the ignition timing for various speeds at different start of fuel injection (SOI). It is evident that for SOI of 300° BTDC, the ignition retards required to avoid preignition was less. This was due to the low volumetric efficiency and the subsequent lower combustion temperatures. The ignition retard was maximum at speed of 3000 rpm, i.e., the speed at which the volumetric efficiency was highest.

Figure 7 shows the variation of the ignition timing with respect to changes in air–fuel ratio. As the air–fuel ratio approached stoichiometrically, the combustion was more prone to preignition. In general, depending on the speed, there was approximately a 3–6° ignition retard needed for every 0.1 λ as it was approaching stoichiometric.

Figure 8 shows the corresponding cylinder pressures for different air–fuel ratios. It is evident that as a result of the retarded timing, the peak pressures were moving away from the typical optimum of about 15° ATDC as the air–fuel ratio was approaching stoichiometric. Thus, the maximum brake torque (MBT) timing was only possible on hydrogen operation at leaner mixtures. At close to stoichiometric, ignition timing was much retarded with a consequent penalty on the engine performance, as shown by the curve for lambda value of 1.16, which indicates that combustion starts very

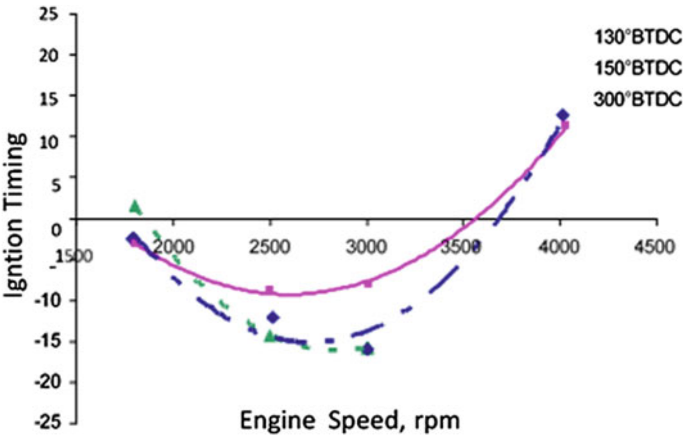


Fig. 6 Ignition timing for start of fuel injection of 130°, 150°, and 300° BTDC

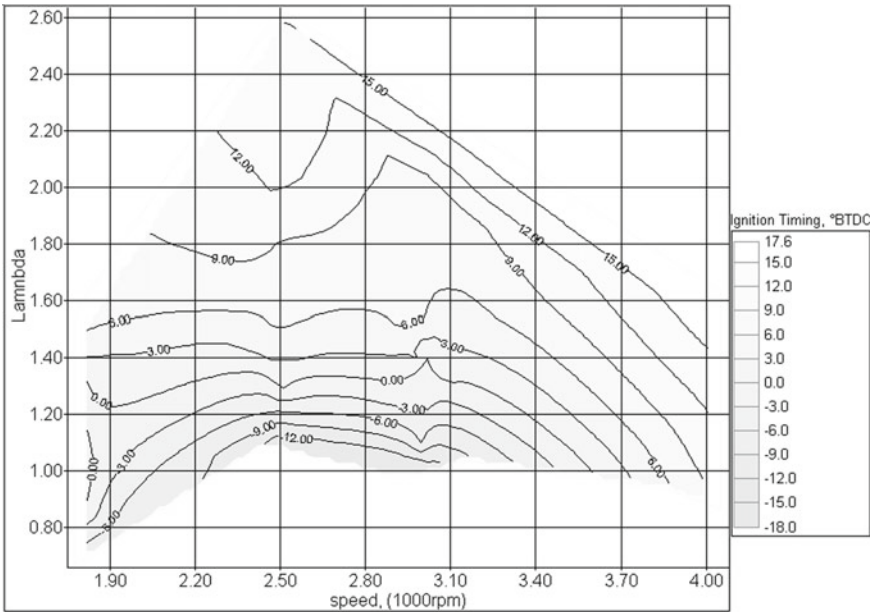


Fig. 7 Ignition timing map for stoichiometric air–fuel ratio of hydrogen

late in the expansion stroke. Thus, it is of interest to investigate whether operating at slightly leaner ratios with MBT timing would offset the performance deficit at stoichiometric.

It is important that for future efforts to adopt hydrogen for this engine, this abnormal combustion issue is addressed. It is even more important with higher engine

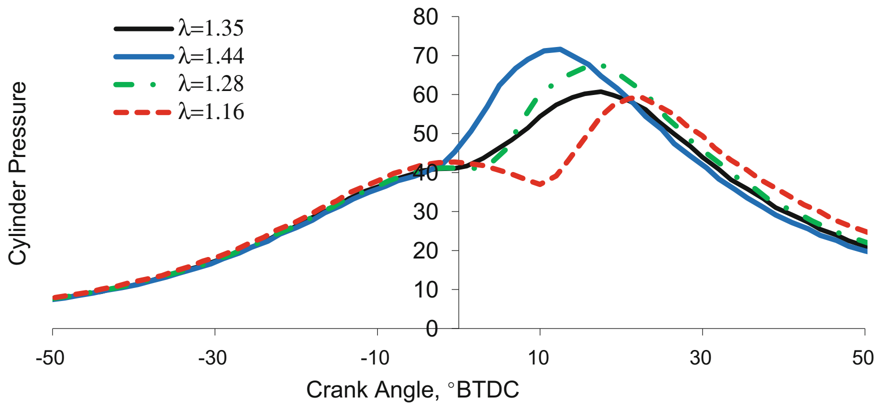


Fig. 8 Cylinder pressure showing the effect of retarding the ignition to avoid preignition at 3000 rpm

speeds as the engine is expected to be running hotter and the risk of melting the engine is higher. For all the tests conducted in this study, the best possible ignition timing without abnormal combustion was used throughout.

Effect of Air–Fuel Ratio to the H2ICE

Figure 9 shows the map of the engine torque across the various speed and the air–fuel ratios. It is worth noting that at leaner ratios ($\lambda > 1.2$), the ignition timing can be advanced and MBT timing can be achieved. However, as shown in the map, the increase in torque was not sufficient to offset the performance drop caused by the cleaning of the intake charge.

The torque map also shows the potential of controlling the power of the engine through mixture quality control method at low loads with the unthrottled operation. It was shown that the engine could be run at least to 50 or 60% load without throttle. The advantage of running the engine lean unthrottled is the increased thermal efficiency.

A map showing the indicated thermal efficiency across the engine speed and BMEP is depicted in Fig. 10. An indicated thermal efficiency of as high as 46% was achievable on this engine. Note, however, as the objective of the study is to improve the performance at full load, tests at higher air–fuel ratios were not conducted. Thus, the possibility of unthrottled operation at much lower loads was not assessed.

The emission of hydrocarbons is depicted in Fig. 11. In general, the emissions concentration is shown to decrease as the air–fuel ratio became leaner. As discussed in the previous section, the emission of hydrocarbon was probably originated from the lubricating oil and was related to quenching distance of the flame. As leaner ratios reduced the speed and lower the flame temperature, the quenching distance was higher. This explains the lower hydrocarbon emissions at leaner ratios.

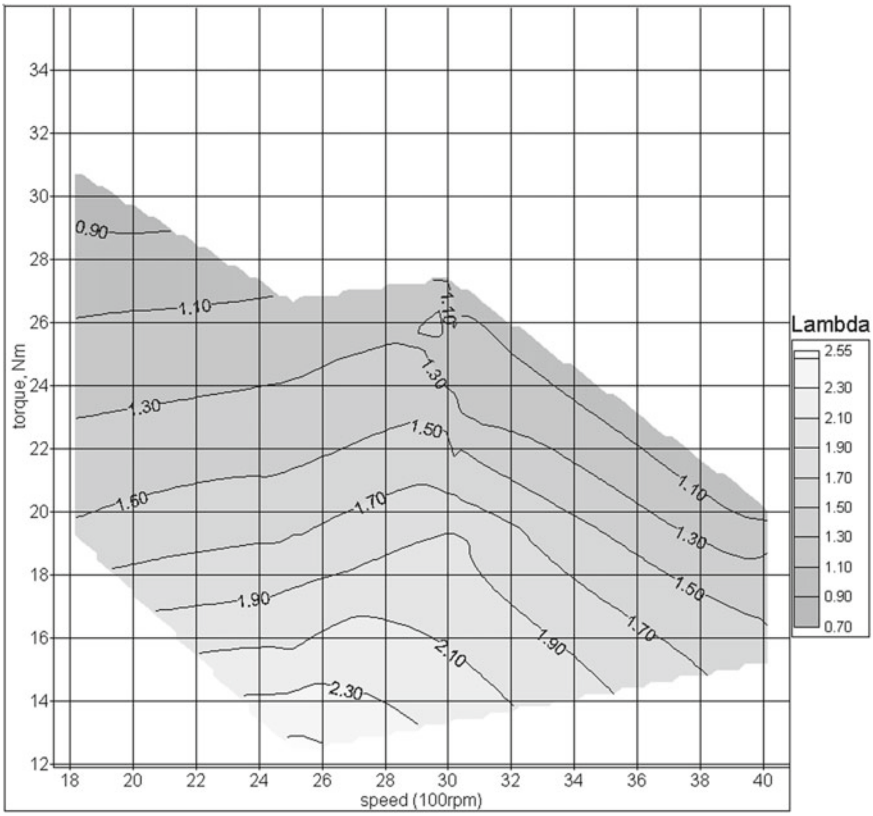


Fig. 9 Contour map of engine torque with respect to speed and air–fuel ratio (SOI = 130° BTDC except at 4000 rpm SOI = 160° BTDC)

As discussed in the previous section, the emission of CO is not significant at lean air–fuel ratio or close to stoichiometric. However, at the slightly rich air–fuel ratio, there were detectable CO emissions of up to 700 ppm. The possible explanation for this phenomenon is that the rich air–fuel ratio caused the flame to quench closer to the walls, resulting in oxidation of the engine oil film. This was more predominant at early injection timing, suggesting that the more homogenous is the mixture, the shorter would be the quenching distant.

NO_x emission variation with changes in air–fuel ratio is also shown in Fig. 11. Overall, NO_x emission was highest at lambda of around 1.1, at 3000–4000 ppm. At lambda ratios of more than 1.3, the NO_x emission was reduced as leaner ratios resulted in lower temperatures, which is also illustrated in the figure showing the peak cylinder pressures. This agrees with the general understanding of NO_x emission trend for air–fuel ratios of hydrogen. At stoichiometric air–fuel ratio, the NO_x emission is similar to that at lambda of 1.3, suggesting that for low NO_x operation, air–fuel ratio higher than lambda of 1.3 is desired.

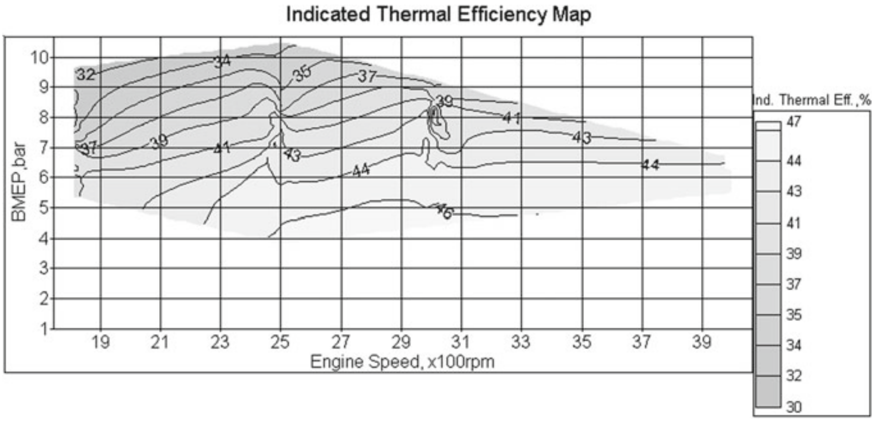


Fig. 10 Contour map of the engine's indicated thermal efficiency

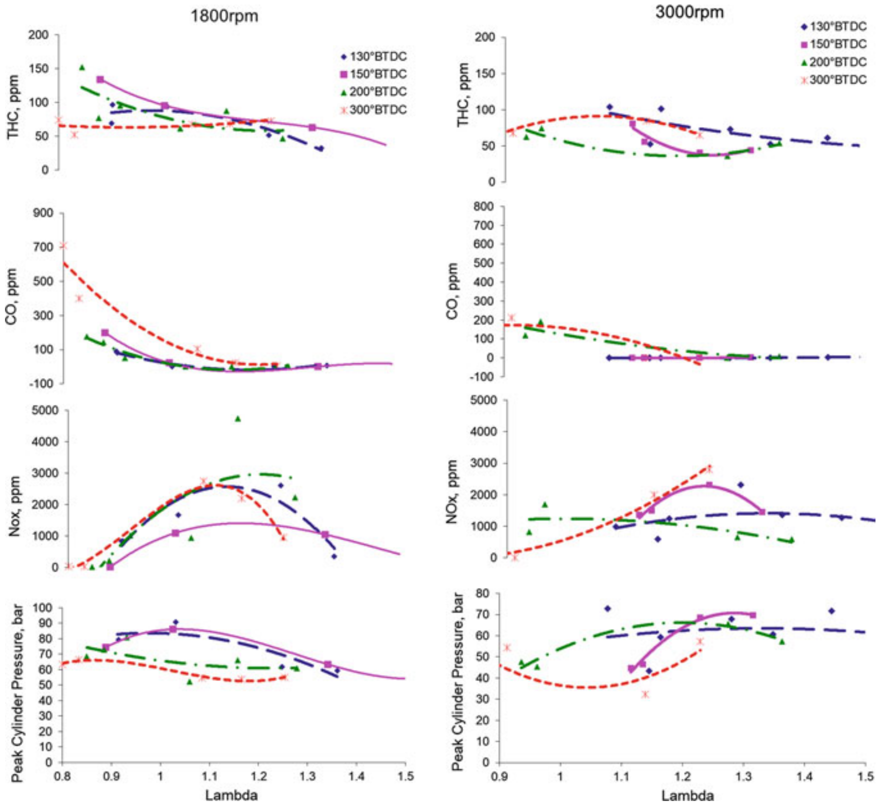


Fig. 11 Engine out emissions when using hydrogen at different air fuel ratios

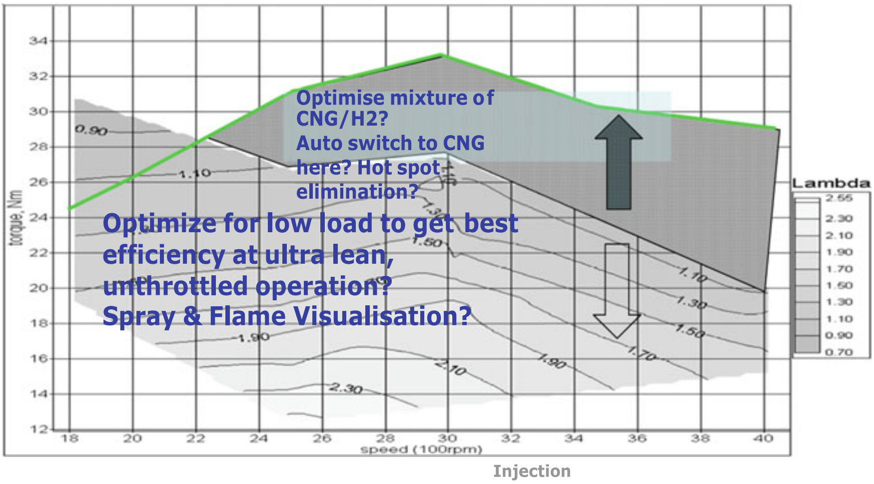


Fig. 12 Recommended future works for optimizing the hydrogen engine

Further optimization is required in order to fully operate the engine in hydrogen mode to cover the whole spectrum of engine operation. Figure 12 summarizes the recommended future works on this engine. In particular, the abnormal combustion needs to be eliminated. Cooler engine operations, reduction of hotspots in the combustion chamber, or running a mixture of hydrogen–natural gas are among the options available to be explored.

Hydrogen and natural gas could be simultaneously introduced into the engine at different ratios through port and direct injection, respectively. The effects on the combustion and flame propagation properties could be an important area of study to further optimize the performance and emissions. Optimum mixture ratios and timing of the injections may be determined to overcome the abnormal combustion while gaining efficiency and performance.

For low load applications, the unthrottled operation is recommended for thermal efficiency gain. In this mode, the injection duration is short, and opportunity exists for optimization in terms of injection parameters. The stratified charge could improve performance and allow very low loads such as idle to be run unthrottled. Flow and combustion visualization, as well as modeling, would help understand the processes so that optimization can be made.

Summary

Direct injection hydrogen enables operation with stoichiometric air–fuel ratio without abnormal combustion at low engine speed, i.e., below 2500 rpm. At higher engine speed, abnormal combustion limits the operation of the engine to leaner ratios or

retarded ignition with significant performance penalty. Although MBT ignition timing can be achieved at slightly lean operation, the performance gained by the timing advance is not sufficient to overcome the loss due to the leaning effect. This is particularly obvious at low engine speed, 1800 rpm. Even though the efficiency and BSFC was minimum at lambda value of 1.2, the torque deficit was almost 5 Nm. Thus, for full load operation, stoichiometric operation is preferred at low engine speed.

The torque and power were relatively constant up to lambda value of 1.2 while the BSFC and efficiency were maximum at engine speed of 3000 rpm. Thus, for this speed, operation at lambda value of 1.2 was optimum.

Lean unthrottled operation of the engine leads to 46% indicated thermal efficiency. This mode of operation is suitable for lower load application such as during cruising. At this load and low engine speeds, the engine performance is less sensitive to the injection timing.

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