

A Hydrogen Direct Injection Engine Concept that Exceeds U.S. DOE Light-Duty Efficiency Targets

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ABSTRACT

Striving for sustainable transportation solutions, hydrogen is often identified as a promising energy carrier and internal combustion engines are seen as a cost effective consumer of hydrogen to facilitate the development of a large-scale hydrogen infrastructure. Driven by efficiency and emissions targets defined by the U.S. Department of Energy, a research team at Argonne National Laboratory has worked on optimizing a spark-ignited direct injection engine for hydrogen. Using direct injection improves volumetric efficiency and provides the opportunity to properly stratify the fuel-air mixture in-cylinder. Collaborative 3D-CFD and experimental efforts have focused on optimizing the mixture stratification and have demonstrated the potential for high engine efficiency with low NO_x emissions. Performance of the hydrogen engine is evaluated in this paper over a speed range from 1000 to 3000 RPM and a load range from 1.7 to 14.3 bar BMEP.

Engine maps show the hydrogen direct injection engine operating above 35% brake thermal efficiency (BTE) over approximately 80% of the tested operating range. A more detailed characterization of engine efficiency is done by quantifying the effects of different loss mechanisms in the engine at relevant points throughout the engine map. The dominant loss mechanism is heat loss to the combustion chamber walls and as a function of both engine speed and load there exists a trade-off between wall heat losses and other partial losses. As a result, the peak BTE is observed at 2000 RPM, 13.5 bar BMEP.

A series of engine maps show efficiency improvements due to optimal injection timing and also show efficiency and NO_x improvements due to injector nozzle design. The most promising engine configuration uses a 4-hole nozzle which shows improvement over the previous 5-hole nozzle. The final engine map shows a peak BTE of 45.5% and part-load BTE of 33.3%, demonstrating the ability of the hydrogen direct injection engine to exceed both U.S. DOE light-duty efficiency targets. The 4-hole nozzle also provides a mixture that is less potent for NO_x than the 5-hole nozzle which correlates to a significant decrease in NO_x emissions at the peak efficiency operating point. The corresponding map of NO_x emissions is dominated by less than 0.10 g/kWh of NO_x.

CITATION: Matthias, N., Wallner, T. and Scarcelli, R., "A Hydrogen Direct Injection Engine Concept that Exceeds U.S. DOE Light-Duty Efficiency Targets," *SAE Int. J. Engines* 5(3):2012, doi:10.4271/2012-01-0653.

BACKGROUND

Hydrogen is envisaged as a sustainable energy carrier because it can potentially be generated using renewable energy sources and it is a relatively clean reacting fuel. To substantiate the use of hydrogen as a transportation fuel, the U.S. Department of Energy (DOE) facilitates research to address different aspects of hydrogen usage. One aspect is the use of hydrogen as a fuel for internal combustion engines (H₂-ICE) which are seen as a cost effective consumer of hydrogen to aid in the development of a larger hydrogen infrastructure. To that end DOE has defined efficiency and emissions targets for H₂-ICEs. These targets include 45% peak brake thermal efficiency (BTE), 31% part-load BTE (2

bar BMEP @ 1500 RPM), and Tier II Bin 5 emissions [1, 2]. The Tier II Bin 5 designation limits NO_x emissions to 0.07 g/mile [3] while all other emissions are negligible for hydrogen combustion [4].

STRATEGY

To meet the DOE targets, a research group at Argonne National Laboratory has been focusing on developing advanced direct injection strategies on a single-cylinder H₂-ICE. Firstly, direct injection (H₂-DI) has been chosen because it has the potential to achieve power density comparable to light-duty gasoline engines [5]. 14 bar BMEP at 3000 RPM is a routine operating point with the H₂-DI engine.

The second reason for choosing H₂-DI is a reduced risk of combustion anomalies such as backfiring, pre-ignition, and knock compared to more conventional hydrogen port fuel injection engines [4]. The research engine has a compression ratio of 12.9:1 to take advantage of the high knock resistance of hydrogen under lean conditions [4, 6]. The combustion chamber has a pent roof head and a flat piston top as shown in Figure 1. The pent roof angle is 140° (for more detail, a surface model of the head is provided in Figure 12). Also shown in Figure 1 are the H₂-DI injector and spark plug which are located near the center of the combustion chamber. The engine is equipped with a piezo actuated injector specifically designed by Westport Innovations Inc. for this application. The piezo injector along with relatively simple combustion chamber geometry offers many degrees of freedom for optimizing the injection strategy using injector nozzle design coupled with injection timing.

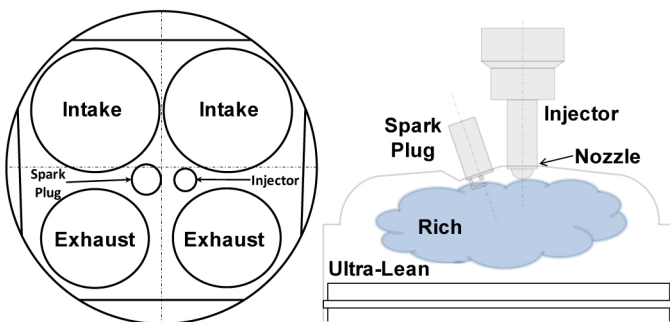


Figure 1. Combustion chamber geometry with a top view (left) showing the head configuration and a cross section (right) showing the flat piston top, injector and spark plug angles, and a depiction of ideal mixture stratification

Figure 1 (right) has a schematic of optimal mixture stratification wherein there is a relatively rich mixture localized near the spark plug and an ultra-lean mixture, ideally air, close to the cylinder walls. A rich mixture localized near the spark plug at time of ignition leads to high burning velocity and increases combustion efficiency. The rich mixture being isolated from the combustion chamber walls also reduces wall heat losses. Optimal mixture stratification was detailed in [7] where a 5-hole nozzle coupled with delayed injection provided an estimated peak BTE of 45.1% at 2000 RPM, 14.1 bar IMEP.

This paper builds on the previously demonstrated potential of the H₂-DI engine by mapping the engine performance throughout a broader range of operating conditions, in addition to the peak BTE point. Engine maps will be shown for an expanded operating range from 1000 to 3000 RPM and 1.7 to 14.2 bar BMEP using an engine control strategy based on the following experimental factors.

EXPERIMENTAL FACTORS

Research Engine

The combustion chamber geometry is relatively simple as mentioned in the previous section (Figure 1) and further engine specifications are provided in Table 1. The single-cylinder H₂-DI has an 89 mm bore and 0.66 L displacement representative of a light-duty automotive engine. The engine is coupled to a 190 hp GE DC electric dynamometer (TLC-3462) which far exceeds the potential output of the single-cylinder engine and provides consistent speed control regardless of engine load. Steady-state engine temperature is maintained regardless of speed and load by controlling the coolant inlet temperature to 85 °C. The oil temperature is not directly controlled but gradually reaches a temperature near 85 °C as it is circulated through the engine block. Since the engine operates at steady-state, contour plots seen in subsequent engine maps are an aggregate of steady-state operating conditions distributed about the operating range of interest (data points are shown on maps).

Table 1. Single-cylinder research engine specifications

H ₂ -DI Engine Specifications	
Displacement [L]	0.66 L
Bore/stroke [mm]	89/105.8 mm
Compression ratio [-]	12.9
Intake valve timing	IVO = 15° BTDC IVC = 35° ABDC
Exhaust valve timing	EVO = 45° BBDC EVC = 10° ATDC
Injector actuation	Piezo
Injection pressure	100 bar

Simulated Turbocharging

Simulated turbocharging was used to maintain lean operation while allowing the engine to operate at loads as high as 14.9 bar IMEP. To simulate turbocharging the single-cylinder research engine has PID controlled throttle valves to govern both the intake and exhaust pressure. The current configuration does not use exhaust gas recirculation (EGR) and does not recirculate blow-by back into the intake system. Surge tanks between the throttle valves and the engine damp out cyclic pressure fluctuations so the pressures being controlled are steady-state operating pressures. Intake air is supplied to the test cell by a large air compressor allowing the intake pressure to be varied freely. The function in Figure 2 was developed to define the corresponding exhaust back pressure in order to simulate the behavior of a well matched turbocharger.

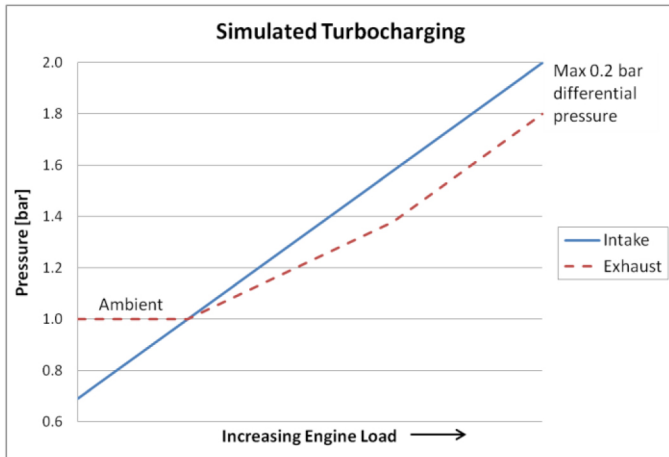


Figure 2. Control strategy used for simulated turbocharging including a range of intake pressures and the corresponding exhaust back pressure

The first part with ambient exhaust pressure represents throttled operation where the turbocharger is not doing any work. Throttled operation is generally undesirable because it increases gas exchange losses but it is necessary in some low load cases in order to avoid operation that is leaner than $\lambda=3.3$. Although hydrogen has wide flammability limits ($0.14 < \lambda < 10$) [8], an excess air ratio of $\lambda=3.3$ was chosen here to be the practical lean limit because combustion phasing becomes challenging to control in leaner mixtures as a result of slower flame speeds. That equivalence ratio ($\lambda=3.3$) was found to be sufficiently rich to allow optimal combustion phasing while lean enough to moderate NOx formation. The same lean strategy was used whenever possible for higher load conditions as well.

With increasing load, the engine was boosted to maintain the desired equivalence ratio. For initial boost the differential pressure increases from 0 to 0.2 bar as the amount of work being done by the simulated turbocharger increases. This is indicative of a wastegate that is progressively being closed until the max differential pressure of 0.2 bar is reached. The differential pressure represents how well the two parts of the simulated turbocharger (turbine and compressor) are matched. The function in Figure 2 was tuned to assure plausible turbine to compressor matching based on typical turbine and compressor isentropic efficiency values (85% and 70% respectively [9]) along with measured intake and exhaust conditions.

The thermodynamic analysis was also extended to assure plausible turbocharger to engine matching. Since exhaust temperature is difficult to predict, measured exhaust conditions were used to calculate the amount of energy available to a turbocharger (exhaust enthalpy). Exhaust port temperature increased with engine speed and load in the range from 230 °C to 420 °C. With the measured exhaust port temperature and without EGR, the exhaust enthalpy was found to be sufficient at each operating condition to provide

the required boost while adhering to the practical limits of a real turbocharger. In Figure 3, the intake boost pressure over the entire operating range is shown. Within this operating range the peak simulated turbocharger net efficiency was around 60% (net efficiency defined as the product of turbine and compressor isentropic efficiencies [10] and mechanical efficiency) [9]. The simulated turbocharger to engine matching is therefore deemed plausible because 60% net efficiency is achievable in modern turbochargers [11].

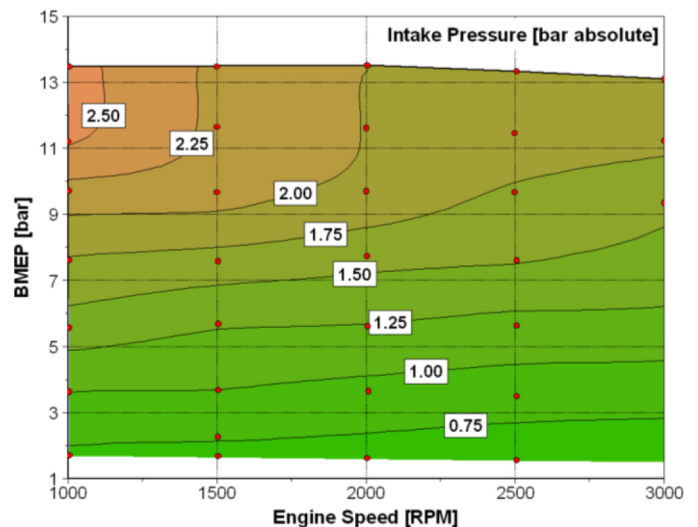


Figure 3. Engine map of intake boost pressure that was required to maintain the constant $\lambda=3.3$ control strategy

It can be seen in Figure 3 that the maximum boost was needed at high load and low engine speed. The general trend is that more boost is needed at lower engine speeds and this observation is likely a function of the intake port geometry, valve timing, and the resulting intake flow dynamics. Boost is limited in the high-load region along the upper bound of the engine map to avoid the peak in-cylinder pressure limit of the engine. A practical peak in-cylinder pressure limit of 140 bar is specified for the research engine to prevent over-stressing mechanical components of the engine such as the connecting rod and crankshaft. The peak pressure limit supersedes the aforementioned constant $\lambda=3.3$ control strategy so the engine operates as rich as $\lambda=2.5$ at high load. Boost being limited at high load contributes to the shape of the boost map in Figure 3 where the isolines curve upward.

Friction

Friction in the single-cylinder engine is greater than for a multi-cylinder engine as a result of the way the engine block, bearings, etc. have been designed for single-cylinder research. Therefore, evaluating the engine concept in terms of brake performance requires an estimate of friction. Since the engine geometry is representative of a light-duty automotive engine, friction values are taken from a friction-optimized multi-cylinder engine that is intended for the same application. Friction mean effective pressure (FMEP) values

shown in Figure 4 are taken from the multi-cylinder engine and apply to the single-cylinder research engine. These friction values as a function of engine speed and load have been universally used to calculate all BMEP and BTE values presented in this paper.

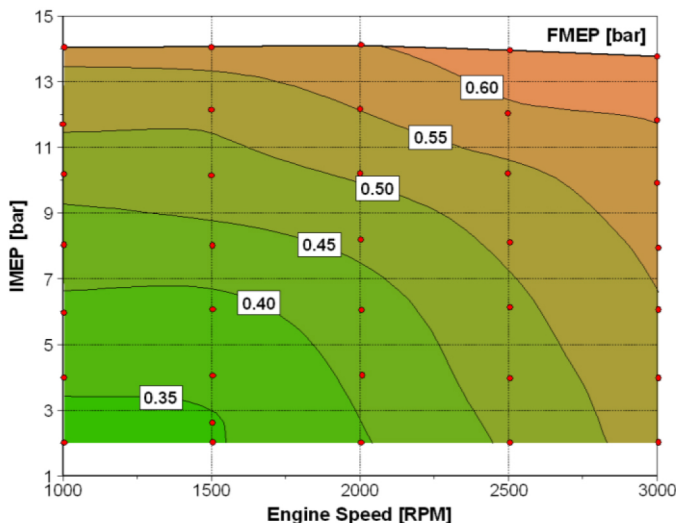


Figure 4. Friction values taken from a multi-cylinder engine that were used to evaluate the brake performance of the single-cylinder research engine

The trend in engine friction, in general, is that it increases with both speed and load. Engine speed affects FMEP because the power loss due to friction between the piston and cylinder increases with piston speed. As engine speed increases, mean piston speed increases resulting in greater friction [12]. The effect of engine load on FMEP is partly due to greater forces in the connecting rod and crankshaft causing more friction in the bearings. Additionally, greater in-cylinder pressure with increasing engine load forces the piston rings against the cylinder walls resulting in greater friction [12]. The relative magnitude of assumed friction losses, calculated based on FMEP values in Figure 4, will be evaluated in the results section in the context of other partial losses within the engine.

Accuracy

Accuracy of empirical ITE values ultimately depends on fuel flow rate and high speed in-cylinder pressure measurements. H_2 flow measurements are made using a Micro Motion CMF010P coriolis meter characterized by uncertainty that depends on flow rate (uncertainty increases for lower flow rates) [13]. Since the fuel flow rate depends on speed and load, so does the uncertainty in ITE. In-cylinder pressure measurements are made using two AVL pressure transducers mounted at different points in the combustion chamber to provide a crosscheck for accuracy to confirm the IMEP stability. The IMEP stability is expected to be <2% [14]. A capacitive TDC sensor was used for crank angle alignment to assure accurate calculation of IMEP.

The worst case scenario for ITE measurements (and the resulting BTE values) would be if IMEP was less than the recorded value and H_2 flow rate was greater than the recorded value. Assuming the worst case scenario, uncertainty ranges from 1.7% BTE at 1000 RPM, 1.7 bar BMEP to 1.1% BTE at 3000 RPM, 14.2 bar BMEP.

NOx measurements are made using a Pierburg CLD PM 2000 NO/NOx Analyzer with a sensitivity of 1.5 ppm, 1% of the measurement range [15]. It is calibrated daily using zero and span calibration gases. The emissions bench is also leak checked daily to ensure the integrity of emissions transfer lines.

RESULTS

PRELIMINARY ENGINE MAP

This section includes the first of a series of maps that will be used to describe the H_2 -DI engine performance through a much broader range of operating conditions than that seen in previous publications [7, 14]. The preliminary engine map provides a baseline for engine optimization. The injection strategy includes one injection, defined by the engine crank angle at the start of injection (SOI). An early SOI time of 140 °CA BTDC has been chosen in this case because it is near intake valve closing and therefore provides the longest possible time for mixture formation without risking combustion anomalies due to H_2 in the intake system. With ample time for mixing, the resulting mixture is relatively homogeneous. Results for this baseline case, in terms of BTE, are shown in Figure 5. In general, the H_2 -DI performs well in terms of efficiency with approximately 80% of its operating range above 35% BTE and a significant island on the map above 43% BTE.

As BMEP decreases so does the engine efficiency. This behavior is expected because the partial losses within the engine do not decrease proportionally and thus account for a greater percentage of the required fuel energy. This behavior is more pronounced in stoichiometric engines as a result of intake throttling; however, throttling does not occur until less than 4 bar BMEP with the lean $\lambda=3.3$ strategy that has been employed. For this reason the efficiency drops off more quickly below 4 bar BMEP due to gas exchange losses. Taking these general observations from the preliminary engine map, the following discussion of partial losses provides a more comprehensive understanding of the H_2 -DI efficiency as a function of BMEP.