

Influence of Injection Strategy in a High-Efficiency Hydrogen Direct Injection Engine

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ABSTRACT

Energy security and climate change are two of the main drivers for development of sustainable and renewable transportation solutions. Entities around the globe have been working on strategic plans to reduce energy consumption and curb greenhouse gas emissions. In this context hydrogen is frequently mentioned as the fuel and energy carrier of the future. The U.S. Department of Energy's (DOE's) FreedomCAR and Vehicle Technologies (FCVT) Program has identified hydrogen-powered internal combustion engine (ICE) vehicles as an important mid-term technology on the path to a large-scale hydrogen economy. DOE has set challenging goals for hydrogen internal combustion engines including 45% peak brake thermal efficiency (BTE).

This paper summarizes recent research engine test results employing hydrogen direct injection with different injection strategies. A dedicated research engine optimized for lean hydrogen operation with a compression ratio of 12.9:1 was built and fast-acting Piezo injectors are used for the test sequence. Turbocharged high-load conditions are analyzed at several engine speeds to provide a full spectrum of relevant engine results for achieving peak efficiencies. Three injector nozzle designs are experimentally tested for their ability to meet the 45% efficiency target and their implication on engine-out NO_x emissions. Complimentary to experimental work 3-D CFD simulation is employed to analyze the mixture formation process and predict the ability of newly designed nozzles to provide favorable charge stratification.

The result of this integrated approach is a combustion system that is capable of achieving high engine efficiencies while minimizing the NO_x emissions penalty. Peak indicated efficiencies of 46-47% were reached at full load conditions translating into peak brake thermal efficiencies of approx. 45%. At the same time engine-out NO_x emissions below 200 ppm could be maintained with certain operating conditions reaching NO_x levels below 100 ppm.

INTRODUCTION

Hydrogen internal combustion engines (H_2 -ICEs) have been studied for decades as a promising approach to reduce harmful emissions from vehicles while meeting the required performance in terms of power output [1,2,3]. In particular, direct injection of hydrogen (DI- H_2) has the potential to achieve high power density while reducing the risk for combustion anomalies compared to a more conventional hydrogen port-fuel injection (PFI) strategy [4]. Research projects on DI H_2 -ICEs, funded by the DOE since 2005, target ambitious goals including the same power density as comparable gasoline engines, 45% peak BTE and NO_x emissions as low as 0.07 g/mile [5].

A research group at Argonne National Laboratory has been focusing on developing advanced direct injection strategies on a single-cylinder H_2 -ICE [6,7] with the goal of increasing engine efficiency by optimizing mixture formation and combustion processes. A recent upgrade of the single-cylinder geometry, consisting of increasing compression ratio and engine stroke, aimed at achieving higher theoretical efficiencies. In addition, the replacement of the solenoid-operated injection system with an improved piezo-actuated system allowed for delivering much higher quantities of hydrogen in the same amount of time [8]. This upgrade also extended the operating envelope to higher engine speeds and loads which is expected to further increase the peak BTE.

The proper injection timing is a key-factor in DI engines and improvements in engine efficiency can be achieved by optimizing the start of injection (SOI) for every condition of engine load and speed. In this paper, the analysis of the influence of engine speed and SOI on the mixture stratification and combustion, and thus on the engine efficiency, is reported. The main focus of this study is on turbocharged operating conditions which result in increased power and BTE and also present the most critical scenario for NO_x emissions. The analysis includes three injector configurations and highlights the significant influence of the nozzle geometry on the mixture formation process. This paper extends the range of operation of

previous work [8] and represents a comprehensive analysis and useful baseline for further optimization of engine parameters and nozzle geometries in engines employing direct injection of a gaseous fuel.

In addition to extensive experimental activities, 3-D CFD simulations were performed to gain insight into the physics involved in the mixture formation process of H₂ DI-ICEs. A commercial CFD code was used to provide an accurate description of the injection and mixture formation process. Numerical results aim at supporting experimental work and providing guidelines for further engine optimization, allowing innovative geometries to be evaluated within hours and thus drastically reducing the test time and cost of prototypes.

BACKGROUND

In order to optimize the efficiency of hydrogen internal combustion engines (H₂-ICEs), it is crucial to analyze the major factors influencing the engine efficiency and the related loss mechanisms. Analysis of losses is a means of identifying individual losses that can be attributed to factors influencing efficiency. The analysis starts with a thermodynamic model of the engine cycle which is used to calculate the efficiency of the ideal engine with real charge. From that, the following series of losses are subtracted to derive the actual indicated thermal efficiency (ITE) of the engine: injection during compression stroke Δ ICS, incomplete combustion Δ IC, real combustion Δ RC, wall heat losses Δ WH, and gas exchange losses Δ GE. The efficiency of the ideal engine with real charge is defined as the theoretical efficiency achievable using measured fuel and air mass while assuming constant volume combustion at TDC, indefinitely short fuel injection at TDC as well as adiabatic processes. The loss due to injection during the compression stroke considers non-ideal injection with actual start of injection (SOI) and finite injection duration. Incomplete combustion accounts for the unburned fuel, which is measured using a hydrogen sensor in the exhaust. The losses due to real combustion account for the actual combustion process with a finite duration as well as phasing different from TDC. Finally, losses due to heat transfer to the cylinder walls during combustion as well as gas exchange are calculated.

Previous work on the single-cylinder H₂-ICE at Argonne National Laboratory was aimed at improving the engine efficiency by upgrading the geometry [8]. The upgraded engine compared to its predecessor features an increased compression ratio along with a decreased bore/stroke ratio (see Table 1). ITE improved by approximately 6% over the previous engine configuration and a detailed analysis of the partial losses showed that the increase was mainly due to higher theoretical efficiencies and reduced wall heat losses. With the upgraded engine, an ITE of 43.0% was demonstrated for the engine under naturally aspirated conditions (2000 RPM, 8 bar IMEP) and a peak ITE of 45.6% was achieved under turbocharged conditions (3000 RPM, 14.3 bar IMEP). These results also suggest that the DOE target of 45% peak brake thermal efficiency is within reach and

can be achieved through minimizing each of the above-mentioned losses. Progress documented in [8] provides a foundation for the current work, aimed at further improving efficiency by studying the effects of injection strategy and injector nozzle geometry.

Table 1: Research engine geometry

Displacement [L]	0.66
Bore/stroke [mm]	89/105.8
Bore/stroke ratio [-]	0.84
Compression ratio [-]	12.9
Intake valve timing	IVO = 15 °CA BTDC IVC = 35 °CA ABDC
Exhaust valve timing	EVO = 45 °CA BBDC EVC = 10 °CA ATDC
Injector actuation	Piezo

To enable optimization of injection strategy, an advanced fuel delivery system equipped with a piezo-actuated injector was tested. The specific injector used was a prototype injector provided by Westport Innovations, Inc. Faster acting piezo-injectors with higher mass flow rates allow greater flexibility in terms of injection strategy because the injection duration is shorter than with the previous generation of solenoid injectors. Thus, the engine can be operated at higher speeds and loads and the injection can start later in the compression stroke. The evident benefit of a late SOI is the reduction of the compression work.

The potential for efficiency improvement due to later injection has previously been assessed [3,9]. Figure 1 shows the theoretical losses due to injection during the compression stroke as a function of both SOI and air fuel ratio. Injection duration and compression ratio are held constant at 40 °CA and 10.5 respectively. Losses decrease with later SOI and reduced amount of injected fuel represented as increased air fuel ratio. The plot suggests that at $\lambda=2$ the losses decrease by approximately 0.2% if SOI is delayed from 140 °CA BTDC to 120 °CA BTDC and by another 0.4% if SOI is further delayed to 100 °CA BTDC.

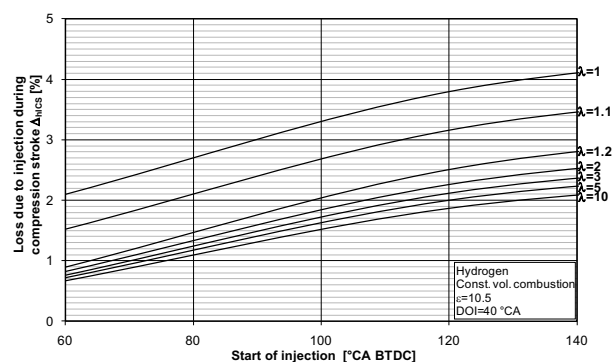


Figure 1: Theoretical losses due to injection during the compression stroke [9]

Nevertheless, delaying the start of injection also affects the mixture formation process. Another effect of a late SOI is an increase in mixture stratification. Delayed SOIs provide a shorter time for the fuel jet penetration within the combustion chamber and proper mixing with air. With insufficient time for complete mixing, part of the charge may be

stoichiometric or even rich at ignition timing even though the global equivalence ratio is lean (typical values of equivalence ratio in this study are $\phi \sim 0.4/\lambda \sim 2.5$). However, a proper stratification of the mixture around the spark plug at the ignition time can increase the combustion efficiency (rich mixture close the spark plug) while maintaining low losses due to wall heat transfer (ultra-lean mixture close to the cylinder walls). This research focuses on how to take advantage of the mixture stratification due to direct injection of hydrogen during the compression stroke and analyzes the effect on individual losses as well as NO_x formation.

Incomplete combustion accounts for hydrogen that leaves the cylinder without participating in the combustion reaction. Major sources determining the amount of unburned fuel include the crevice volume between the piston and cylinder walls, mixtures outside H_2 flammability limits, and flame quenching close to the combustion chamber walls. In general H_2 has wide flammability limits, defined by equivalence ratios between $0.1 < \phi < 7.1$ ($10 > \lambda > 0.14$) [3]. Ideal mixture stratification would provide mixtures within the H_2 flammability limits and negligible amount of H_2 close to the walls or trapped in the crevice volume in order to avoid incomplete combustion.

Losses due to real combustion account for the fact that the combustion process is not instantaneous. In the ideal engine it is assumed that constant volume combustion occurs at TDC, which is not physically possible. The amount of time required for combustion depends on many factors including the flame speed which is largely a function of equivalence ratio. In general, rich mixtures burn more quickly than lean mixtures [4]. Accordingly, faster reactions (i.e. richer mixtures) will minimize the loss due to real combustion. Therefore, as above-mentioned, ideal mixture stratification would have a rich mixture close to the spark plug which rapidly develops the first flame kernels and spreads the flame front throughout the combustion chamber. Another important factor in the analysis of losses due to real combustion is the combustion phasing. If one does not consider wall heat losses, the ideal phasing of the combustion would be symmetrical around TDC. However, taking wall heat losses into account a combustion phasing with a 50% mass fraction burned (MFB) location around 8°CA ATDC is widely considered optimal.

Wall heat losses account for heat transfer from the in-cylinder gases to the cylinder walls. The driving force is the temperature gradient between the combustion gases and the cylinder walls. In other words, high temperature gases in close proximity to the walls result in high heat transfer. Measurements of wall heat transfer on a hydrogen engine suggest that disadvantageous mixture stratification results in a non-linear increase in local wall heat transfer [9]. Furthermore, due to the shorter quenching distance, wall heat losses with hydrogen are potentially increased compared to gasoline operation. Assuming the highest temperature gases are those resulting from rich H_2 combustion, ideal mixture stratification would avoid rich mixtures close to the cylinder walls.

In order for hydrogen engines to be a considered a viable solution they must meet the most stringent NO_x emissions regulations preferably without after-treatment systems. To this aim it is important to understand the effect of mixture stratification on NO_x emissions. Homogeneous H_2 combustion is a good point of reference for evaluating NO_x emissions. Figure 2 shows the NO_x emissions trend for homogeneous lean operation as a function of equivalence ratio. From this trend it is obvious that lean H_2 combustion produces very little NO_x and there is a critical equivalence ratio where NO_x increases greatly. The critical level is approximately $\phi = 0.5$ ($\lambda = 2$), where a sharp increase in NO_x followed by a decrease approaching stoichiometric combustion can be observed [4]. Applying this knowledge to stratified combustion, an overall lean mixture ($\phi < 0.5$) with rich pockets ($\phi > 1.0$) is expected to be beneficial in terms of NO_x , compared to a homogeneous mixture at $\phi \sim 0.75$.

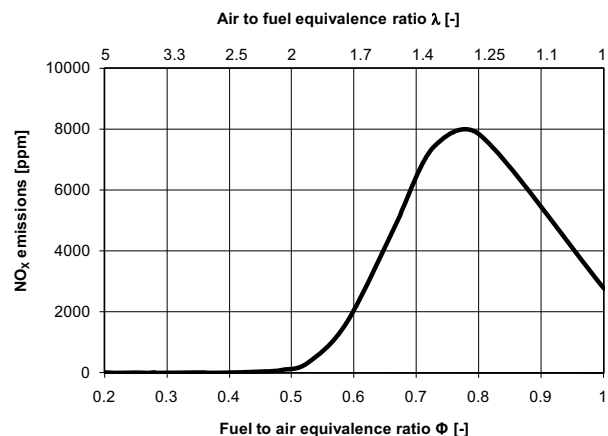


Figure 2: NO_x emissions trend for homogeneous H_2 combustion [4]

Overall, H_2 -ICEs show great potential to meet the efficiency and emissions targets set forth by DOE. Improved engine geometry and fast-acting piezo-actuated injectors have shown promising results in previous work [8]. With this foundation a considerable effort is being made to optimize the injection strategy with specific focus on mixture stratification. Ideal mixture stratification has been conceptualized and the approach to achieving such stratification through an iterative process including design, assessment employing 3D-CFD simulation and experimental testing of different injector nozzles is summarized in this paper.

NUMERICAL AND EXPERIMENTAL DETAILS

3-D CFD SIMULATION

This paper includes CFD results on direct injection of hydrogen in addition to experimental results, in order to provide further insight in the processes occurring within the combustion chamber during the mixture formation and compression stroke. Details on the numerical approach employed in this study have already been published [10] and are only briefly discussed here. In this work, numerical simulation

focus on direct injection during the compression stroke, thus only a portion of the actual engine geometry (including combustion chamber and nozzle configuration) is considered and only a fraction of the engine cycle (from IVC to TDC) is analyzed. Nevertheless, the flow field and the main physical quantities at IVC (initial conditions) are taken into account and are the result of previous calculations which are not reported here. Such conditions are likely to be stable at a fixed engine speed and load, since conditions in the intake and exhaust zones are mostly unvaried and the injection timing (SOI) or the nozzle geometry has no significant effect on the in-cylinder pressure at EVO. Previous work [10] validated the flow-field numerical predictions against experiments in terms of PIV data in a similar, but optically accessible, engine.

With the geometry defined and initial conditions being known, the only information which has to be set as input to the computational code is the injection profile, which is calculated on the basis of experimental data (mass of fuel injected and injection duration) with the assumption of a trapezoidal injection law. This approach has demonstrated the capability to accurately simulate jet penetration (quantitatively) and air/fuel mixing (qualitative) based on the validation against optical data [10,11]. Simulating injection in this way has been proven effective in previous work as it was used to improve the analysis of engine performance and highlight the effect of injection timing and nozzle configuration on mixture formation [8].

NOZZLE DESIGN AND GEOMETRY

A schematic view of the combustion chamber of the single-cylinder research engine used for this study is shown in Figure 3. The injector is placed in a central location close to the spark plug, although its location is not perfectly symmetric with respect to the cylinder. The schematic also shows the injector protrusion and its location relative to the spark plug on a vertical plane through the injector and spark plug axes.

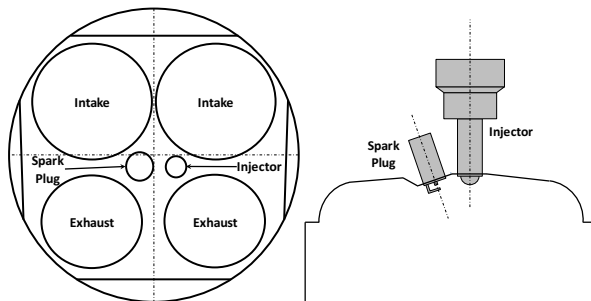


Figure 3: Schematic of the research engine geometry

The central location of both injector and spark plug presents a challenging scenario for optimizing mixture formation through design of injector nozzle and injection sequence. Under these boundary conditions properly stratifying the gaseous jet around the spark plug is not trivial. It is worth noting that for the majority of results presented in this study a hydrogen injection pressure of 100 bar was used. Such a pressure

results in supersonic jets with velocities downstream of the nozzle in the order of 1250 to 2500 m/s (in the under-expanded region $Ma \gg 1$). Accordingly, even with directing the hydrogen jet directly towards the spark plug it is not possible to localize most of the hydrogen around the spark plug at ignition timing.

Table 2: Injector nozzles specifications

Nozzles Configuration			
Number of holes	5	13	2
Injection angle (°)	35	30	70
Hole diameter (mm)	0.61	0.313	1
Total flow area (mm ²)	1.48	1.00	1.57
Instant. H ₂ mass flow (at 100 bar) (g/s)	5.83	4.57	6.87
Inj. duration (3000 RPM 14bar IMEP) (°CA)	54.5	67	47

Thus, the choice of the nozzle geometry is a consequence of the injection strategy, which can provide jets impinging the piston, the cylinder walls or spreading throughout the combustion chamber. Three different nozzles were designed and tested in this study, characterized by 2, 5 and 13 holes (Figure 4). Main specifications of the nozzle geometries are provided in Table 2. A previous study used a 5-hole injector to provide a homogeneous mixture at ignition timing with early injection (SOI=140°CA) [8]. The 2-hole and 13-hole nozzles were subsequently designed with the goal either to increase the mixture stratification (2-hole) or to increase the air entrainment in the hydrogen jets (13-hole).

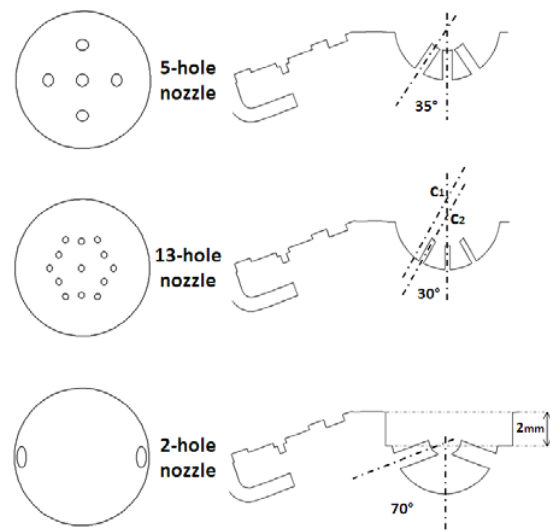


Figure 4: Schematic geometries of the examined injector nozzles

The 13-hole nozzle was designed with a central hole and 12 holes at 30° angle with respect to the vertical and with a zig-zag distribution due to the different location of the rotation center (see Figure 4). This nozzle was expected to show strong jet-to-jet interaction which ultimately resulted in a single jet aiming towards the piston (Figure 5). The 2-hole nozzle was designed with a significantly wider angle with respect to the vertical (70°) and without central hole for two main reasons. First, any possible interaction between the jets is avoided, furthermore the effect of a different strategy (jet impinging the cylinder walls and interacting with the piston as it