2023-01-0408 Published 11 Apr 2023

SAE INTERNATIONAL®

Potential of a Hydrogen Fueled Opposed-Piston Four Stroke (OP4S) Engine

Philip Zoldak and Julien Douvry-Rabjeau Enginuity Power Systems

Antowan Zyada Wayne State University

Citation: Zoldak, P., Douvry-Rabjeau, J., and Zyada, A., "Potential of a Hydrogen Fueled Opposed-Piston Four Stroke (OP4S) Engine," SAE Technical Paper 2023-01-0408, 2023, doi:10.4271/2023-01-0408.

Received: 18 Nov 2022 Revised: 26 Jan 2023 Accepted: 31 Jan 2023

Abstract

The aim of this study is to develop a pathway towards Hydrogen combustoin on an opposed-piston four stroke engine (OP4S) by using 1D simulation code from Gamma Technologies. By its configuration, the OP4S engine has significant thermal efficiency benefits versus conventional ICE. The benefit of the OP4S is reduced heat losses due to elimination of the cylinder head, which increase the brake thermal efficiency.

A hydrogen-fueled (H2) opposed-piston four stroke (OP4S) engine was modeled using GTPower to determine the potential on performance, thermal efficiency and emissions targets. The 1D model was first validated on E10 gasoline using experimental data and was used to explore changes to fuel type in NG and H2, fueling location (TPI and DI), fuel mixture strength (stoichiometric and lean), for an optimized plenum volume and turbocharger selection.

The impact of these changes on volumetric efficiency, rated power, brake thermal efficiency and finally emissions for naturally aspirated and boosted conditions was determined. The simulation study demonstrates an engine design strategy for H2 fueled OP4S to meet power target of 20kW, brake thermal efficiency target of 40% and US EPA-Class II emission regulations for non-road small SI engine. The results also found that with a boosted stoichiometric burn direct injection H2 strategy combined with the OP4S can meet all performance and emission targets.

Introduction

Global decarbonization efforts have mandated zero carbon emission vehicles by 2035 and have led several automakers to focus on battery electric vehicles (BEV). One area that is hindering widespread market adoption of BEVS is range anxiety. Thus, automakers have been working on hydrogen fuel cell electric vehicles (FCEV) which require the utilization of costly pure hydrogen (>99.0% pure H_2). Alternatively, lower purity hydrogen (~90% H_2) could be utilized effectively in ICE's (H_2-ICE) and have potentially a wider market adoption because the technology can be applied to both new production engines as well as retrofits of existing IC marketplace thus having the potential for immediate and wider impact on global CO2 reduction.

Additionally, BEV with range extenders called New Energy Vehicles (NEVs) are being considered for certain markets such as China. This has the potential to extend the existence of the internal combustion engine as a reliable, cost-effective, and zero carbon emission platform for the foreseeable future.

Hydrogen (H_2) has garnered significant attention recently due to its ability to burn cleanly (ultra-clean with minimum NOx production) and operate at high efficient levels to do its knock avoidance [1]. This requires a dedicated supply chain for hydrogen production, storing, and transporting to make it available around the world. A driver for adapting this technology would be the robustness in operation and the cost and changes in regulations compared to alternative energy sources.

Background

Hydrogen Properties

Challenges for H_2 as an energy source are the lowest density (0.09kg/m^3 compared to 0.72kg/m^3 for methane and 730-780kg/m^3 for gasoline), low volumetric energy density (which combine affect the size of the fuel tank), low lubricity, high absorption capacity, and high volatility. Hydrogen can be produced from fossil fuels and sustainable methods in many ways: from natural gas, biomass, or wind and solar energy through electrolysis of water, and the method of production.

The ignition limits of hydrogen are between 4% and 75% by volume, which means an air mixture strength of Lambda within 0.15 and 10:1. By comparison, methane is able to burn between a lambda of 0.7 to 2.1 or 5.3 to 15% by volume, and gasoline between 1 and 7.6% of volume (lambda 0.4-1.4).

Additionally, hydrogen burns much faster than any other fuel with a laminar flame speed is equal to 2m/s at lambda 1) than conventional fuel (0.4m/s for methane and 0.4-0.6m/s for gasoline) which contributes to higher thermal efficiency due to the shorter burn duration. Another advantage is the lower heating value of 120MJ/kg is much higher than methane (50MJ.kg) and gasoline (43.5MJ/kg). The stoichiometry of hydrogen is twice higher than methane (34.3 for hydrogen compared to 17.2 for methane).

However, due to Hydrogen fuel have fast flame speeds and low flammability limits it results in a tendency to pre-ignite leading to engine knock, which can cause backfire in the intake and exhaust manifold, pre-ignition, and rapid rise of pressure rates when this occurs during combustion. The minimum ignition energy of hydrogen cannot show that tendency as it is much lower than other fuels (0.02mJ versus 0.29mJ for methane and 0.24mJ). The auto-ignition temperature of hydrogen is 585C, which is higher that other fuels, such as methane (540C) and gasoline (350C). Another advantage for hydrogen is that it has a very high RON of >130.

Compared to hydrocarbon fuels, hydrogen has no carbon content and during combustion only produces water in the products of combustion. This means the exhaust emission doesn't contain CO, CO2, particulate matter (PM), neither unburned hydrocarbon (UHC) [2]. During idle, some traces of UHC are found in hydrogen emission due to the participating of lubricating oil in combustion. Hydrogen combustion at very lean AFR reduces the adiabatic flame temperature which leads to low NOx emissions, therefore NOx emissions are produced in low quantities in the exhaust when operating with lambda 2 or greater [3]. Lean H_2 combustion produces very little NOx and there is a critical equivalence ratio where NOx increases greatly. The critical level is approximately λ=2, where a sharp increase in NOx followed by a decrease as we approach stoichiometric combustion λ=1 can be observed [3].

Relevant Gaseous Fueled ICE Work

Hydrogen ICEs are currently can be categorized in 2 basic types based on their point of injection in the induction system. H2 throttle port injection (TPI) and H2 direct injection (DI). H2 combustion can occur at various air fuel ratios from stoichiometric (λ=1) to ultra-lean burn combustion λ=3.

H2-TPI provides the simplest way to convert a conventional PFI SI engine to Hydrogen. Due to the higher flame speed of hydrogen compared to gasoline fuel, the ignition timing should be retarded by up to 40 CAD compared to that of the operation at similar conditions with gasoline [2]. Lee et al. showed in their research that it is possible to fit a hydrogen port fuel injection system directly to an SI engine with minor modification [4]. To avoid lower volumetric efficiency issue due to argon replacement by hydrogen, fuel injection is retarded and timed prior to intake valve close timing. To overcome backfire issue, lambda is kept very lean a $\lambda > 2$ [5]. Boosting is beneficial here to overcome the impact on volumetric efficiency and improve power of the engine.

In-cylinder direct injection of NG was studied extensively in both experimental and simulation studies by Zoldak [6] and Zoldak & Naber [7, 8, 9, 10]. In these works, it was demonstrated that in particular late DI of gaseous NG fuel enabled a new mode of NG combustion known as partially stratified combustion (PSC). This mode was enabled by a high pressure (>50bar) DI injection, late DI injection timing (50 to 100degBTDC), and lean burn operation (lambda >1.5) and central spark ignition. This mode was characterized by higher thermal efficiency due to the volumetric efficiency benefit of late DI, in-cylinder mixture stratification of the gas and air as well as higher turbulent kinetic energy (TKE) at start of combustion (SOC) compared to early DI or PFI combustion modes. It is believed the late DI of H2 will have similar benefits if not better performance than late DI of NG in the PSC combustion mode.

In-cylinder H2-DI can overcome the impact to volumetric efficiency experience with H2-TPI. Several research works confirm this to be the case [14, 15, 16, 17]. Using a DI system allows more fresh air to be inducted into the cylinder, and prevents hydrogen from displacing this air, which leads to improved power density [13]. The DI injection can be timed after the closure of the intake valve to avoid backfire. Additional benefit of H2-DI showed improvement compared to TPI in terms of pre-ignition and knock resistance, which is attributed to stratification of the H2 fuel and air [1].

The H2 DI injector plays a large role in achieving the optimum engine performance and efficiency as well as low emission targets. Key considerations for injector design include good homogenization within a narrow injection window and thus fast and targeted quantity input, a large quantity spread (idle to full load), dry-running capability, and very good internal and external tightness. Furthermore, injector packaging into the engine must consider the size of the injector must be designed in such a way, that as few changes as possible must be made to existing cylinder head concepts.

Opposed Piston Four Stroke Engines

Opposed piston four stroke (OP4S) engines have inherent thermodynamic advantages over conventional four-stroke (4S) engines as reported by Wahl et. al. [12]. They compared both 4S and opposed piston four stroke (OP4S) engine configurations, it found it that the primary factor leading to the increased thermal efficiency for the OP4S engine is reduced heat transfer. In their modeling work, the efficiency loss due to heat transfer decreased 2.2% of fuel energy input. The reason for the decreased heat transfer is attributed to the significantly smaller surface area to volume ratio for the OP4S engine compared to the 4S. A result of combining two of the 4S cylinders into one OP4S cylinder and thereby removing the cylinder head. By reducing heat transfer losses, the fueling rate required to achieve the power target is reduced. Additionally, the reduced fueling rate allows for the combustion duration to be reduced slightly while maintaining the desired maximum pressure rise rate. The burn duration can be 1.1 degrees shorter for the OP4S engine compared to the

4S engine at the same maximum pressure rise rate, a change that decreases the loss in efficiency due to finite duration combustion by 0.2 %fuel.

Previous work by Zoldak et.al. simulated and validated a model of the Enginuity Power Systems (EPS) 1L OP4S engine on E10 gasoline with boosted Natural gas operation [11]. The simulation study showed that the EPS 1L OP4S had the potential for 58kW of brake power at 3000rpm and a 12% improvement in brake thermal efficiency over E10 gasoline baseline SI mode.

Objective

The objective of this study is to demonstrate the potential for a small H_2-DI fueled OP4S 1L engine for residential power generation. The engine will be required to meet specific targets listed below in Table 2. Targets for this study are brake power of 20kW and engine out combined emissions NOx and HC that meet small engine US EPA targets of a BS(NOx+HC) < 8g/kWhr [18].

Technical Approach

The technical approach for this study was as follows:

1. Build and Validate model with E10 Fuel PFI NA
2. Switch to NG-TPI NA and Boosted (NG-TPI-TC)
3. Conduct plenum sweep with boosted (NG-TPI-TC)
4. Switch to boosted NG DI and optimized plenum
5. Switch to H_2 boosted DI with optimized plenum
6. Explore Lambda and Compression Ratios impact

The study was performed at WOT with speed ranging from 1800rpm to 3000rpm. The study assumed no aftertreatment system would be applied to the engine, therefore sweeping lambda away from stoichiometric λ=1 to ultra-lean burn λ=3 would be used to lower NOx to acceptable tailpipe out levels. Although effective at reducing NOx, EGR was not studied in this work since the focus was on developing a lean burn system. The plenum volume was optimized to improve volumetric efficiency, other intake system parameters were also studied but were not included in this report. The impact on volumetric efficiency, brake power, and brake thermal efficiency on emissions were reported.

TABLE 1 Targets for study

Parameter	Target
Rated Power (kW)	20kW @ 3000rpm
Brake Thermal Efficiency (%)	40 %
US EPA Category	Non-road SI engine for stationary power Class II. [18]
BS (NOx + HC) limit (g/kWhr)	8.0
CO limit (g/kWhr)	610

The Enginuity 1L OP4S Engine

The 1L OP4S two-cylinder engine was designed EPS and is presented in Figure 1 below. Advantages the engine features a 2-cylinder opposed piston design with conventional poppet valves for both intake and exhaust. The engine can operate in SI mode on several fuels: gasoline, ethanol, NG, propane, JP8, H_2 and plans are underway to develop single cylinder 0.5L variant and compression ignition variants. The engine specifications are presented in Table 2.

Model Build

The EPS 1L OP4S two-cylinder model was built using GTPower V7.1 with details shown in Figure 2. The OP4S model included the 3D intake air filter, intake throttle, 3D plenum, intake runners, intake ports as well as intake valves. The exhaust manifold was simply routed to open atmosphere so therefore no exhaust outlet restriction applied. The combustion model was set to Wiebe SI model and was subject to the opposed piston slider crank mechanism which was imposed as a mathematical geometry relationship of the relative piston motion to the volume as function of crank angle. The TPI injection model was configured with an air fuel ratio feedback

FIGURE 1 EPS 1L OP4S 2 Cylinder engine.

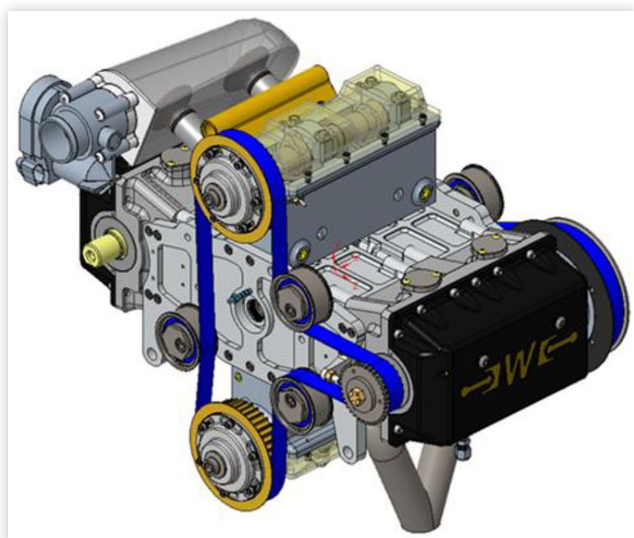
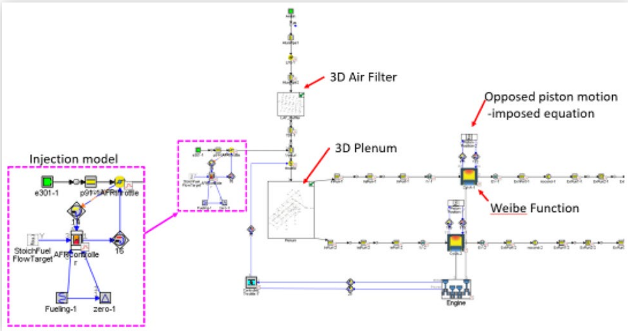


TABLE 2 Engine Specifications

Engine	EPS 1L OP4S
Compression Ratio [-]	8.5:1
Displacement [cm^3]	1.0L
# of cylinders	2
Fuel System	TPI-NG, TPI-H_2, DI-H_2
Boost System	Natural aspirated and turbocharged using GT08R
Rated Power (kW)	20kW @ 3000rpm

FIGURE 2 1D GTPower model of EPS 1L OP4S engine using E10 gasoline.



controller to the boundary condition setpoint. The initial fuel model for validation was set to the E10 gasoline model, however subsequent fuels were set to natural gas and hydrogen.

Model Validation

The results in Table 3 show a comparison between engine test data set[1] and the E10 gasoline simulation results at 3033 rpm with a 180deg firing order. The power for the simulation data was 24.88kW, meanwhile the power for the test data set was 25.03kW only a 0.6% difference for brake power for nearly an identical lambda set point.

Figure 3 shows the high-speed cylinder pressure data comparison for cylinder 1 and cylinder 2. Cylinder 1 maximum pressure value matches well for both test and simulation, whereas Cylinder 2 shows a slight gap at max cylinder pressure. That differences are mainly due to the side intake runner port which cannot give good air partition between both cylinders. Central plenums have been simulated in subsequent work and the results showed a better air flow partition and a smaller difference between the cylinder pressures. See Appendix for more details. Overall, the E10 gasoline model shows good agreement with the engine test data.

The validated E10 gasoline GTPower model of the 1L OP4S was updated with a turbocharger model as shown in Figure 4. The model was updated for a 360degree firing order and the fuel type was changed to NG, NG fueled model has a fuel composition of 88% methane, 6% ethane and 6% propane with a LHV of 49.6 MJ/kg. The turbocharger and intake system

TABLE 3 Simulation Validation Results – 1D GTPower model of 1L OP4S on E10 Gasoline

Parameter	Units	Test Data	Sim Data
Fuel Type		AKI 87 E10 Gasoline	10% ethanol 90% indolene
LHV	MJ/kg	41.93	41.93
Throttle Position	Degrees	90 (WOT)	90 (WOT)
Speed	Rpm	3033	3033
Torque	Nm	78.82	78.83
Power	kW	25.03	24.88
Lambda		0.918	0.919

FIGURE 3 In-cylinder pressure validation, test data versus simulation data for E10 gasoline run.

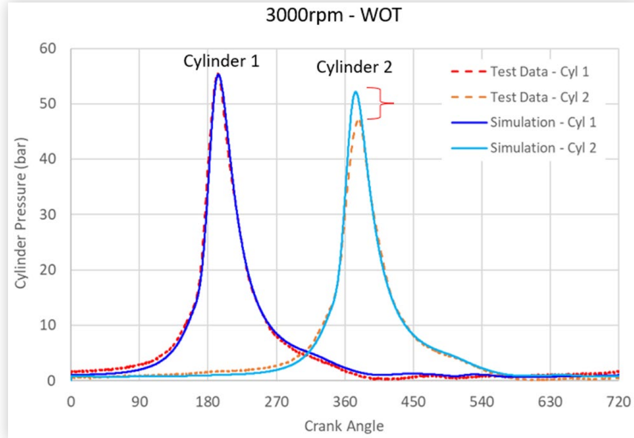
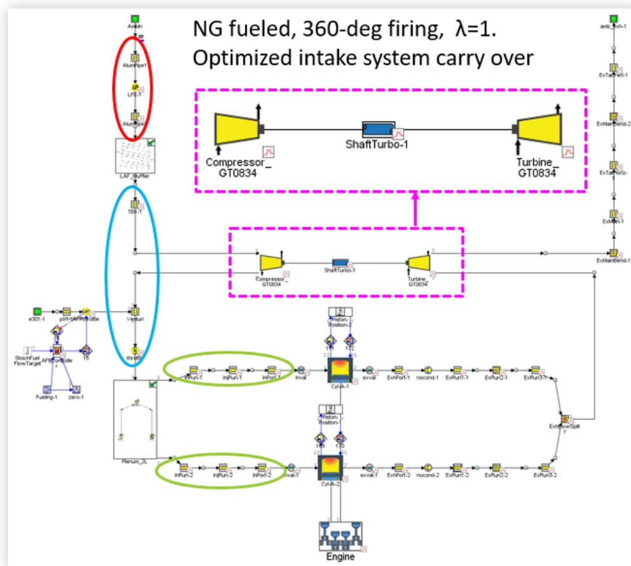


FIGURE 4 Turbocharger NG TPI model for the 1L OP4S engine.



optimization were carried over from previously published work by the authors [11].

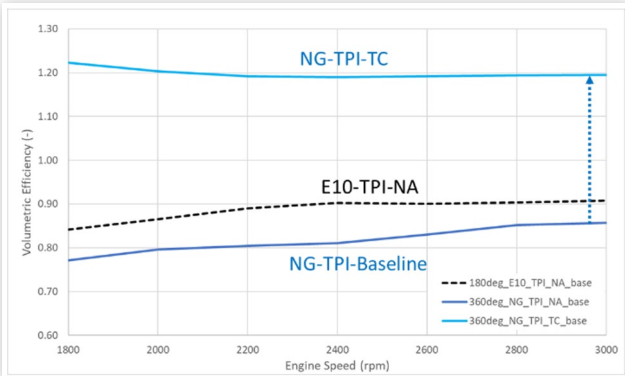
Simulation Results

Simulation studies were performed across the power curve for wide-open throttle (WOT) operation from 1800 rpm to 3000 rpm. Initial focus was on maximizing volumetric efficiency and assessing impacts to brake power and brake thermal efficiency, followed by emissions impacts.

Natural Gas Throttle Point Injection

A WOT power sweep was conducted from 1800 rpm to 3000rpm for both naturally aspirated and boosted operation

FIGURE 5 Volumetric efficiency of NG TPI naturally aspirated versus turbocharged versus E10 gasoline TPI naturally aspirated



and the results for volumetric efficiency are shown in Figure 5. With NA NG TPI the volumetric efficiency result had decreased to 0.85 at 3000rpm compared to 0.90 for the E10 gasoline point. Volumetric efficiency is a key factor for engine performance, and it is a good metric for quantifying the trends and the impact of changing fueling location and fuel type (liquid versus gas) among other system changes. The brake power is comprised of indicated power with associated losses (pumping, friction, heat transfer) associated with it, but ignores fuel pump work. Brake power shown in Figure 6 had decreased from 24.1 kW to 21.2kW for NG TPI NA operation due to the NG gaseous fuel injection in the intake manifold which decreased the available fresh air charge for induction. Although the NG TPI NA showed an improvement in brake thermal efficiency of 10% at 2000rpm (shown in Figure 7), at 3000 rpm the results showed a decrease of 5% in BTE versus the E10 gasoline point. However, upon addition of the optimized turbocharger the NG TPI TC operation improves the volumetric efficiency to 1.2 at 3000rpm compared to the 0.85 for NG TPI NA. This results in a brake power of 30.1 kW and 4% improvement in brake thermal efficiency at 3000rpm and 18% improvement in BTE at 2000rpm. In summary, Natural Gas TPI injection displaces fresh air which results in lower volumetric efficiency and power, but BTE improve for lower engine speeds due to improved LHV of NG at 49.6 MJ/kg vs E10 Gasoline 41.93 MJ/kg.

FIGURE 6 Brake Power for NG TPI NA and NG TPI TC.

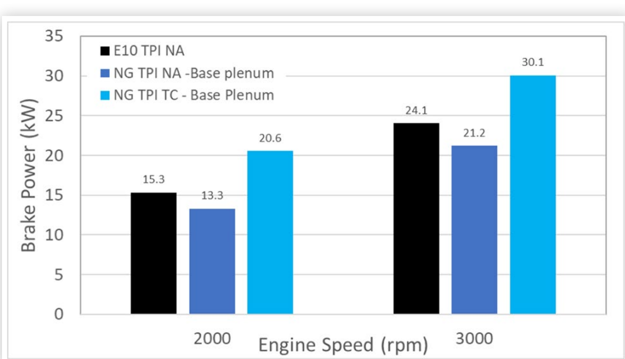
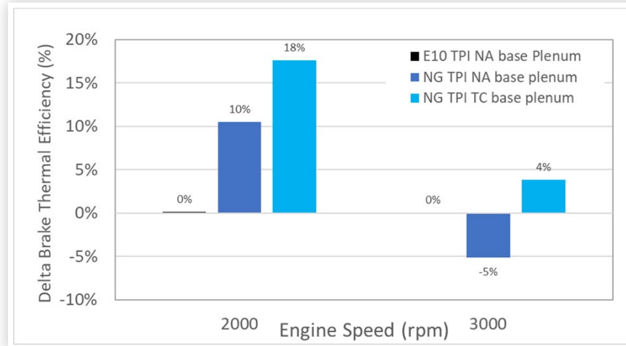


FIGURE 7 Delta brake thermal for NG TPI NA and NG TPI TC.



Plenum Volume Study – NG TPI TC

The objective of the plenum volume study is to improve the balance of the engine between cylinders 1 and 2 and to improve the volumetric efficiency of the engine. Appendix A contains the 3D geometries for the base plenum-side inlet as well as the 1L, 2L, and 3L central inlet plenums. Those plenums have resolved our air partition issue and consequently the cylinder pressure differences between both cylinders thanks to the central inlet.

The results for volumetric efficiency of the plenum volume study for NG TPI TC are presented in figure 8. The results showed at 3000rpm the 1L central inlet plenum resulted in a volumetric efficiency of 1.22 as compared to the base plenum side inlet 1.194. The 2L had the largest improved in volumetric efficiency under boosted operation with a 1.34 result at 3000rpm compared to the 3L plenum which had a result of 1.31. Figure 9 shows the results for brake power for the volumetric plenum study. The 2L plenum had a brake power result of 33.3kW for NG TPI TC operation compared to the NG TPI NA operation with base plenum which had 21.2kW at 3000 rpm. Additionally, the 3L plenum also showed promise with 32.7kW brake power at 3000 rpm.

FIGURE 8 Intake plenum volume study for central inlet plenum volumes 1L, 2L and 3L compared to the base plenum 1.3L.

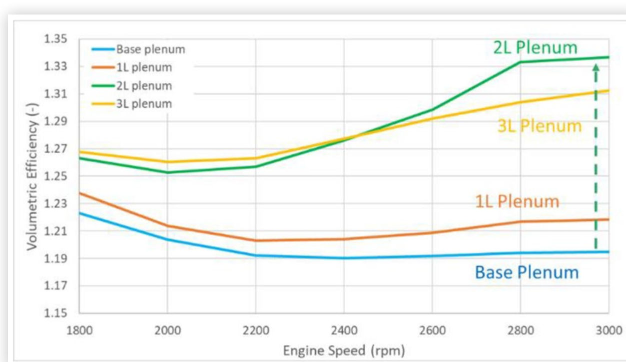
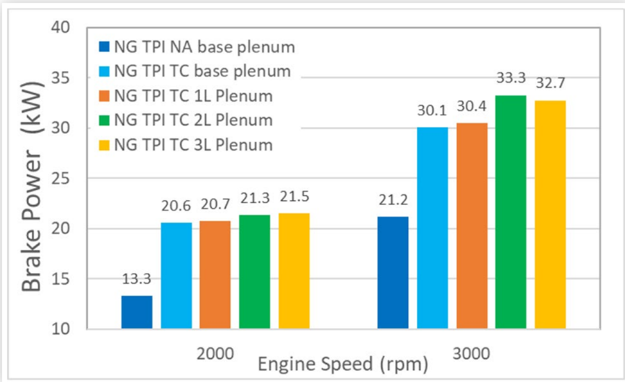


FIGURE 9 Brake Power results for plenum volume study



Natural Gas Direct Injection

The 1D GTPower model was modified to be able to conduct NG DI injection by adding a DI injector object to the combustion model and removing the TPI port fuel injection model we had previously. The NG DI injector flowrate was set to 10 g/sec in the DI model which was based on a 50bar injection pressure. The updates to the combustion model are shown in Figure 10. Additionally, the combustion model was modified to include the SI turbulent flame combustion model called 'SITurb'. This model provides a two-zone combustion, entrainment and burn-up model. The template predicts the in-cylinder burn-rate based on the initial conditions, emissions and knocking occurrence for spark-ignited engines. A flame geometry object has been set with a spark plug location based on our current setup. The X location is equal to the length of our long reach spark plug and the Z location is situated in between the two pistons at TDC. The Y location is set to 0. Regarding the spark size, it has been set to 1mm and the spark timing angle has been optimized by GTPower. The methane flame speed model has been used for the NG simulations, as the NG model is composed of 88% of methane, as well as the hydrogen flame speed model has been used for H_2 simulations. Regarding emissions models, an extended Zeldovitch mechanism was using for NOx modeling. The in-cylinder kinetic CO model and the in-cylinder HC model are using the GTPower mechanism. The HC model is using the piston-liner crevice volume and the piston-liner clearance. For those parameters, the assumption is that the values are coming from our current NG running engine. The GTPower HC model uses a simple kinetic model where the air-fuel mixture is pushed and trapped into the crevice volume set and re-enters

FIGURE 10 DI Injection model and combustion model modifications

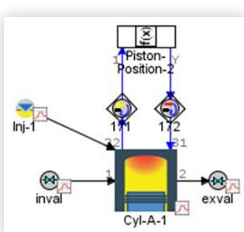
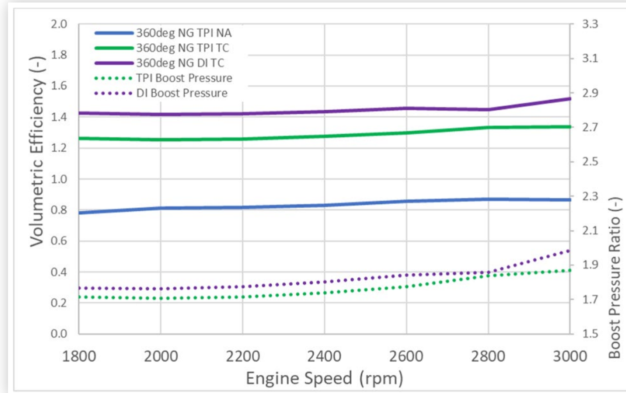


FIGURE 11 Volumetric efficiency and boost pressure ratio for boosted DI and TPI of NG versus TPI NA



into the main cylinder volume either before the flame is quenched by using the combustion model or after the flame is quenched according to the kinetic model specified. For a better emissions accuracy, a detailed air model has been developed with 78.08% of N_2, 20.95% of O_2, 0.93% of Ar and 0.042% of CO_2.The DI model will be used for either NG or H_2 DI fuel injection is shown in Figure 10.

The results of the simulation study investigated the boosted DI-NG resulted in the following in Figure 10. The volumetric efficiency of the boosted DI-NG was 1.5 meanwhile the results of the boosted TPI-NG were 1.35 at 3000rpm. Direct injection allows to have more fresh air inside the cylinder as it does not disrupt the intake air flow. The brake power improvement for the boosted DI-NG case with 37.5kW versus the PFI-NG case which had 33.3kW and the base NA case of 21.2 kW as shown in Figure 12. This resulted in a Brake thermal efficiency delta of 9.2% for the boosted DI-NG case and a delta of BTE of 8.3% for the boosted TPI-NG case as shown in Figure 13.

Hydrogen Gas Direct Injection

Hydrogen gas direct injection was applied to the DI gas model by switching the fuel model to 100% hydrogen fuel vapor using the same flowrate as previously used for NG DI namely 10 g/

FIGURE 12 Brake Power results for Boosted TPI and DI of NG versus base (TPI NA using NG fuel

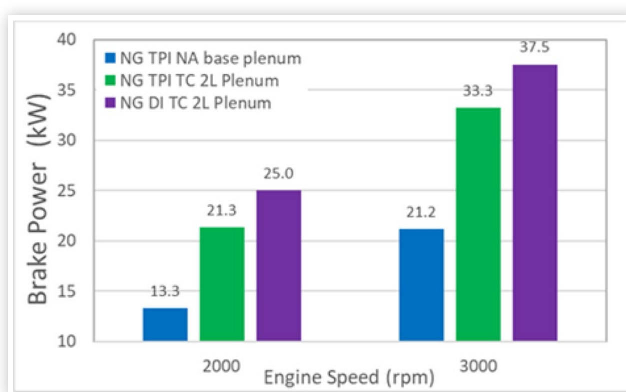
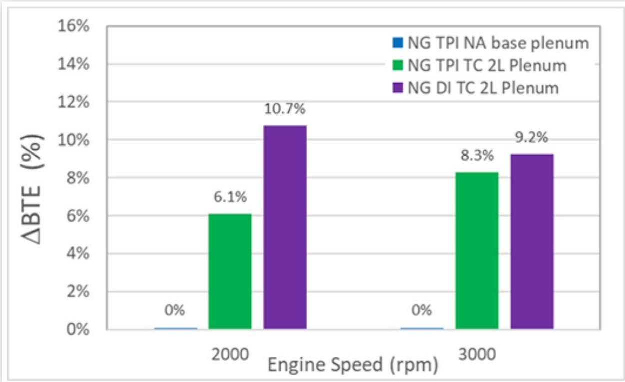


FIGURE 13 Percent Change in brake thermal efficiency for Boosted TPI and DI of NG versus base (TPI NA using NG fuel).



sec. Figure 14 shows the results for volumetric efficiency for Lambda 1.0 operation of boosted H_2 DI compared to other cases. The results showed boosted H_2 DI attained a volumetric efficiency of 2.0 at 3000 rpm, which is 50% higher than for the boosted NG DI case 1.5 and more than double that of the NG TPI naturally aspirated case. At Lambda 1.0, hydrogen gives more ethalpy to the turbocharger, which is the same one used on the natural gas engine. However, as hydrogen gives more energy thanks to its higher heating value, the turbocharger is able to give more power. As we can see below, the pressure ratio is higher than the natural gas engine which is explaining the rise of volumetric efficiency.

The results for brake power were shown in Figure 15 and show that the boosted H_2-DI case attains a power of 68.6kW at 3000rpm which is almost double the power of the boosted NG DI case (37.5kW) and more the three times the power output of the NA NG TPI case at 21.2 kW. This result shows the potential for a boosted H_2-DI OP4S to attain a specific power density of 68.6kW/L or 91.9hP/L at 3000rpm which is quite good considering the engines small size and low engine speed.

Figure 16 shows the percent difference in brake thermal efficiency and the boosted H_2-DI case has the potential to attain 28% improvement in BTE versus the baseline NG TPI NA conditions, compared to the boosted NG DI which has

FIGURE 14 Volumetric efficiency of Boosted H_2 DI versus Boosted NG DI and TPI and base condition.

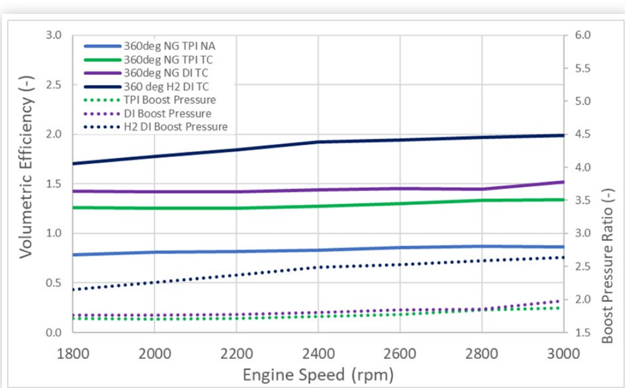


FIGURE 15 Brake power for boosted H_2 DI versus other fuels and injection methods. (Lambda 1.0)

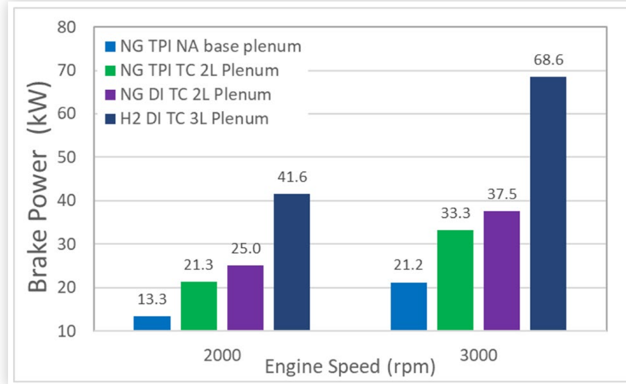
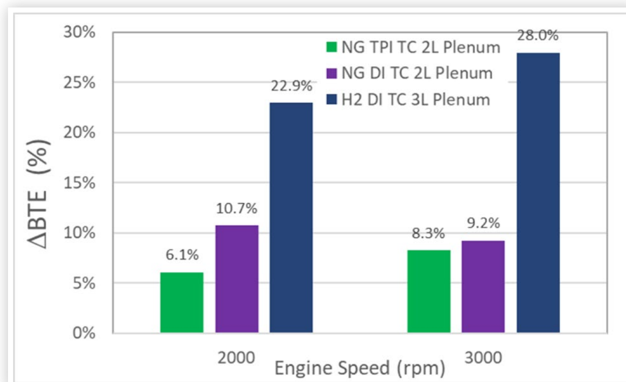


FIGURE 16 Percent Change in Brake thermal efficiency for boosted H_2 DI versus other fuels in injection methods



the potential to achieve 9.2% percent improvement versus the base condition. Hydrogen gives better engine performance at lambda 1 thanks to its high lower heating value (120MJ/kg) as the fuel flow rate is set to be the same than NG engine.

Lambda Sweep - H_2-DI

Lean burn boosted H_2-DI injection strategy has promises to achieve very high BTE, however the impact on brake power and emissions for the 1L OP4S engine is not well understood. A lambda sweep was conducted by varying the injected fuel quantity for both 2000 rpm and 3000 rpm. The boosted H_2-DI combustion mode has a combined BS(NOx + HC) of 2.5 g/kWhr at Lambda 1.0 operation. This is below the EPA regulation for class 2 small engines (8g/kWhr). Figure 17 shows the impact on combined BS(NOx + HC) emissions. For 2000rpm the combined emissions BS(NOx+HC)) increase from 2 g/kWhr to 32 g/kWhr at Lambda 1.5, and then decreases to 3g/kWhr at Lambda 2.0. For 3000rpm the combined emissions increase form 2g/kWhr at Lambda 1.0 to 10g/kWhr at Lambda 1.5 followed by a decreased to 3 g/kWhr at Lambda 2.0.

In Figure 18, the peak brake thermal efficiency at 2000rpm lambda 1.0 starts at 34% and increases to 38% at Lambda 1.75 and then decreases slightly to 37.5% at Lambda 2.0, however

FIGURE 17 Boosted H_2-DI, impact of lambda sweep on emissions

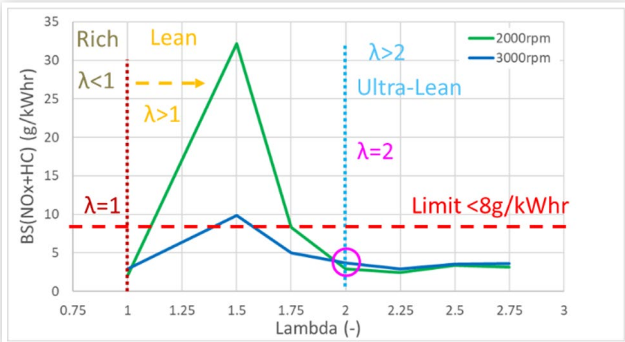
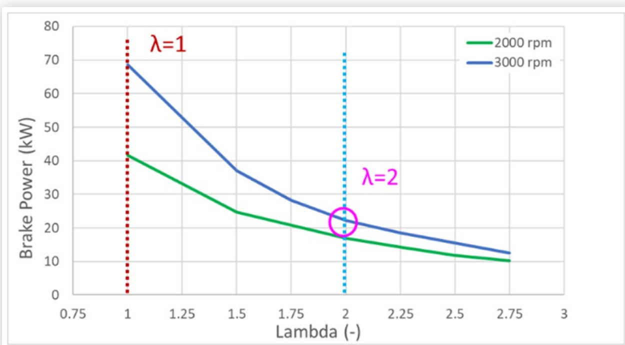


FIGURE 18 Boosted H_2-DI, impact of lambda sweep on Brake Power



continues to decrease to a BTE of 36% for Lambda 2.75. Similar trends were observed for 3000rpm lambda sweep with peak BTE reaching 35% at Lambda 1.75 and dropping to a BTE of 34.2% at Lambda 2.0. The impact of on Brake power was shown for both 2000 rpm and 3000rpm in Figure 19. The Rated Brake Power decreased from 68kW at Lambda 1.0 to 37kW at lambda 1.75 and 21.5kW at Lambda of 2.0 with power continuing to decrease as Lambda was increased to Lambda 2.75. The optimum point in was determined to be Lambda 2.0 with a BTE potential of up to 37.2% at 2000rpm or 34.2% at 3000rpm and Brake Power of 21.5kW at 3000rpm, meanwhile meeting the BS(NO+HC) emission of 3 g/kWhr. NOx emissions tend to be higher lean of stoichiometric ratio at Lambda 1.5 and decrease significantly as lambda is increased o beyond Lambda 2.0.

Compression Ratio Sweep – H_2-DI

Increasing compression ratio is a well understood technique to improve an ICEs overall brake thermal efficiency. With increased compression ratio the motoring pressure and temperature in-cylinder increases, which exposes the air and fuel mixture to higher pressure and temperature. This has the effect of shortening ignition delay and burn duration because the rate of reactions of the combustion process are increased. With stoichiometric operation this can lead to advanced timing and increased tendency for auto-ignition. By increasing the boost pressure the in-cylinder pressure increases but also at the same time we are diluting the in-cylinder air fuel mixture which has a tempering effect on the auto-ignition and delays the combustion process. Thus very lean burn with lambda >2 can enable the use increased compression ratio to reclaim the combustion phasing while avoiding excessive auto-ignition, especially when using fuels such as NG and H2 which have high octane numbers.

In order to achieve the target of 40% BTE a compression ratio sweep was conducted varying the CR from 8.5 to 20:1 to determine what is the minimum CR needed to achieve the target. Figure 19 capture the impact to combined BS(NOx+HC) emissions and the results showed that with CR increase from 8.5:1 to CR 12:1 the combined emissions increased from 3.75 to 4.5 g/kWhr for 3000rpm and as CR was increased to 20:1 the combined emissions decreased to 3.4 g/kWhr. Therefore, CR had minimal effect of increasing the combined BS(NOx + HC) emissions. With hydrogen combustion the unburned hydrocarbon emissions are non-carbon based hydrogen emissions which are minimal, so therefore the combined emission result is predominantly comprised of NOx emissions.

Figure 20 shows the impact of CR on Brake power with similar trends observed for both 2000 rpm and 3000 rpm. As CR was increased from CR 8.5:1 to CR 20:1 the brake power decreased from 22.3kW to 22.0kW. Figure 21 shows the impact of CR on Brake thermal efficiency. As the CR is increased from 8.5:1 to 16:1 we observe a noticeable increase in BTE from 38 to 40% for 2000rpm and then at 20:1 a decreased down to 39%. Whereas for 3000rpm we observe a noticeable increase

FIGURE 19 Boosted H2-DI, Impact of lambda sweep on BTE

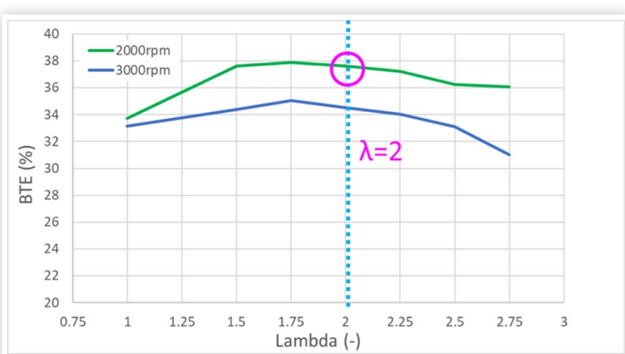


FIGURE 20 Effect of compression ratio on Combined BS(NOx+HC) Emissions.

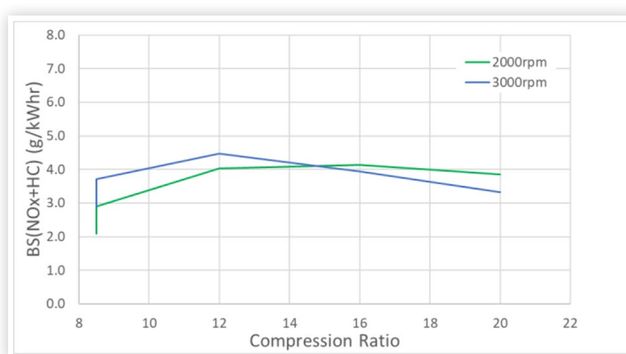


FIGURE 21 Effect of compression ratio of Brake Power Boosted H_2 DI

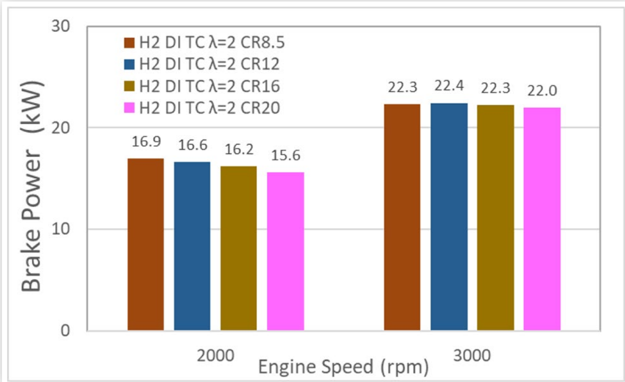
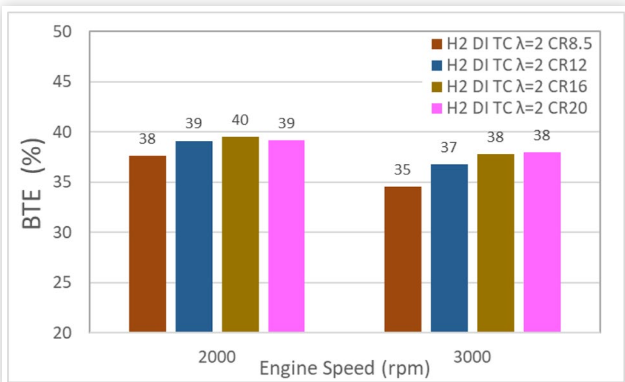


FIGURE 22 Effect of compression ratio on Brake Thermal Efficiency for Boost H_2 DI



in BTE from 35 to 38% as CR is increased from CR 8.5:1 to CR 16:1. CR 16:1 provides the maximum BTE of 40% at 2000rpm and provides the peak BTE of 38% at 3000rpm, with minimal impact to emissions and minimal impact to brake power of 22.3 kW. Increasing CR further to CR20:1 did not provide additional benefits.

Future Work

The results obtained were encouraging and demonstrate the potential of boosted H_2 Direct injection for high efficiency in an OP4S engine. The thermodynamic benefits of an OP4S versus conventional 4S ICE will be explored in a future publication. Additional, 3D engine CFD work such be carried out to optimize the piston bowl geometry and injector nozzle spray angle and nozzle hole size matching to achieve high load whilst improving low load performance. One-dimensional DI injector simulation should be carried out to determine the dynamic flowrate profile of a H_2 DI injector nozzle. The authors have conducted work with high pressure direct injection of NG and designed a 4H nozzle and characterized the gaseous injection process along 3D CFD simulation would be helpful to be conduct using H_2 fuel to develop a gaseous H_2 DI injector nozzle. Future simulation work should consider sweeping EGR instead of AFR as a NOx control method. Additionally, the widespread use of H_2 fuel would be a 90% H2 composition rather than 100% H_2. Thus simulation of 90% H_2 with balance other gases CO, CH4, etc and the impact of on emissions should be considered.

Summary/Conclusions

Based on the results discussed in this paper the following conclusions can be drawn:

1. Lean burn boosted H2-DI combustion mode is able to achieve all target objectives of the study.
2. It was possible to achieve a combined brake specific NOx & HC target of less than 8g/kWhr at both peak torque and rated power operation with lean burn boosted H2 DI combustion (attained less than 4g/kWhr).
3. Brake thermal efficiency target of 40% was able to be achieved at 2200 rpm by combination of high CR (CR16), Lean burn (lambda 2.0) and turbocharging with volumetric efficiency up to 2.0
4. The power target of greater than 20kW was easily achieved at 3000rpm with 22.2kW with a BTE of 38% while meeting the combined emissions target.
5. Future CFD nozzle studies should be performed on H2 DI to determine the highest BTE efficiency potential flowrate.
6. A 1L OP4S engine should be modified to add the turbocharger as well as a H2-DI fuel Injection Equipment (FIE) system, to be able to conduct testing to verify the potential of 40% BTE at 2000rpm.

References

1. White, C.M., Steeper, R.R., and Lutz, A.E., "The Hydrogen-Fueled Internal Combustion Engine: a Technical Review," *International Journal of Hydrogen Energy* 31 (2006): 1292-1305.
2. Rosati, M.F. and Aleiferis, P.G., "Hydrogen SI and HCCI Combustion in a Direct-Injection Optical Engine," *SAE Int. J. Engines* 2, no. 1 (2009, 2009): 1710-1736, doi:https://doi.org/10.4271/2009-01-1921.
3. Wimmer, A., Wallner, T., Ringler, J., and Gerbig, F., "H2-Direct Injection - A Highly Promising Combustion Concept," SAE Technical Paper 2005-01-0108, 2005, https://doi.org/10.4271/2005-01-0108.
4. Lee, S.J., Yi, H.S., and Kim, E.S., "Combustion Characteristics of Intake Port Injection Type Hydrogen Fueled Engine," *International Journal of Hydrogen Energy* 20 (1995): 317-322.
5. Kölsch, R.K. and Clark, S.J., "A Comparison of Hydrogen and Propane Fueling of an IC Engine," SAE Technical Paper 790677, 1979, https://doi.org/10.4271/790677.

6. Zoldak, P.S., "Partially Stratified Combustion of Natural Gas For Spark Ignition Engines," Open Access Dissertation, Michigan Technological University, 2022.
7. https://doi.org/10.37099/mtu.dc.etdr/1449
8. Zoldak, P., Joseph, J., Johnson, J., and Naber, J., "Characterization of Partially Stratified Direct Injection of Natural Gas for Spark-Ignited Engines," in *SAE WCX15*, 2015.
9. Zoldak, P. and Naber, J., "Spark Ignited Direct Injection Natural Gas Combustion in a Heavy Duty Single Cylinder Test Engine - AFR and EGR Dilution Effects."
10. Zoldak, P. and Naber, J., "Spark Ignited Direct Injection Natural Gas Combustion in a Heavy Duty Single Cylinder Test Engine - Start of Injection and Spark Timing Effects," in *SAE WCX15*, 2015.
11. Zoldak, P. and Naber, J., "Spark Ignited Direct Injection Natural Gas Combustion in a Heavy Duty Single Cylinder Test Engine - Nozzle Included Angle Effects," in *SAE WCX17*, 2017.
12. Zoldak, P., Douvry-Rabjeau, J., and Warren, J., "Development of a Gaseous Fueled Boosted Opposed-Piston Four Stroke (OP4S) Engine," in *Gamma Technologies Technical Conference (GTCC)*, Plymouth, MI, USA, 2022.
13. Herold, R., Wahl, M., Regner, G., Lemke, J. et al., "Thermodynamic Benefits of Opposed-Piston Two-Stroke Engines," SAE Technical Paper 2011-01-2216, 2011, https://doi.org/10.4271/2011-01-2216.
14. Wallner, T., Matthias, N.S., and Scarcelli, R., "Influence of Injection Strategy in a High-Efficiency Hydrogen Direct Injection Engine," *SAE Int. J. Fuels Lubr.* 5, no. 1 (2012): 289-300, doi:https://doi.org/10.4271/2011-01-2001.
15. Matthias, N.S., Wallner, T., and Scarcelli, R., "A Hydrogen Direct Injection Engine Concept that Exceeds US DOE Light-Duty Efficiency Targets," *SAE Int. J. Engines* 5, no. 3 (2012): 838-849, doi:https://doi.org/10.4271/2012-01-0653.
16. Roy, M.K., Kawahara, N., Tomita, E., and Fujitani, T., "High-Pressure Hydrogen Jet and Combustion Characteristics in a Direct-Injection Hydrogen Engine," *SAE Int. J. Fuels Lubr.* 5, no. 3 (2012): 1414-1425, doi:https://doi.org/10.4271/2011-01-2003.
17. Rouleau, L., Duffour, F., Walter, B., Kumar, R. et al., "Experimental and Numerical Investigation on Hydrogen Internal Combustion Engine," SAE Technical Paper 2021-24-0060, 2021, https://doi.org/10.4271/2021-24-0060.
18. Bekdemir, C., Doosje, E., and Seykens, X., "H2-ICE Technology Options of the Present and the Near Future," SAE Technical Paper 2022-01-0472, 2022, https://doi.org/10.4271/2022-01-0472.
19. USEPA 2019, "Nonroad Small SparkIgnited (SI)Engines and Evaporative Components," US EPA Compliance Workshop Washtenaw Community College, 2019.

Contact Information

Philip Zoldak, PhD, VP Engineering
Enginuity Power Systems.
phil@enginuitypowersystems.com

Acknowledgments

The authors would like to acknowledge the inventor of Enginuity's OP4S engine Mr. James Warren. The authors would also like to recognize the support of the staff at Gamma Technologies specifically Ron Rogers, Tom Wanat, Param Singh and Kevin Roggendorf for their gracious support of the model build and assistance with sub-model and case setup for GTPower and GEM3D.

Definitions/Abbreviations

ATDC - after top dead center
BMEP - brake mean effective pressure
CA50 - 50%mass fraction burned crank angle
CAD - crank angle degrees
CI - compression ignition
CO - carbon moNOxide emissions
CO2 - carbon dioxide
CR - compression ratio
degC - degrees Celsius
DI - direct injection
FIE - fuel injection equipment
FSN - filter smoke number
FUP - fuel pressure at rail
FUT - fuel temperature inside injector
GCI - gasoline compression ignition
g/kWh - grams per kilowatt hour
HC - hydrocarbon emissions
H_2 - Hydrogen gaseous fuel
HRR - heat release rates
IMEP - indicated mean effective pressure
IMT - intake manifold temperature
ISFC - indicated specific fuel consumption
IVC - intake valve closing
IVO - intake valve opening
kPa - kilopascal
LFE - laminar flow element
LTC - low temperature combustion
NEV - new energy vehicles
NOx - oxides of nitrogen emissions
OEM - original equipment manufacturer
ON - octane number
PFI - port fuel injection
PM - particulate matter emissions
PON - pump octane number
PPC - partially premixed combustion
PRR - pressure rise rates
RON - research octane number

rpm - revolutions per minute
SOC - start of combustion
SOI - start of injection
TDC - top dead center
TPI - Throttle port injection
VVA - variable valve actuation

Appendix 1: Intake Plenum Models

FIGURE 23 Base Plenum Model – Side Inlet 1.3L

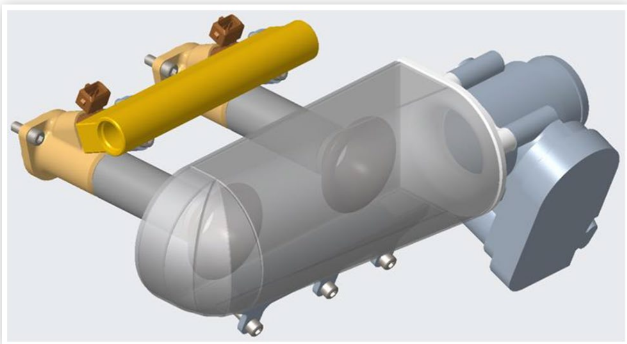


FIGURE 24 GTpower model of base plenum using GEM3D

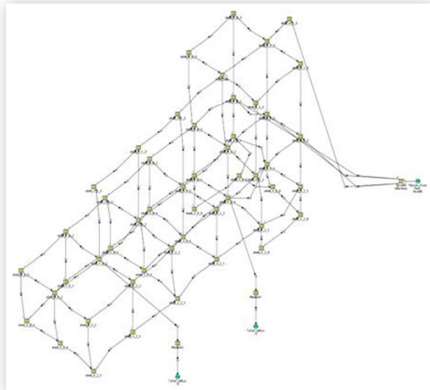
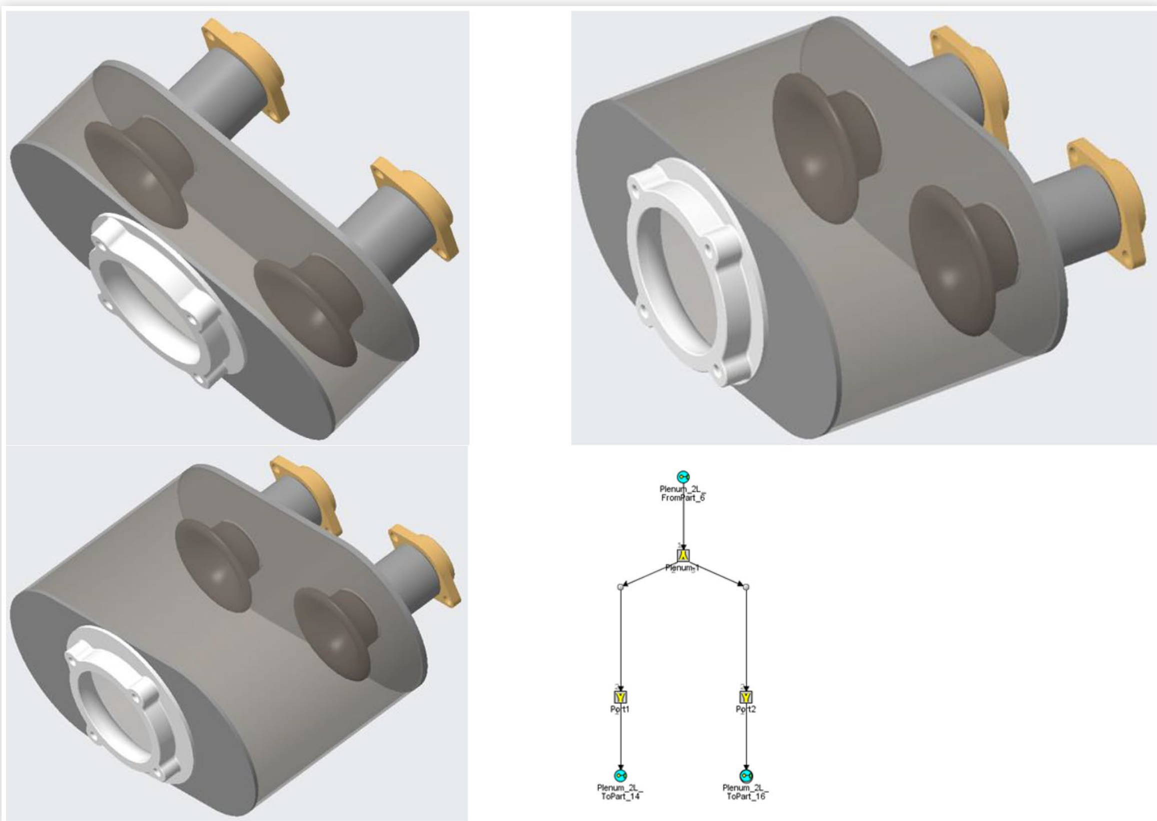


FIGURE 25 Central inlet intake plenums 1L (top left), 2L (bottom left), 3L (top left), GEM3D model (bottom right)



ISSN 0148-7191