HIGH PERFORMANCE COMPUTING AND ANALYSIS-LED DEVELOPMENT OF HIGH EFFICIENCY DILUTE OPPOSED PISTON GASOLINE ENGINE

Siddhartha Banerjee[1]
Pinnacle Engines Inc.
San Carlos, CA 94070
e-mail: sidbannet@gmail.com

Clayton Naber
Pinnacle Engines Inc.
San Carlos, CA 94070
e-mail: clayton@pinnacle-engines.com

Michael Willcox
Pinnacle Engines Inc.
San Carlos, CA 94070
e-mail: tony@pinnacle-engines.com

Charles E. A. Finney
Oak Ridge National Laboratory
Oak Ridge, TN 37830
e-mail: finneyc@ornl.gov

Dean K. Edwards
Oak Ridge National Laboratory
Oak Ridge, TN 37830
e-mail: edwardskd@ornl.gov

ABSTRACT

Pinnacle is developing multi-cylinder 1.2 L gasoline engine for automotive applications using high performance computing (HPC) and analysis methods. Pinnacle and Oak Ridge National Laboratory executed large-scale multi-dimensional combustion analyses at the Oak Ridge Leadership Computing Facility to thoroughly explore the design space. These HPC-led investigations show high fuel efficiency (~46% gross indicated efficiency) may be achieved by operating with extremely high charge dilution levels of exhaust gas recirculation (EGR) at a light load key drive cycle condition (2000 RPM, 3 bar BMEP), while simultaneously attaining high levels of fuel conversion

[1] Siddhartha Banerjee is the corresponding author of this paper. Work presented in this paper was mostly done during corresponding author's tenure at Pinnacle Engines Inc. at San Carlos, CA

efficiency and low NOx emissions. In this extremely dilute environment, the flame propagation event is supported by turbulence and bulk in-cylinder charge motion brought about by modulation of inlet port flow. This arrangement produces a load and speed adjustable amalgamation of swirl and counter-rotating tumble which provides the turbulence required to support stable low-temperature combustion (LTC). At higher load conditions, the engine may operate at more traditional combustion modes to generate competitive power.

In this paper, the numerical results from these HPC simulations are presented. Further HPC simulations and test validations are underway and will be reported in future publications.

INTRODUCTION

Increasingly stringent fuel-economy and emissions standards are compelling significant changes in gasoline-fueled internal combustion engines for automotive applications. Some of the strategies to achieve compliance with regulations are to run the engine with excess air or with recirculated exhaust gases as part of the combustion charge. The Pinnacle opposed reciprocating piston with a 4-stroke combustion cycle is a novel engine architecture which allows stable low-temperature combustion in a highly dilute combusting environment. Compared to conventional engines, Pinnacle's unique sleeve valve engine construction generates much higher levels of in-cylinder turbulence due to the collapse of counter-rotating tumble during the flame propagation event. Additionally, by equipping one crankshaft with a crankshaft phaser, Pinnacle's engine can manipulate the geometric compression ratio with piston phasing in a continuous fashion. This allows the engine to operate at a much higher compression ratio at low- and mid-load operations and gradually lower compression ratios as load increases to avoid engine knock and lower peak cylinder pressures. The combination of higher turbulence and higher compression ratio at a wide range of engine operating conditions enables this engine to operate at very highly dilute conditions compared to conventional piston engines, resulting in higher fuel economy and lower emission levels.

OVERVIEW OF PINNACLE'S ENGINE TECHNOLOGY

Pinnacle's 2nd generation opposed piston gasoline scooter engine utilizes a lean-burn combustion system and has shown in vehicle drive cycle bag emissions tests that Euro 4 emissions can be easily met without NOx after treatment with acceptable combustion stability (<3% COV of IMEP) [1]. In the present work the authors are focused on developing a 4-stroke opposed piston gasoline engine with a goal to meet the latest emissions standards such as Tier 3 Bin 160 and Euro 6 at fuel efficiency comparable to light duty diesel engines and at much lower costs. The work is focused on developing a high efficiency low-emission combustion recipe for a 4-stroke, opposed piston gasoline engine for automotive applications (Figure 1).

Figure 1: CAD model of Pinnacle Engine's 1.2L opposed-piston gasoline engine

A highly dilute combustion recipe is chosen to achieve these goals. Spark ignited (SI) gasoline engines are approaching fuel-economy levels typically achieved by compression ignition (CI) diesel counterparts by employing high compression ratio(s) (CR) with cooled exhaust gas recirculation (EGR) and direct-injection (DI). A well-tuned combustion recipe that leverages higher CR to enhance LTC burn rates can deliver a high effective expansion ratio (EER) together with low heat losses. High EER is essential to extract piston work most effectively and reduce energy losses to the exhaust stream. However, in SI engines, CR is often limited by the occurrence of knock.

In order to alleviate knock in high CR SI engines, a spark retarding strategy is often employed, which in turn reduces EER and increases exhaust gas energy losses. In previous studies, elevated levels of cooled EGR have proven effective to reduce the propensity of knock occurrence [2] in addition to increasing the fuel octane level. Higher dilution levels reduce flame temperature, which in turn reduces NOx emission. However, increased dilution levels also decrease laminar flame speed. To mitigate this or rather extend the dilution tolerance of this engine, it is important to manage charge motion to generate adequate in-cylinder turbulence to enhance flame speeds. In-cylinder turbulence in a 4-stroke opposed piston engine is generated in part by in-cylinder shear planes that occur between cylinder halves during the induction stroke and enhanced as the pistons come together and cause the collapse of a counter-rotating dual tumble during the compression stroke (Figure 2). Higher in-cylinder turbulence with a highly dilute combustion environment creates a novel combustion design uniquely positioned to deliver high fuel economy benefit at low emissions.

Figure 2 shows the presence of counter-rotating dual tumble as visualized by streamlines of velocity in the tumble plane. Pinnacle’s opposed piston architecture incorporates sleeve valves for controlling intake and exhaust flow. The sleeve valves each open a port near the center of the engine. Flow entering through the intake valve flows into both the “primary side” and the “secondary side” representing two halves of the combustion chamber corresponding to the opposed pistons' stroke-path. Two large scale, counter-rotating tumble eddies are produced from the induction stroke. During the compression stroke, the piston compresses these tumble motions resulting in higher small-scale turbulence if compared to collapse of single tumble motion of traditional single-piston engine architecture. This mechanism is responsible for creating a highly turbulent environment that can be taken advantage of in highly dilute combustion system to maintain or enhance burn rates relative to a less-dilute low turbulence system.

In-cylinder flow developed from the intake charge has vortex formation perpendicular to the piston motion. Figure 3 shows the flow velocity vector of in-cylinder flow motion in the swirling plane, generated during the intake stroke. It may be noticed that counter-rotating vertical swirl is generated in this case. By regulating the flow from one of the inlet port legs it is possible to manipulate in-cylinder charge to generate net swirl.

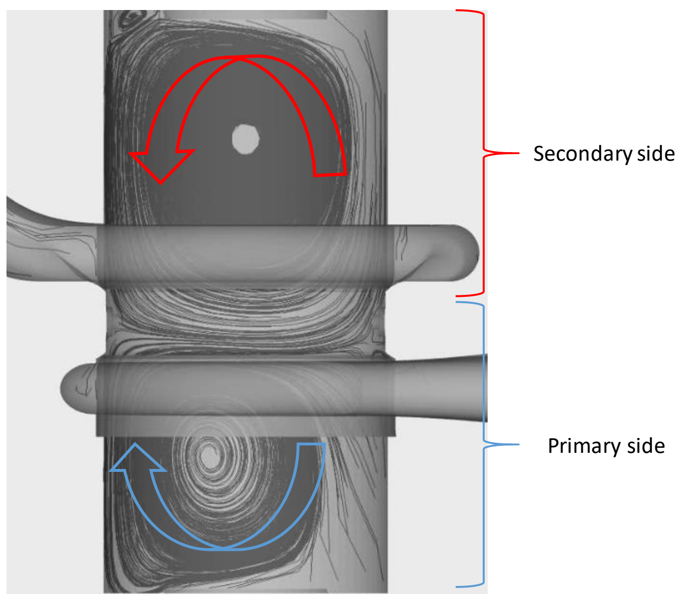

Figure 2: Presence of counter-rotating dual tumble in Pinnacle's 1.2L engine. Streamline of in-cylinder charge motion at IVC

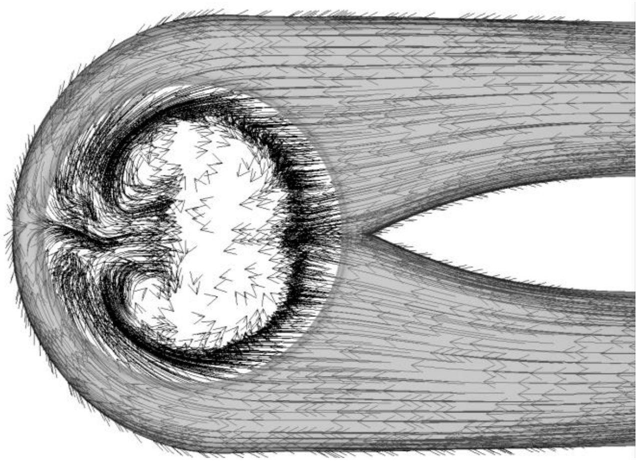

Figure 3: In-cylinder velocity vector from the charge intake in the plane perpendicular to the piston motion

Pinnacle's 1.2L engine is designed with Variable Valve Timing (VVT) as well as Variable Compression Ratio (VCR) system. The VVT is capable of continuously adjusting intake and exhaust valve timing independently, whereas VCR is capable of continuously adjusting the phasing of the secondary side piston relative to primary side piston, giving ability to adjust compression ratio. Because of the ability to independently control the intake and exhaust valve phasing, the internal residual charge (internal EGR) can be controlled with VVT. Combining VCR with VVT makes it possible to couple a load and speed dependent high geometric compression ratio (and therefore expansion ratio) with delayed exhaust valve open (EVO) event. The naturally aspirated performance goals for this engine in a non-hybrid vehicle motivated the selection of a late (Atkinson) inlet lift profile to reduce pumping work and effective compression ratio, but an early closing inlet profile (Miller) is also feasible to leverage depending on the performance requirements [3]. Figure 4 shows example intake and exhaust valve profiles and timing ranges.

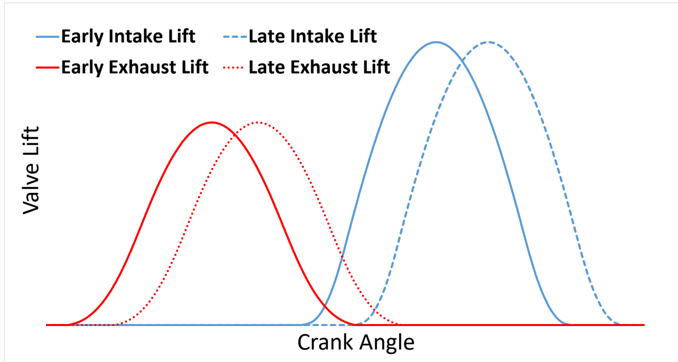

Figure 4: Valve lift profiles with the VVT system of Pinnacle's 1.2L Engine

The VCR system is used to increase CR at low-load operation with high dilution levels. At high-load operation, the engine operates at lower charge dilution levels and to avoid knock, compression ratio can be lowered by lagging the secondary piston's arrival to its highest position with respect to the primary piston. The specifications of the 1.2L engine are given in Table 1 below. In traditional single-piston engine architectures, VCR capability is often limited to a limited discrete number of piston adjustment mechanisms [4]. This crank phasing method in an opposed-piston architecture enables continuously adjustable CR by leveraging already proven cam phaser technologies.

Table 1: Engine specification of Pinnacle's 1.2L engine

Bore/Stroke (single piston) ratio [-]	0.937		**Fuel delivery**	Port Fuel Injection
Peak Surface to Volume Ratio [1/cm]	4.76		**In-cylinder flow system**	Dual tumble and adjustable Swirl (Swumble)
Compression Ratio System	VCR - Continuous Adjustment between 10:1 and 20:1 CRs		**Valve Timing**	VVT – Continuous Adjustment of Cam Phasing
Ignition system	Synchronously fired dual spark plugs		**Combustion system**	Spark Ignition with cooled EGR

Schematic of the 1.2L engine system is given in Figure 5. The primary throttle regulates the amount of fresh inlet charge mixed with cooled EGR into the intake manifold. A secondary throttle placed in one of the legs of each inlet port regulating the type of in-cylinder flow. When the secondary throttle is closed, the flow from one of the inlet port legs is shut off, generating a high swirl motion inside the cylinder enhancing dilution tolerance. On the other hand, when the secondary throttle is open, flow from both inlet port legs occurs resulting in no net in-cylinder swirl. The in-cylinder swirl is continuously modulated by the secondary throttle over different loads and speeds. Regardless of the secondary throttle control, the ensuing dual tumble in-cylinder charge motion persists (Figure 2). When the swirl is present along with the dual tumble, a new mode of in-cylinder charge motion is generated. This co-existing swirl and dual tumble in-cylinder charge will be referred to as "swumble" flow in this paper. There are two spark plugs located inside each of the cylinder of combustion chamber. The spark plugs are located on the side of the cylinder due to absence of a cylinder head in the opposed-piston engine architecture.

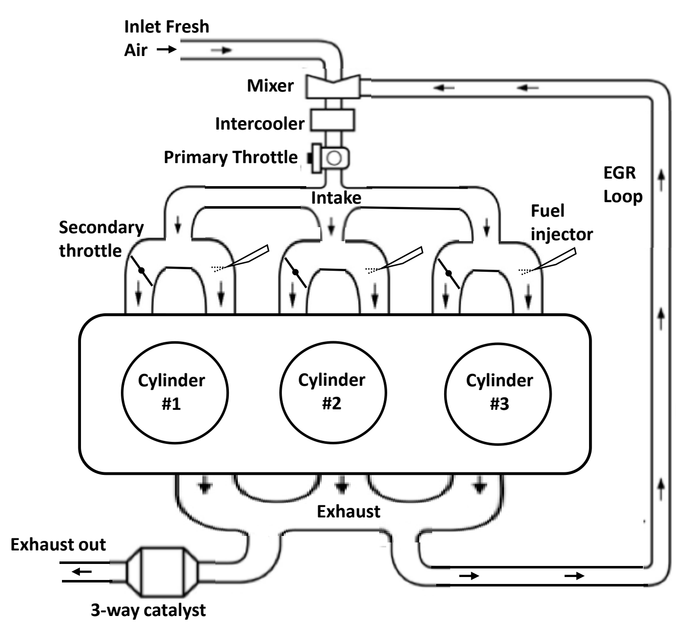

Figure 5: Schematic of Pinnacle's opposed piston, 4-stroke dilute gasoline engine

ANALYSIS-LED DEVELOPMENT OF COMBUSTION SYSTEM

In this work, multi-dimensional engine CFD simulations are performed using the commercially available CONVERGE^{TM} software package [5]. The CONVERGE^{TM} software can perform numerical solutions for a chemically reactive, two-phase flow problems with moving boundary to emulate piston and valve motions. CONVERGE^{TM} uses a structured, orthogonal, finite volume computational grid with automatic mesh refinement (AMR) and mesh-embedding methodologies to reduce numerical viscosity of the multi-dimensional flow solution. It uses a time-splitting numerical scheme for the flow solver with a 1st order Euler in time and a 2nd order accurate central difference scheme in space.

In this paper, the multi-dimensional numerical investigation is conducted using the RNG kε turbulence model [6] and detailed chemistry solver for combustion chemistry [7]. The law of wall model is used for near-wall physics [8] and 0D crevice model to represent gas dynamics between cylinder charge and piston crevice regions [9]. For ignition, an energy dump model is chosen with prescribed inductive coil energy dump profile based on prior experimental measurements. The NOx formation and reduction is captured using extended Zel'dovich reaction kinetics [10].

A best-practice method of an Analysis-Led Design and Development (ALDD) process has been developed for the 1.2L development program. The engine design is carried out in parallel with results from high-fidelity numerical models. In order to simulate thermal and fluid phenomena, a one-way coupled approach is adopted using a 1D/0D system level simulation (performed using the GT-POWER® simulation package) and the multi-dimensional numerical tool where spatial resolution is needed. This approach, portrayed in Figure 6, shows that boundary conditions are generated by the 1D/0D system models. The turbulent reactive combustion simulation with higher fidelity physical sub-models is performed to predict combustion, emissions, and combustion knock characteristics using CFD simulations. Results from CFD results are used to improve the accuracy of the system level models.

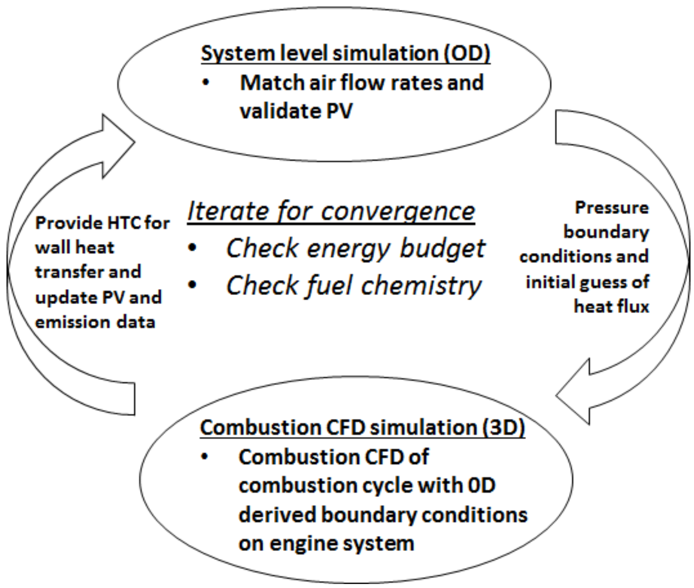

Figure 6: Schematic of Pinnacle's analysis-led process for combustion design

Using the ALDD process, massively parallel simulations of many designs and control degrees of freedom have been launched at the Oak Ridge Leadership Computing Facility (OLCF). In this work, meta-models (model of models) are constructed based on results from the large number of parallel CFD results [11]. The meta-models are constructed using statistical model construction principles of deriving response functions from large set of sample results.

For example, a large number of numerical design of experiments (DoE) cases are conducted with design and control degrees of freedom such as piston crown shapes, port design, EGR level, spark timing, swirl level, CR, valve timing, and engine speed and load. Based on the CFD results, a set of response models for individual parameters of interest (such as GIE, fsNOx, fsHC, Knock, combustion noise, COV of IMEP) are constructed relative to other parameters of interest (such as speed/load, 50% mass fraction burned (CA50), swirl, tumble, etc).

In this work quadratic functions are used to construct the response models. The collection of all such response functions are referred to here as a meta-model. In this work, meta-models are used to rapidly explore continuously a much wider range of design options and control authority in order to arrive at an optimal engine design within a set of constraints (such as emission, noise, and/or knock). This model-based prediction and optimization workflow is shown in Figure 7 below. The robustness of the optimal solution is also calculated using Monte Carlo simulation based on meta-models.

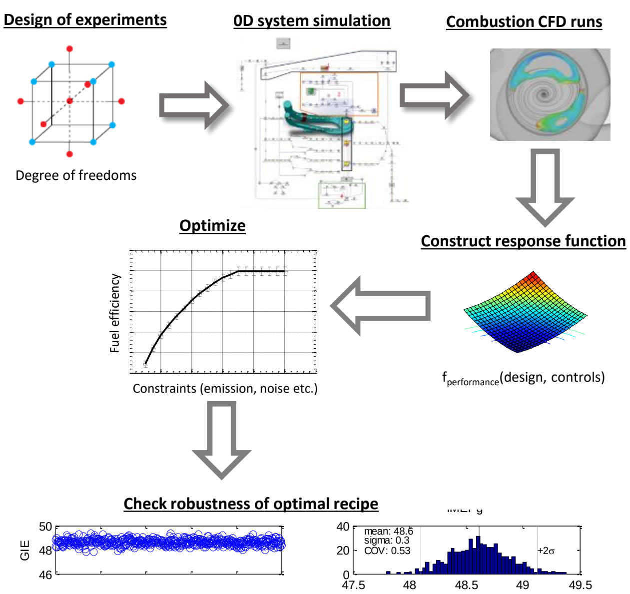

Figure 7: Design and control optimization using model based prediction

An example of the influence of two parameters, namely combustion duration and phasing, on heat release is shown in Figure 8 below. As the resultant combustion phasing and many other parameters are unknown at DoE generation, the spark timing is leveraged as a degree of freedom to modulate the resultant combustion phasing across bounds were it is expected to perform optimally. Subsequently, the meta-model is utilized to precisely identify the combination of conditions where a constrained optimum is found. The influence of EGR dilution level on both lowering the peak heat release rate and extending combustion phasing is also visible from these results.

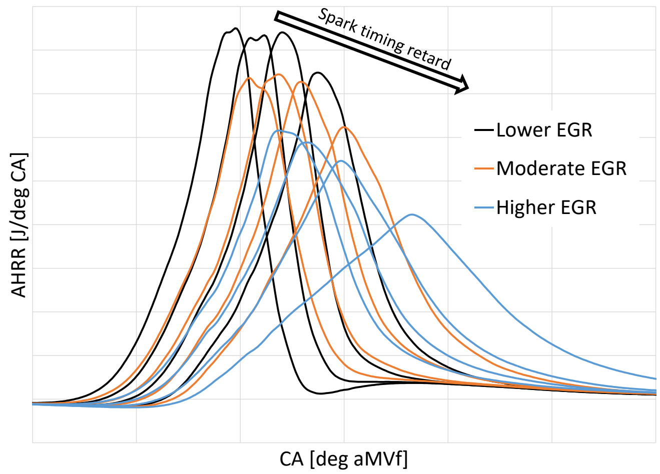

Figure 8: Apparent Heat Release Rate (AHRR) with spark timing and EGR level sweeps

RESULTS AND DISCUSSION

This section highlights the results from the numerical simulation of Pinnacle's 1.2L development engine. In this section, two port designs and two piston crown design will be discussed. The effect of charge motion on dilution tolerance, fuel efficiency, and emissions is also discussed. Finally, the capability of the VCR to increase fuel efficiency will be presented.

Intake port and charge motion

The design of the intake port is critical to the development of in-cylinder charge motion. In this study, two families of inlet ports were used, one symmetric in design and the other which used different design concepts of each flow leg (Figure 9). Each port housed a secondary throttle in one of the legs and port fuel injector on the other (Figure 5).

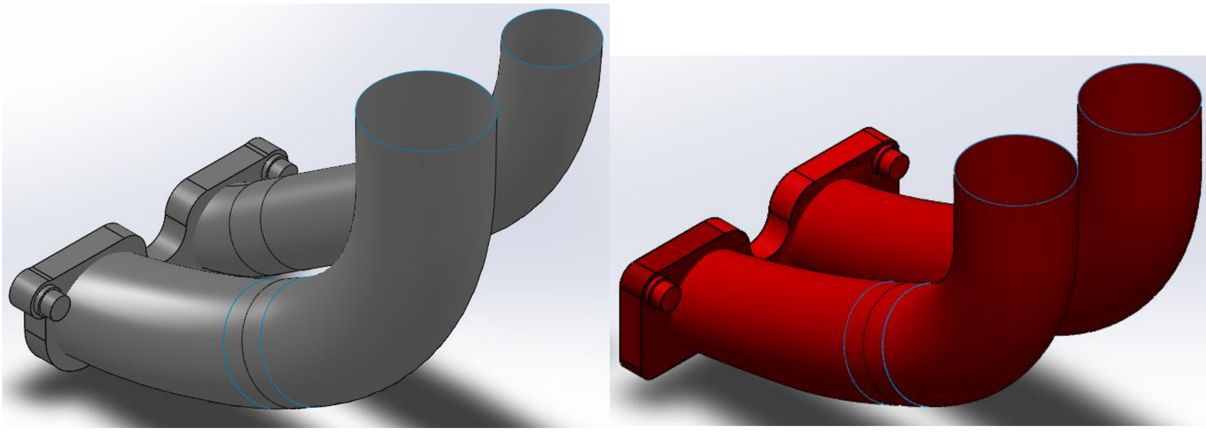

Figure 9: Inlet port designs for 1.2L engine. Asymmetric inlet port legs (left), Symmetric inlet port legs (right)

The asymmetric inlet port design uses dissimilar hydraulic flow areas between its port legs, while the symmetric port design maintains equivalent areas. Figure 10 shows a top view of the in-cylinder flow at spark timing for each port with fully open and fully closed throttle at a plane coinciding with the location of the spark plugs. Results show the asymmetric port design is able to generate some level of in-cylinder swirl even without the restriction of the secondary throttle due to preferred flow between the port legs. With a restriction added by the secondary throttle closing, the swirl generated by the asymmetric inlet port is higher than the symmetric port design. Figure 11 shows tradeoff between in-cylinder turbulence (quantified by modeled TKE at minimum cylinder volume) and swirl. The asymmetric port can provide higher turbulence than the symmetric port at low swirl levels and deliver a higher swirl magnitude when the secondary throttle is closed. When the secondary throttle is fully open and inlet port legs deliver similar flow (Figure 3) and pure dual tumble flow is generated inside the combustion chamber. In both flow configurations, the bulk in-cylinder turbulence is generated by the collapse of the dual tumble (Figure 2). When the swirl is present, it acts to supplement the in-cylinder turbulence to enhance flame kernel development. In asymmetric inlet port leg design, the dynamics of the dual tumble collapse is undisturbed by the presence of net swirl. Because of its higher swirl capacity, the asymmetric inlet port provides an in-cylinder flow mode that transports the flame propagation tangentially around the combustion chamber. It also provides the dual-tumble collapse turbulent energy necessary to enhance flame speed in tumble mode. Hence, this port provides multiple modes with which to extend dilution tolerance.

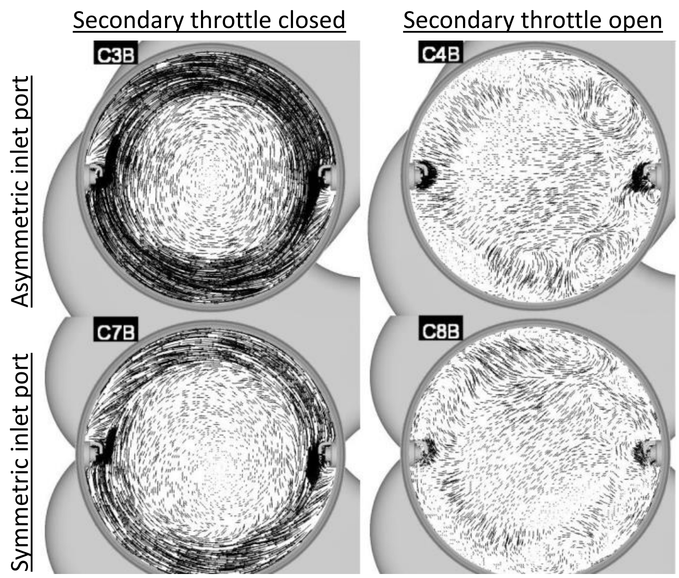

Figure 10: In-cylinder flow field. Comparison between inlet port design and secondary throttle operation

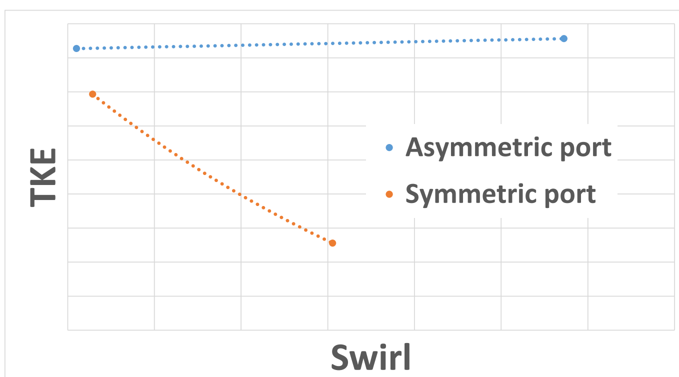

Figure 11: Tradeoff between in-cylinder turbulence and swirl - a comparison between two inlet port designs

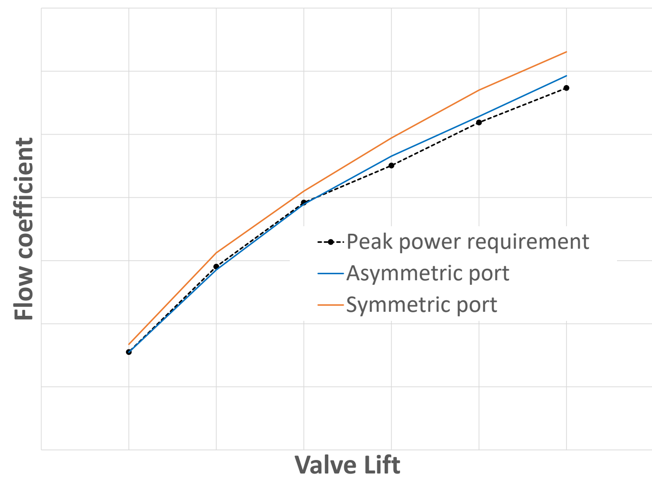

Figure 12: Intake flow coefficient against valve lift

It is also important to confirm the valve flow coefficient curve is sufficient to achieve the target full throttle power and torque requirements. Figure 12 shows the normalized flow coefficient against valve lift position for the inlet port designs with secondary throttle wide open. Though the asymmetric port design has a lower flow coefficient, both port designs provide the necessary flow to achieve performance targets. Overall, as the asymmetric port satisfies the flow requirement and offers advantages in both turbulence and high swirl (which both enable extended highly dilute low-temperature combustion), the asymmetric inlet port design is selected for further combustion studies.

Charge Motion – Mixing field and flame propagation behavior

The effect of swirl on in-cylinder mixing field and flame propagation is briefly discussed in this sub-section. Swirl with tumble (referred to as swumble) has a profound effect on in-cylinder mixing and flame kernel generation. It is observed from the spatial distribution of in-cylinder residual gases (Figure 13) that pure tumble flow field generates more evenly distributed in-cylinder mixing than swumble, while swumble provides radial stratification in residual gas. Spatial stratification in charge dilution, and in-cylinder residuals in swumble flow configuration can thus be used to design the combustion recipe including pathways for flame development.

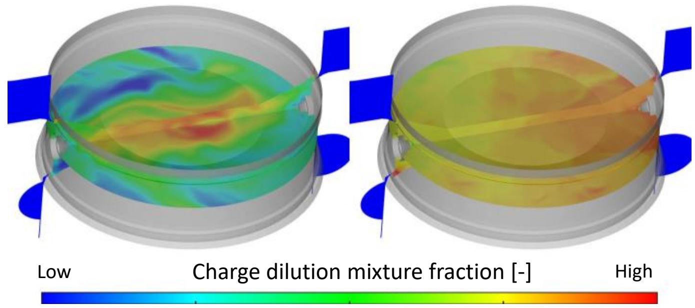

Figure 13: Distribution of internal EGR inside the combustion chamber at CA50 (left: Swumble, right: Tumble)

Figure 14 shows flame propagation behavior during early kernel growth. In this figure, the flame is depicted by an iso-temperature surface colored by local TKE. It shows that the swumble system promotes kernel growth along the periphery of the cylinder chamber. In the swumble system, flame is assisted along the peripheral region due to presence of mean cylinder velocity (swirl), and followed by an inward flame propagation towards the center of the cylinder with chemical conversion. Conversely, in the tumble system, the flame propagates towards the center of the combustion chamber from the two spark plugs placed diametrically opposite in the cylinder and is promoted by turbulence enhancement.

Flame propagation to the periphery of the chamber serves an important role in combustion efficiency and HC emissions which are most prevalent in the geographic location in the cylinder that the flame reaches last. It is observed that the presence of swumble helps in mitigating HC emissions by facilitating kernel development in the regions where flame has the highest propensity to quench.

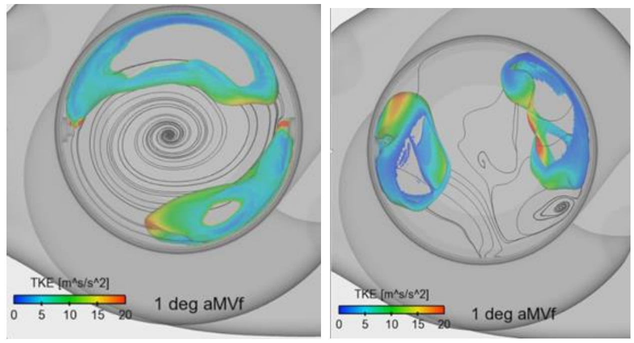

Figure 14: View of flame propagation in swumble (left) and tumble (right) configuration

Results from massively parallel multi-dimensional combustion simulations at 2000 RPM, 3 bar BMEP

A computational campaign has been carried out to survey performance, emission, knock, and stability limits of the engine operation at the 2000 RPM and 3 bar BMEP (Table 2). At each of these cases, multiple simulations are carried out with spark timing sweeps to understand the effect of CA50 variations and ascertain the response of important combustion and emission parameters against combustion phasing. Between cases 1-12, the net in-cylinder EGR is varied between 36 to 42%. Both swumble and tumble combustion systems are investigated along with two different piston crown shapes (Figure 15). The piston crown shapes chosen for this study are both axisymmetric about the bore axis and identical inlet to exhaust side of the opposed piston combustion chamber. The difference between these piston crowns is most prominent the central riser as the SIM0336 design offers a piston-to-piston squish zone and intended to minimize flame touch-down with a swumble/peripheral combustion flame propagation. An 18:1 CR is maintained for both pistons when cranks are phased in-sync. The CR can be reduced from 18:1 by the action of a VCR crank phaser.

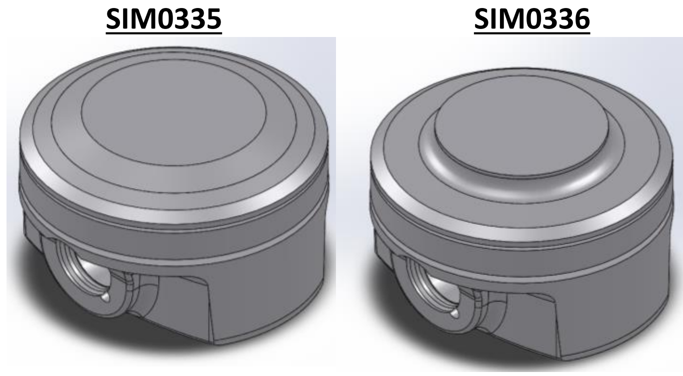

Figure 15: Piston crown for 1.2L engine with maximum CR of 18:1

Table 2: Dilution level, flow configuration and piston crown investigated at the 2000 RPM, 3 bar BMEP condition

Case ID	EGR [%]	In-cylinder Flow Configuration	CR	Piston Crown ID
1	42	Swumble	18:1	SIM0335
2	42	Tumble	18:1	SIM0335
3	42	Swumble	18:1	SIM0336
4	42	Tumble	18:1	SIM0336
5	37	Swumble	18:1	SIM0335
6	46	Swumble	18:1	SIM0335
7	37	Tumble	18:1	SIM0335
8	46	Tumble	18:1	SIM0335
9	37	Swumble	18:1	SIM0336
10	46	Swumble	18:1	SIM0336
11	37	Tumble	18:1	SIM0336
12	46	Tumble	18:1	SIM0336
21	42	Partial Swumble	18:1	SIM0336
22	42	Partial Swumble	17:1	SIM0336
23	42	Partial Swumble	16:1	SIM0336

Fuel efficiency maps

Results from meta-model correlations of gross-indicated efficiency (GIE) are presented for the different engine designs studied (i.e., flow configuration and piston crown) across EGR and CA50 phasing degrees of freedom. These GIE maps are overlaid with contours of knock and unacceptable combustion stability regions. The combustion stability limits are derived based on correlation of spark to CA10 timing based on previous experimental analysis. The knock is correlated based on crank-angles resolved local pressure data from several virtual sensors placed across the combustion chamber at minimum volume. Knock is directly predicted from the multi-dimensional simulations using detailed chemistry and surrogate fuel formulations. The numerical model and methods used in this paper is further elaborated in a different publication by the authors in this technical conference. Figure 16 shows these GIE maps for tumble and swumble configurations respectively for the piston crown SIM0336 at one compression ratio.

The key takeaway is that while the swumble system has significantly faster combustion and an improved dilution tolerance, there is an associated gross fuel efficiency penalty compared to tumble system because of higher heat losses from the higher wall flow velocities. Maximum gross indicated efficiency occurs at CA50 between 5-10 deg aMVf with 42 % EGR (tumble) and 44% (swumble).

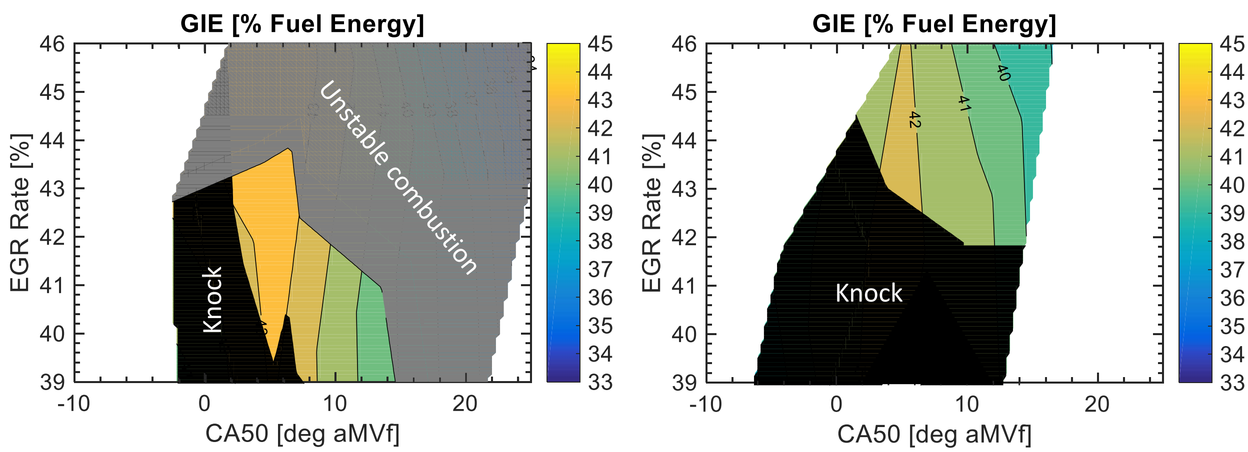

Figure 16: Gross indicated efficiency map overlaid with regions of unstable combustion (grey) and knock (black) with piston SIM0336 in tumble (left) and swumble (right) configurations

GIE maps of the piston crown SIM0335 in tumble and swumble configurations (Figure 17) are also presented to study the effect of piston crown shapes on efficiency, knock, and combustion stability. It is concluded that there is little difference in fuel efficiency between these two piston crown shapes. However, it is observed that knock propensity can be manipulated effectively with the piston crown shapes. This provides insight that future studies on piston crown shapes may be leveraged to reduce knock propensity.

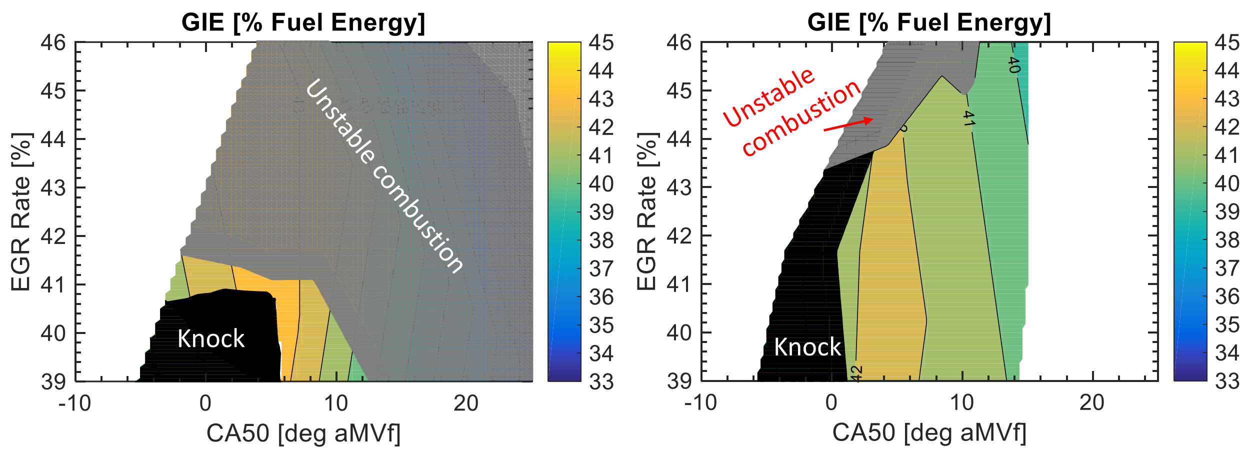

Figure 17: Gross indicated efficiency map overlaid with regions of unstable combustion (grey) and knock (black) with piston SIM0335 in tumble (left) and swumble (right) configurations

Effect of swirl ratio on GIE map

Since swirl can be manipulated by the secondary throttle restriction level, a follow-up investigation was conducted with partially open secondary throttle position. By modulating swirl at a fixed dilution level, the effect of swirl against GIE, knock, and combustion stability is presented (Figure 18). It is seen that gross indicated efficiency is improved up to a swirl level of approximately 3, after which GIE reduces due to increased wall heat transfer. It is also observed that at swirl ratio of ~3, there is much wider region of control authority over combustion than at 0 swirl (pure tumble configuration). This leads to the conclusion that the secondary throttle plays an important role to improve engine performance over variability in cylinder-to-cylinder or primary engine condition control parameters.

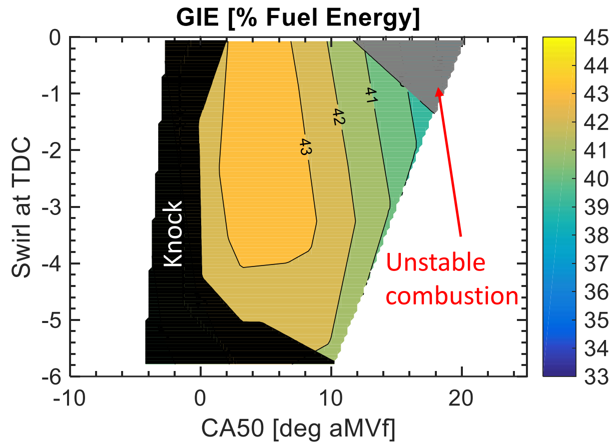

Figure 18: Gross indicated efficiency map overlaid with regions of unstable combustion (grey) and knock (black) with fixed dilution (42% EGR) with piston SIM0336

Emission maps

Fuel-specific NOx (fsNOx) in g of NOx residual in the combustion chamber per kg of injected fuel is calculated at EVO. Contour maps of fsNOx from tumble and swumble configurations (Figure 19) are compared. For most of the engine operation at 2000 RPM, 3 bar BMEP, the fsNOx is below 5 g/kg-f. This low NOx emission is due to low-temperature combustion enabled by highly dilute cooled EGR. It is observed from these results that flow configurations of the swumble system enables stable operation in a region that can offer NOx emission reductions

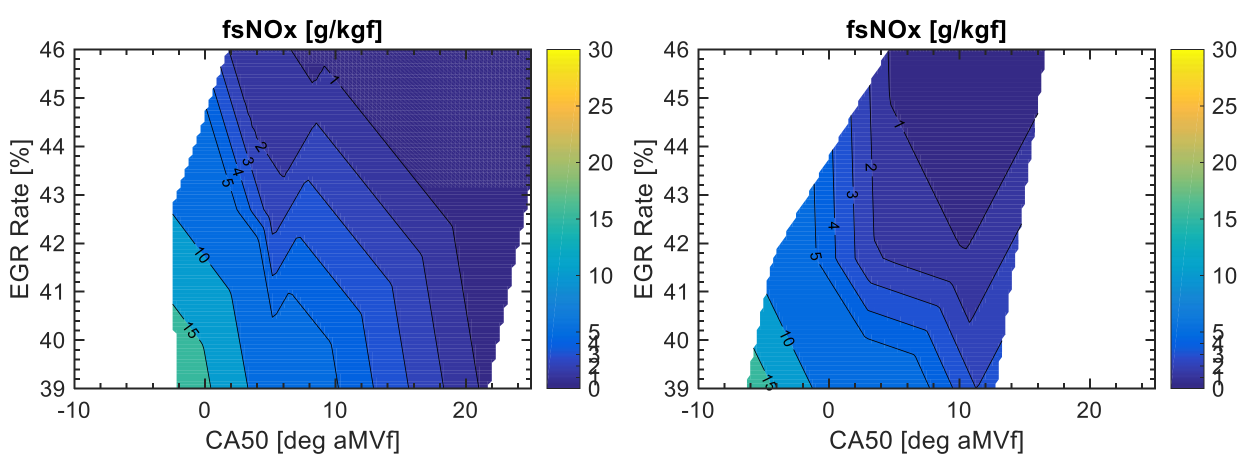

Figure 19: Fuel specific NOx map in tumble (left) and swumble (right) configurations with piston SIM0336

Fuel-specific HC emission (fsHC) in g of un-burned HC in combustion chamber per kg of injected fuel is calculated at EVO. Contour maps of fsHC from tumble and swumble configurations are compared in Figure 20.

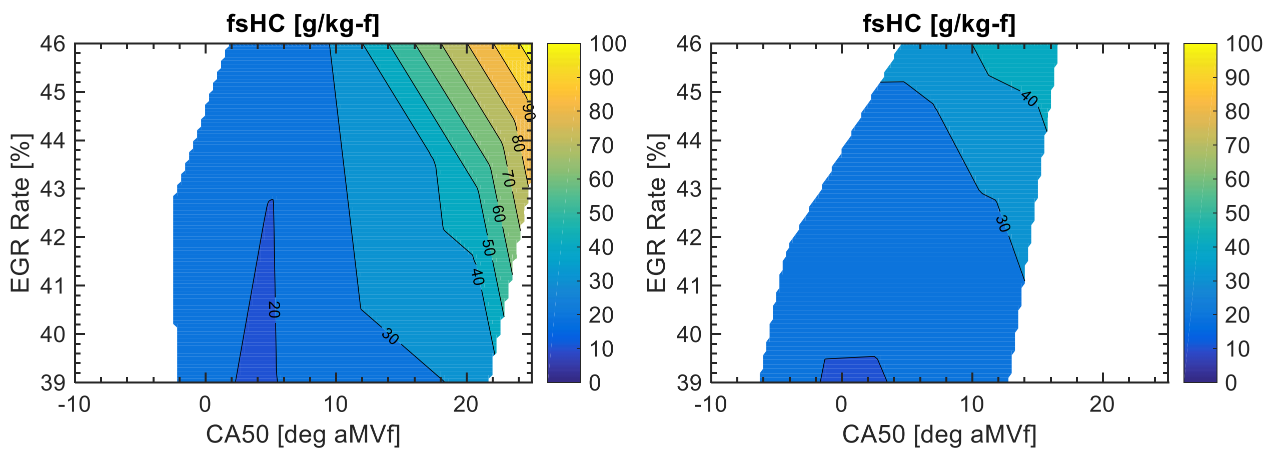

Figure 20: Fuel specific HC map in tumble (left) and swumble (right) configurations with piston SIM0336

It is observed that operating at optimal EGR and CA50 phasing, similar fsHC emissions are generated between tumble and swumble systems. However, with the tumble system at high EGR levels, it is noted that fsHC increases quicker with ignition delay (larger CA50 or combustion phasing retard) than the quicker-burning swumble system. This because the swumble system promotes flame propagation into areas more prone to quenching than seen with the tumble configuration.

Both fsNOx and fsHC (Figure 21) emission responses are presented against swirl and CA50 at a fixed dilution level. It is seen from this figure that both NOx and HC have a high degree of sensitivity to combustion phasing (CA50). fsHC emissions do not respond strongly to swirl level, but fsNOx shows the impact of swirl on peak NOx is highest at approximately 3. This peak in NOx production is proposed to be due to competing interactions of combustion speed and heat transfer, both of which increase with swirl. Faster heat release tends to reduce HC and improve GIE if unburned volumes exist, whereas increasing heat transfer tends to reduce GIE from thermal losses. This peak NOx highlights the swirl at which in-cylinder temperatures are highest and hence heat release is fastest, akin to a slightly lean condition in a conventional engine. The reduction of NOx with increasing swirl is interesting as it is likely the increasing rate of heat transfer is expected to mitigate the increase in chemical heat release.

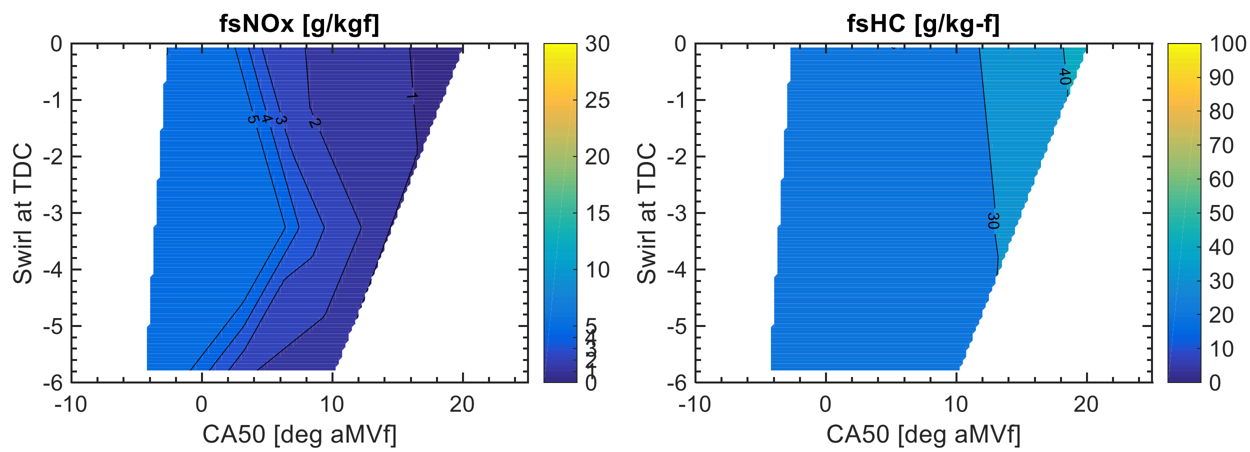

Figure 21: Fuel specific NOx (left) and HC (right) maps with fixed dilution (42% EGR) with piston SIM0336

It is proposed the reason for the HC not reducing measurably with increasing swirl (e.g., relative to zero swirl level) is that the tumble system (i.e. no swirl) already has a thorough conversion of fuel at this condition. Hence, negligible additional unburned fuel can be converted with the increased heat release rate offered by swirl. This insensitivity of HC against swirl is not expected to maintain at an engine operating condition where the tumble system leaves some volume of unburned fuel.

<u>Effect of compression ratio on fuel efficiency</u>

With the VCR mechanism, CR can be manipulated as a control parameter to study its effect on fuel efficiency, knock, and combustion stability. Figure 22 shows a GIE map against CR and CA50 combustion phasing for 2000 RPM, 3 bar BMEP engine operating condition. Using second-order statistical meta-model (quadratic response function), an optimization of GIE against CR is conducted. Figure 23 shows that for every 0.5 increase in compression ratio, ~0.5% gross indicated fuel efficiency gain is realized between 16:1 and 18:1 compression ratio. The scope of this study was limited to 18:1 compression ratio.

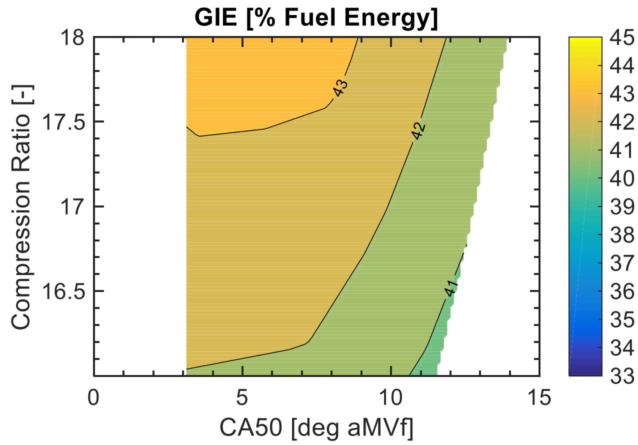

Figure 22: Gross indicated efficiency map with VCR at fixed dilution (42% EGR) and with piston SIM0336

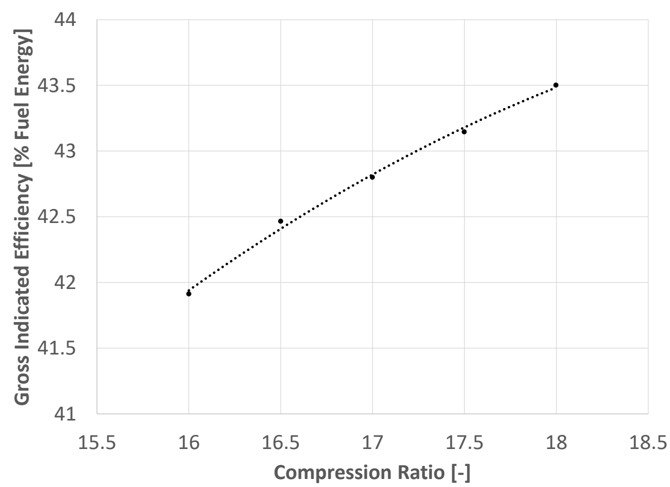

Figure 23: Maximum GIE against CR with fixed EGR and swirl number while maintaining emission, knock and combustion stability margins

<u>**Engine operating map**</u>

A system level 1D/0D simulation is carried out following the multi-dimensional simulation results to evaluate drive cycle averaged emission and fuel economy against development target (Table 3). The targets for emissions are derived from Tier3, Bin 160 (similar to Euro 6) emission for vehicle class N1 (<1350 kg), “positive ignition” engine [12].

Table 3: Drive cycle emission and efficiency. System level simulation vs. target for the 1.2L development engine

	CO [g/km]	HC [g/km]	NOx [g/km]	Fuel Economy [km/L]
Target	0.5	0.03	0.036	24
Non-catalyzed engine out	0.11	1.34	0.028	24.2

Based on the numerical studies it is found that 1.2L engine will be able to meet Tier 3, Bin 160 (similar to Euro 6) emission standards with only HC oxidation catalyst. For vehicle class N1 this 1.2L development engine is on target to meet CO and NOx emissions without catalyst. From these simulations it is expected that the vehicle fuel economy target of 24.2 km/L will be met.

CONCLUDING REMARKS

This paper presents high-performance computer assisted numerical studies of 1.2L 4-stroke opposed piston sleeve valve development engines. Massive computational campaign was carried out to evaluate design and control concepts of the development engines using multi-dimensional engine simulations. From these computational combustion results, reduced order meta-models were constructed to accelerate analysis-led design evaluations. Later system level simulations were carried out to evaluate drive cycle emission and efficiency and compared with Tier 3, Bin 160 (similar to Euro 6) emissions requirement as well as fuel economy targets for engine program

Design concepts were evaluated using multi-dimensional engine modeling and it is observed that:

- Swumble system concept enhances dilution tolerance of low-temperature combustion operation.
- Collapse of dual tumble in opposed piston architecture produces enhanced in-cylinder turbulence to support dilute gasoline combustion
- In-cylinder design features do play important role to manipulate flame dynamics and to improve combustion stability
- Using crank phasers, the compression ratio can be adjusted to result in higher fuel efficiency at cruising condition. To limit knocking, compression ratio is lowered at higher load engine operating conditions.

Currently Pinnacle is building a prototype engine to conduct engine testing based on design principles and guidance obtained from these massive computation campaign. After obtaining engine test data, these models will be validated further to guide further refinements for production ready engine.

NOMENCLATURE

AHRR	Apparent Heat Release Rate
ALDD	Analysis-Led Design and Development
AMR	Automated Mesh Refinement
aMVf	After Fired Minimum Volume
BMEP	Brake Mean Effective Pressure
CA	Crank Angle time
CA50	Crank Angle corresponding to 50% heat release
CAD	Computer Aided Design
CFD	Computational Fluid Dynamics
CI	Compression Ignition
COV	Co-efficient of Variation
DI	Direct Injection
DoE	Design of Experiments
EER	Effective Expansion Ratio
EGR	Exhaust Gas Recirculation
fsHC	Fuel-specific Hydro Carbon
fsNOx	Fuel-specific Nitrogen Oxides
GIE	Gross Indicated Efficiency
HC	Hydro Carbon
HPC	High Performance Computing
HTC	Heat Transfer Co-efficient
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Close
LTC	Low-Temperature Combustion

NOx Nitrogen Oxides

OLCF Oak Ridge Leadership Computing Facility

RPM Revolutions Per Minute

SI Spark Ignition

TDC Top Dead Center

TKE Turbulent Kinetic Energy

TWC Three Way Catalyst

ACKNOWLEDGMENTS

This research used resources of the Oak Ridge Leadership Computing Facility, which is a DOE Office of Science User Facility supported under Contract DE-AC05-00OR22725. This manuscript has been authored by UT-Battelle, LLC under Contract No. DE-AC05-00OR22725 with the U.S. Department of Energy. The United States Government retains and the publisher, by accepting the article for publication, acknowledges that the United States Government retains a non-exclusive, paid-up, irrevocable, worldwide license to publish or reproduce the published form of this manuscript, or allow others to do so, for United States Government purposes. The Department of Energy will provide public access to these results of federally sponsored research in accordance with the DOE Public Access Plan (http://energy.gov/downloads/doe-public-access-plan). Authors would also like to acknowledge Convergent Science for their technical support in conducting this work.

FUNDING

This research work was funded by the U.S. Department of Energy (DOE) under Small Business Voucher Program Cooperative Research and Development Agreement (CRADA) NO. NFE-16-06356.

REFERENCES

1. Willcox, M. A., & Cleeves, J. M. (2013). Drive Cycle Fuel Economy and Engine-Out Emissions Evaluation Using an Opposed-Piston Sleeve-Valve Engine with Lean Operation and Ignition Delay for NOx Control. SAE 2013-32-9064
2. Alger, T., Walls, M., Chadwell, C., Joo, S. et al., (2016). The Interaction between Fuel Anti-Knock Index and Reformation Ratio in an Engine Equipped with Dedicated EGR. SAE Int. J. Engines 9(2):786-795
3. Gottschalk, W., Lezius, U., & Mathusall, L. (2013). Investigations on the Potential of a Variable Miller Cycle for SI Knock Control. In SAE Technical Paper 2013-01-1122. http://doi.org/10.4271/2013-01-1122

4. Ishikawa, S., Kadota, M., Yoshida, K., Takahashi, K., & Kawajiri, S. (2009). Advanced Design of Variable Compression Ratio Engine with Dual Piston Mechanism. SAE International Journal of Engines, 2(1), 2009-01–1046. http://doi.org/10.4271/2009-01-1046
5. Richards, K. J., Senecal, P. K., & Pomraning, E., (2016) CONVERGE (v2.3), Convergent Science, Inc., Madison, WI
6. Fan, L., Reitz, R., & Trigui, N. (1999). Intake flow simulation and comparison with PTV measurements. SP-Society of Automotive Engineers, 1999(724). http://doi.org/10.4271/1999-01-0176
7. Naik, C. V, Puduppakkam, K., Wang, C., Kottalam, J., Liang, L., Hodgson, D., & Meeks, E. (2010). Applying Detailed Kinetics to Realistic Engine Simulation: the Surrogate Blend Optimizer and Mechanism Reduction Strategies. SAE International Journal of Engines, 3(1), 2010-01–0541. http://doi.org/10.4271/2010-01-0541
8. O'Rourke, P. J., & Amsden, A. A. (2000). A Spray/Wall Interaction Submodel for the KIVA-3 Wall Film Model. In Engineering. http://doi.org/10.4271/2000-01-0271
9. Shi, Y. H.-W. G. R. R. (2011). Computational Optimization of Internal Combustion Engines. http://doi.org/10.1007/978-0-85729-619-4
10. Han, Z., Uludogan, A., G. Hampson, & Reitz, R. (1996). Mechanism of Soot and NOx Emission Reduction Using Multiple-injection in a Diesel Engine. In SAE (Society of Automotive Engineers) Transactions (Vol. 960633). http://doi.org/10.4271/960633
11. Finney, C. E. A., Stoyanov, M. K., Pannala, S., Daw, C. S., Wagner, R. M., Edwards, K. D., … Green, J. B. (2014). Application of High Performance Computing for Simulating the Unstable Dynamics of Dilute Spark-Ignited Combustion (pp. 259–270). http://doi.org/10.1007/978-3-319-02925-2_23
12. Williams, M., & Minjares, R. (2016). A technical summary of Euro 6/VI vehicle emission standards. Retrieved from http://www.theicct.org/sites/default/files/publications/ICCT_Euro6-VI_briefing_jun2016.pdf

ANNEX A

PRILIMINARY VALIDATION OF SIMULATION MODEL

The multi-dimensional simulation sub-models were developed in lieu with predicting combustion and emission characteristics of opposed-piston 4-stroke engines. Test data from a lean-burn version of Pinnacle Engine was used to validate and test the multi-dimensional model. A more detailed validation method is discussed in ASME ICEF2017-2618. Below is summary of the model validation results.

Here is a summary of model validation. A test engine operating condition (4000 RPM, 3 bar IMEP) was chosen to validate model. A nominal compression ratio of 15:1 and in-cylinder equivalence ratio of 0.626 (lean-burn) is chosen for the validation test. A new chemical kinetics mechanism was developed to match laminar flame speed data from various literature source representing wide ranging conditions ranging both lean and dilute conditions. The CFD simulation with modified mechanism at the test engine operating condition showed good match (Figure 24). More details of the validation methods, new mechanism development is discussed in the ASME ICEF2017-2618 paper.

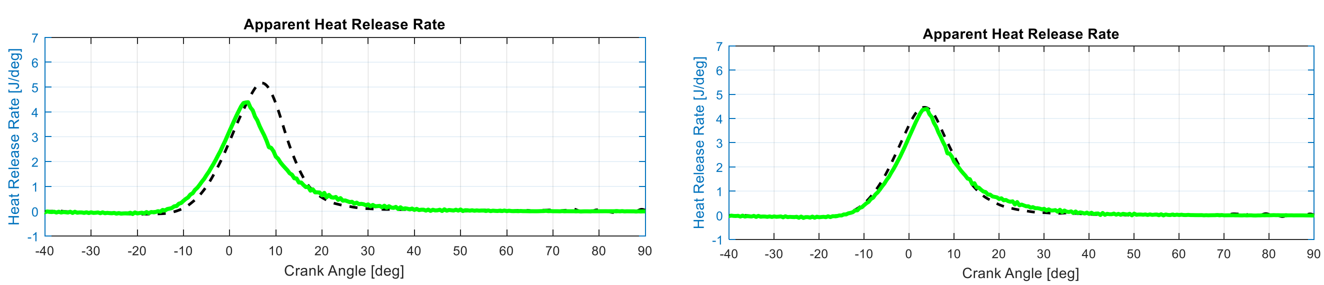

Figure 24: Apparent heat release rate from simulation (light green) and test (dotted black). Left: With original chemical kinetics mechanism, Right: From newly developed chemical kinetics mechanism and validation of the simulation model

Figure Caption List

Table Caption List