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INITIAL RESULTS ON A NEW LIGHT-DUTY 2.7L OPPOSED-PISTON GASOLINE COMPRESSION IGNITION MULTI-CYLINDER ENGINE

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ABSTRACT

Gasoline compression ignition (GCI) is a cost-effective approach to achieving diesel-like efficiencies with low emissions. Traditional challenges with GCI arise at low-load conditions due to low charge temperatures causing combustion instability and at high-load conditions due to peak cylinder pressure and noise limitations. The fundamental architecture of the two-stroke Achates Power Opposed-Piston Engine (OP Engine) enables GCI by decoupling piston motion from cylinder scavenging, allowing for flexible and independent control of cylinder residual fraction and temperature leading to improved low load combustion. In addition, the high peak cylinder pressure and noise challenges at high-load operation are mitigated by the lower BMEP operation and faster heat release for the same pressure rise rate of the OP Engine. These advantages further solidify the performance benefits of the OP Engine and demonstrate the near-term feasibility of advanced combustion technologies, enabled by the opposed-piston architecture.

This paper presents initial results from a steady state testing on a brand new 2.7L OP GCI multi-cylinder engine. A part of the recipe for successful GCI operation calls for high compression ratio, leading to higher combustion stability at low-loads, higher efficiencies, and lower cycle HC+NO_X emissions. In addition, initial results on catalyst light-off mode with GCI are also presented. The OP Engine's architectural advantages enable faster and earlier catalyst light-off while producing low emissions, which further improves cycle emissions and fuel consumption over conventional engines.

NOMENCLATURE

AHRR	Apparent heat release rate
aMV	After Minimum Volume
BMEP	Brake Mean Effective Pressure
BSCO	Brake Specific CO
BSFC	Brake Specific Fuel Consumption
BSHC	Brake Specific HC
BSNOx	Brake Specific NO_X
BSSoot	Brake Specific Soot
BTE	Brake Thermal Efficiency
CA50	Crank Angle Location of 50% Mass Fraction Burned
CO	Carbon Monoxide
CoV	Coefficient of Variation
CR	Compression Ratio
EGR	Exhaust Gas Recirculation
FTP75	Federal Test Procedure
GCI	Gasoline Compression Ignition
HC	Hydrocarbon
HP EGR	High Pressure EGR
HRR	Heat Release Rate
IMEP	Indicated Mean Effective Pressure
ITE	Indicated Thermal Efficiency
LD	Light-duty
LP EGR	Low Pressure EGR
MCE	Multi Cylinder Engine
MY	Model Year
NMOG	Non-methane organic gas

NOx	Nitrogen Oxides
OP	Opposed-Piston
SCE	Single Cylinder Engine
SCR	Selective Catalyst Reduction
SOI	Start of Injection

OPPOSED-PISTON ENGINE FUNDAMENTALS

Reduced Heat Transfer Losses

The Achates Power Opposed-Piston Engine configuration has two pistons facing each other in the same cylinder, combining the stroke of both pistons to increase the effective stroke-to-bore ratio. The OP Engine architecture eliminates the cylinder head of a conventional engine, thus reducing the surface area to volume ratio, reducing heat transfer losses and increasing thermal efficiency [1-6]. A conceptional comparison between a conventional engine and the OP Engine is shown in Figure 1. At the same bore, the surface area to volume ratio is reduced by more than 30% for the OP Engine.

Engine	Conventional	OP Engine
Cylinders	6	3
Trapped Volume/Cylinder	1.0L	**1.6L**
Stroke/Bore Ratio	1.1	**2.2**
Trapped Compression Ratio	15:1	15:1
Combustion Chamber Surface Area/Volume (mm^{-1})	0.28	**0.18**

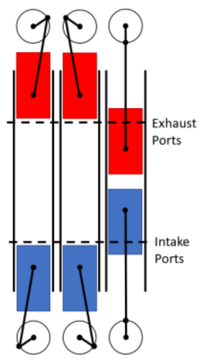


Figure 1: OP engine schematic, figure not to scale

Further heat loss reductions are enabled due to higher wall temperature of the two piston crowns from two-stroke operation compared a cooling stroke and presence of a cylinder head in conventional engines, reducing the temperature differential between hot combustion gases and the wall.

An additional benefit of the reduced heat losses in the OP Engine is the reduction in radiator size and fan power, enabling lower vehicle drag losses and increasing vehicle fuel efficiency.

Lower Pumping Losses

The pistons in an OP Engine are decoupled from inducting fresh air and exhausting combustion products. The scavenging of the cylinder is governed by the pressure ratio across the intake and exhaust ports, which is controlled by the supercharger, variable geometry turbocharger, and backpressure valve positions (schematic shown in Figure 2). This configuration minimizes engine pumping losses as the cylinder does not need to be fully scavenged every cycle, i.e. during idle or low-load conditions, only a fraction of the exhaust gases are scavenged and replaced with fresh air, just sufficient enough for the next combustion cycle. This architectural advantage is a key enabler to the flat fuel map of the OP Engine. Partial scavenging of the cylinder also enables control over the trapped residual fraction, enabling high combustion stability and rapid engine warm up from cold start [7-9].

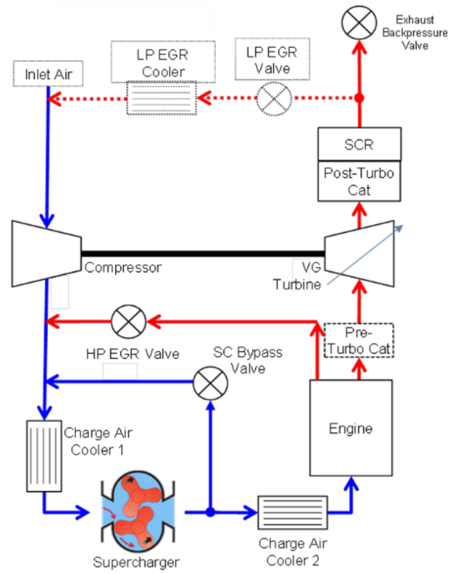


Figure 2: OP Engine air handling schematic

An additional pumping advantage of the OP Engine is the larger intake and exhaust port flow area compared to a conventional engine, reducing choked and restricted flow and further decreasing pumping losses.

Earlier and faster combustion

Equation 1 describes the first law of thermodynamics for conventional and OP engines, where Q is the heat released, θ is the crank angle, γ is the ratio of specific heats, p is the cylinder pressure, and V is the cylinder volume. The larger combustion volume, highlighted by the grey boxes, for the given amount of energy released also enables shorter combustion duration while preserving the same maximum pressure rise rate [10]. The faster combustion improves thermal efficiency by reaching a condition closer to constant volume combustion. The lower heat losses as described above lead to a 50% mass fraction burn location closer to minimum volume, as shown in Figure 3, and reduces fuel consumption further.

Equation 1: 1st law of thermodynamics showing faster rate of heat release with OP Engine at same pressure rise rate

1st Law of Thermodynamics:

Conventional Engine

$$\frac{dQ_{4S-AHR}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \boxed{\frac{dp}{d\theta}} \quad \text{same}$$

OP Engine

$$\frac{dQ_{2S-AHR}}{d\theta} = \frac{\gamma}{\gamma-1} p \frac{d(1.6V)}{d\theta} + \frac{1}{\gamma-1} (1.6V) \boxed{\frac{dp}{d\theta}}$$

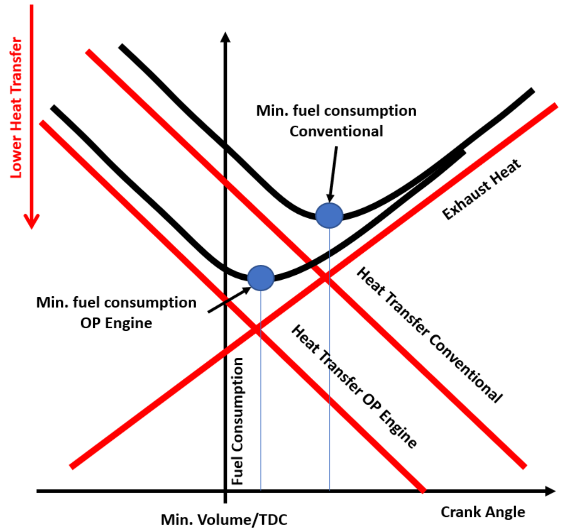


Figure 3: Lower OP Engine heat transfer losses enable earlier combustion phasing for lower fuel consumption

Cleaner combustion

Due to the elimination of the cylinder head, fuel is introduced tangentially to the piston surface and does not use the piston to break apart the fuel spray as in conventional diesel engines. This allows for the optimization of the piston shape to generate high turbulent kinetic energy while minimizing combustion surface area to volume ratio (heat transfer), leading to improved spray atomization, vaporization, and lower soot emissions. Additionally, the lower load two-stroke operation of the OP Engine and ability to retain internal EGR without incurring additional pumping work results in lower NO_X emissions.

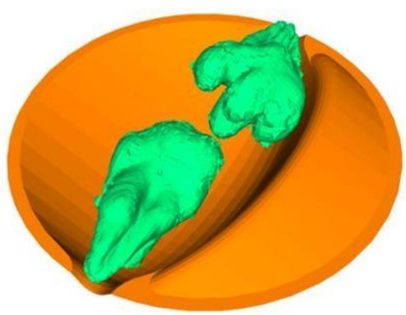


Figure 4: Diametrically opposed fuel injectors injecting fuel tangentially to the piston surface

COMBINING OP AND GCI

A significant amount of pioneering research as been conducted on GCI [11-22]. The opportunities and lessons learned form the basis for GCI on the OP Engine, with the added benefit of the opposed-piston architecture addressing some of the four-stroke GCI challenges.

Mixture preparation

Robust and clean GCI combustion requires a stratified charge, with locally lean and rich areas, and multiple injection events. Delphi Technologies has achieved excellent GCI combustion results in conventional engine configurations with an injector inserted through the cylinder head injecting towards an approaching piston

The OP injection environment offers significant potential to improve charge stratification. Diametrically opposed dual injectors spray across the diameter of cylinder. Each injector can be independently controlled to more easily manage staggered injections for ideal mixture distribution and, therefore, efficient and controlled heat release [23, 24].

Charge temperature management

At low-loads, GCI requires higher temperatures for combustion. Four-stroke engines normally push the entire content of the cylinder out during the exhaust stroke and therefore require a complex variable valvetrain to re-open the exhaust valve during the intake stroke to re-induct the exhaust back in the cylinder to increase the charge temperature to the level necessary for GCI ignition.

The OP Engine, however, can retain exhaust gas in-cylinder after combustion, even at low-loads when relatively little additional intake oxygen is required, by reducing the scavenging of the cylinder. At low-loads, the OP Engine can reduce the supercharger work used to boost the intake manifold pressure. This has four benefits: it reduces the amount of work by the supercharger, reducing pumping; it keeps in-cylinder temperatures high for good combustion stability; it provides a natural or internal EGR effect for low NO_X combustion and, it provides high exhaust gas temperatures for catalyst light-off and sustained activity.

2.7L OPPOSED-PISTON MULTICYLINDER DESIGN

Engine specifications

A new multi-cylinder OP Engine was designed and built from scratch and is geared toward the light-duty vehicle sector. Specifications for the engine are shown in Table 1, with a labeled Computer-Aided Design (CAD) image of the engine shown in Figure 5.

Table 1: 2.7L OP GCI engine specifications

Displacement (L)	2.7
Cylinders	3
Compression ratio (-)	18.5
Power (kW)	200
Torque (Nm)	650
Bore (mm)	80
Stroke (mm)	177
Exhaust crank lead (deg)	8-12
Air handling	VG Turbo charger, super charger, high pressure EGR
Fuel injection system	Delphi Technologies injectors, 2 per cylinder, capable of 6 injection events per injector
ECU	Pi Innovo Open

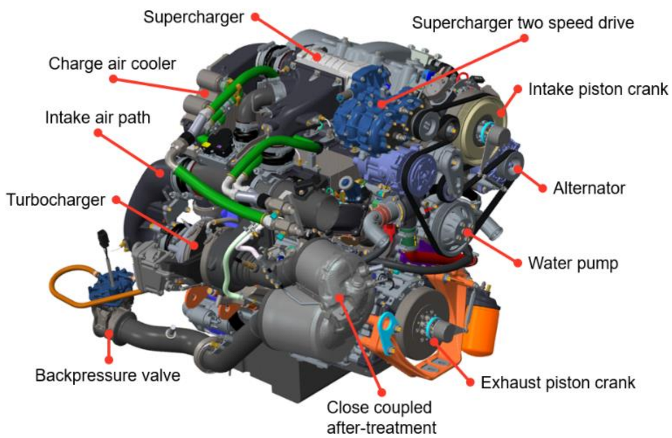


Figure 5: Isometric CAD view of the new 2.7L OP GCI engine

The intake piston crank is located on top of the engine, with the exhaust piston crank on the bottom of the engine. The cylinders are tilted 30 degrees from vertical to package into currently existing vehicles. The mechanical connection that links the two crankshafts together is a novel 3 gear geartrain, with power take off on the exhaust crankshaft and is shown in Figure 6.

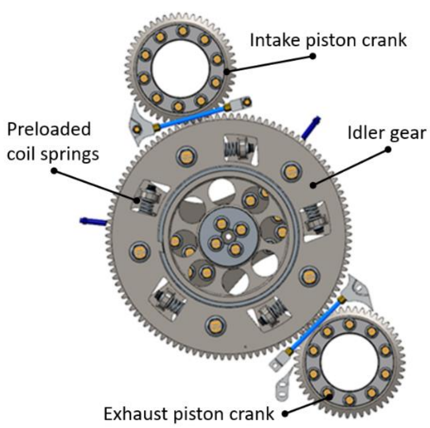


Figure 6: 2.7L OP Engine gear train connecting intake and exhaust crankshafts

The air handling of the engine is packaged on the opposite side of the tilted cylinders, giving the visual impression of a Vee style engine. The air path is similar to previous papers. The fresh airpath is as follows (Figure 2): air is inducted by the turbocharger compressor, mixed with high pressure EGR, cooled by a charge air cooler, compressed by a supercharger, flow is split between supercharger recirculation and flow through an intercooler, and finally into the intake chest. After combustion, exhaust gases split between the high-pressure EGR loop and VG turbine flow. After the VG turbo charger, the exhaust gas flows through a close-coupled after-treatment system (not studied in this paper), through a backpressure valve, through an underfloor SCR (not studied in this paper), and then to the test cell air management system.

An electric water pump was used for engine cooling, and the power consumption is accounted for in the brake numbers presented. An alternator efficiency of 60% was assumed.

Fuel system specifications

The fuel injection process and fuel sprays are key to achieving a successful combustion system with high efficiency, low emissions, and low combustion noise. The injection pressure requirement of 1800 bar is higher than gasoline fuel systems currently. Therefore, a diesel fuel system was specified for operation on US E10 gasoline with a lubricity additive.

A CAD rendering of the fuel system is shown in Figure 7. It is comprised of two independent systems, each with one pump, one rail, high-pressure lines, and three injectors for each side of the engine. Two injectors are mounted diametrically opposed in each cylinder. The two fuel rails may be operated at different pressures. This configuration provides great flexibility in the injection process for fuel quantity, timing, and splits.

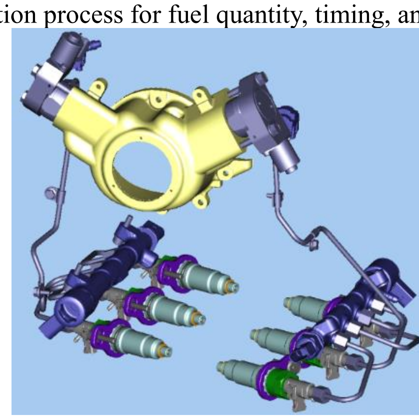


Figure 7: 2.7L OP Engine fuel injection system, with two independent pumps, rails, and injectors

Two diesel unit pumps with roller lifters are mounted on the front cover of the engine and are driven simultaneously by the intake crankshaft with a three-lobe cam. The pumps (Figure 8) are very compact, are lubricated by engine oil, and are mechanically very efficient.

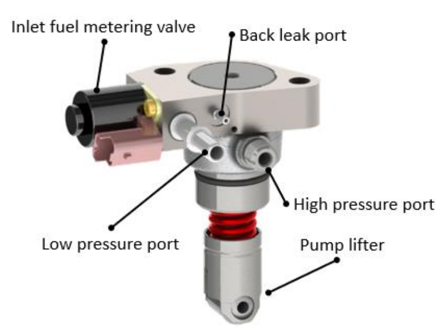


Figure 8: Delphi Technologies diesel unit pump with roller lifter and inlet metering valve

The injectors shown in Figure 7 were specially built for an opposed-piston engine operating on gasoline fuel. The injector features top feed fuel inlet, electrical connection on the body side, and short overall injector length (137mm). Since gasoline fuels have very low viscosity relative to diesel, back leak flows will be significantly increased, and more pump work will be required. This injector features a pressure balanced control valve, which greatly reduces back leak flows, especially at higher pressures. The injector features fast response for near square injection profiles. Figure 9 shows typical injection rate and drive current at 1200 bar fuel pressure.

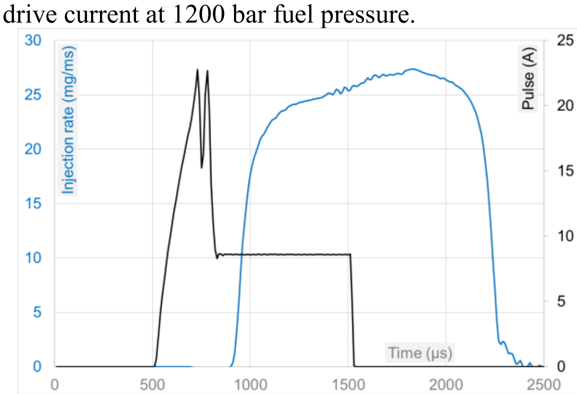


Figure 9: Injection rate and drive current at 1200 bar fuel pressure

Testing specifications

Gasoline fuel specifications are shown in Table 2. The fuel flow is measured using a Resol fuel system (model number RS474BCX-40), the air flow is measured using a Meriam laminar flow element (model number Z50MH10-5), CO, O_2, CO_2, and HC emissions are measured using a CAI emissions analyzer, NO_X emissions are measured using a MKS FTIR, and soot values are measured with an AVL 415 smokemeter.

Table 2: Gasoline fuel specifications

Fuel		Gasoline
Ethanol	%vol	10
RON	-	91
MON	-	83
AKI	-	87

INITIAL RESULTS

Initial cycle average results

The following results are after only 10 weeks of testing the brand new 2.7L OP GCI engine. Operating the engine over a 10-mode steady state representation of the transient FTP75 cycle yields a cycle average BTE of 31.7% on the hot LA4 cycle (Table 3). Even after minimal development time, the engine is already showing a 11% improvement compared to a competitive MY2015 four-stroke engine [25].

Table 3 also shows the cycle average emissions. The initial targets for the OP GCI engine is U.S. EPA light duty Tier 3 Bin 160, which has a tailpipe NMOG + NO_X requirement of 160 mg/mile, CO requirement of 4.2 g/mile, and PM requirement of 3 mg/mile, and a final target of U.S. EPA light duty Tier 3 Bin 30. An initial study with an after-treatment supplier using off the shelf diesel after-treatment components and the 10-mode approximation of a transient cycle indicated successful achievement of Bin 160 levels. Tier 3 Bin 30 emissions levels are expected with a gasoline specific after-treatment, implementation of catalyst thermal management, and an actual transient cycle instead of a steady state approximation.

Table 3: 2.7L OP GCI hot LA4 cycle average results

BSFC	272.1	g/kWh	$BSNO_X$	2.0	g/kWh
ISFC	204.7	g/kWh	BSSoot	0.03	g/kWh
BTE	31.7	% Fuel	BSCO	3.5	g/kWh
ITE	42.1	% Fuel	BSHC	1.3	g/kWh
Pumping Loss	1.8	% Fuel			
Friction Loss	8.5	% Fuel	FTP NO_X	0.82	g/mi
			FTP Soot	0.011	g/mi
			FTP HC	0.553	g/mi

A sample cylinder pressure, combustion profile, and fuel injection traces are shown in Figure 10 at 1275 RPM, 173 Nm of torque. An early pilot is utilized during the compression stroke of the engine and a main injection event near the minimum volume location, which is similar to other published works [13, 20, 26]. The early timing is required to overcome the longer ignition delay of gasoline fuel and helps to premix part of the fuel with air, creating a homogenized mixture. The main injection timing occurs around the premixed combustion spike of the pilot fuel mixture. This serves to control the rate of heat release, reducing combustion noise and increasing combustion controllability. The main injection event results in a diffusion flame, similar to that of diesel combustion. The fuel split between the pilot and the main at this condition is 30% pilot, 70% main, however the split depends on the engine load.

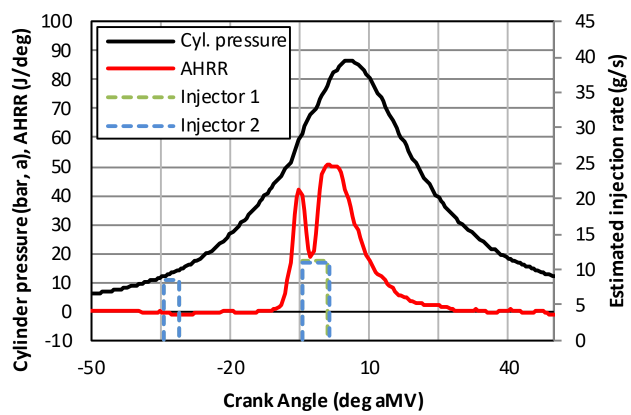


Figure 10: Cylinder pressure, combustion profile, and fuel injection traces at 1275 RPM, 173 Nm

Modal data

Figure 11 shows the preliminary indicated and brake thermal efficiencies across the 10 modes. This preliminary data illustrates the high thermal efficiencies of the OP GCI engine at part load/low-load conditions, which is due to lower heat transfer losses and lower pumping work inherent to the opposed-piston architecture. Figure 12 illustrates the 50% mass fraction burn location for the 10 modes, in degrees after minimum volume. As stated earlier in Figure 3, the 50% mass fraction burn location tends to be earlier and combustion duration tends to be shorter for the OP engine.

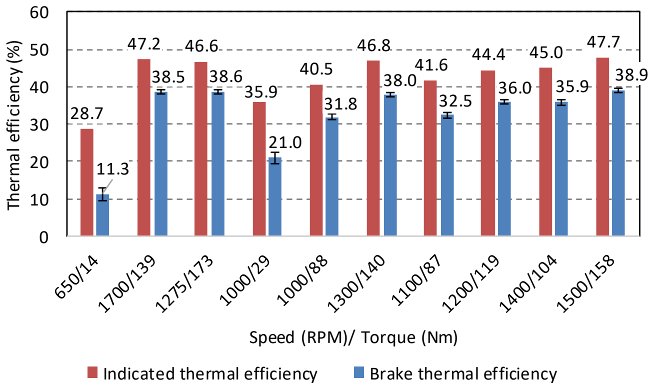


Figure 11: GCI indicated and brake thermal efficiency over 10 modes

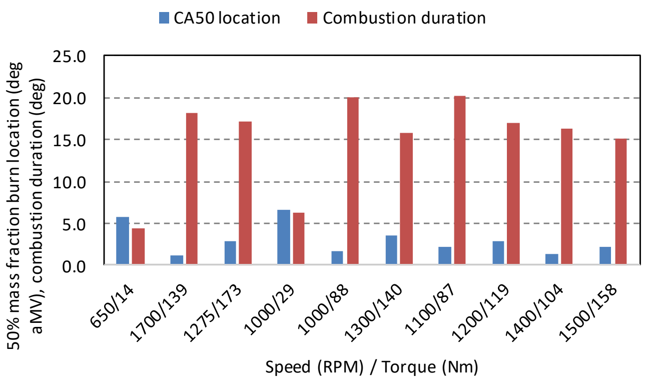


Figure 12: 50% mass fraction burn location and combustion duration

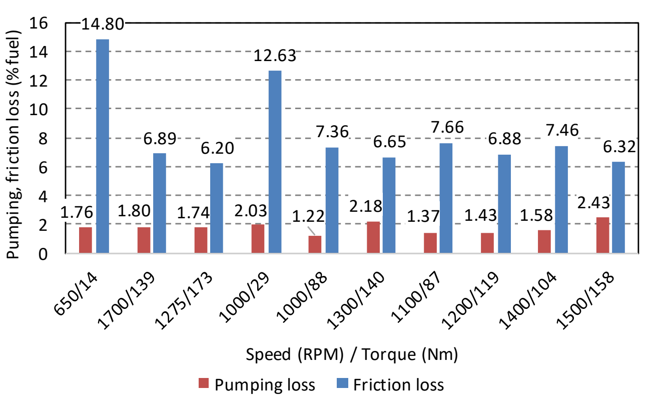


Figure 13: Pumping and friction loss over 10 modes

The pumping loss resulting from supercharger work, Figure 13, while lower for the OP Engine compared to conventional engines, has considerable opportunity for improvement through cylinder ports, turbocharger, and backpressure optimization. As this paper discusses initial results from the new engine, air path optimization is the subject of future work.

Pumping is required to scavenge the cylinder and introduce fresh charge for the next combustion cycle. Two scavenging metrics related to pumping loss are scavenging efficiency (ratio of delivered air mass retained to mass of trapped cylinder charge) and scavenging ratio (ratio of delivered air mass to mass of trapped cylinder charge) and are shown in Figure 14. For most cases, the scavenging efficiency is similar to scavenging ratio. However, when scavenging ratio is greater than scavenging efficiency, fresh charge is escaping the cylinder through the exhaust ports, incurring additional pumping loss.

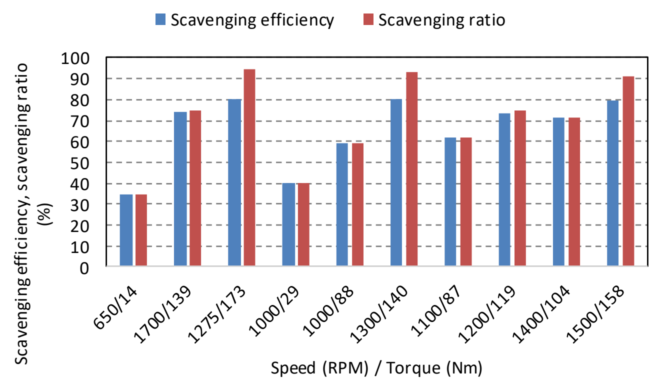


Figure 14: Scavenging efficiency and scavenging ratio over the 10 modal points

The friction loss from the engine is also shown in Figure 13. The new 2.7L engine incorporated several frictional improvements over the research grade Achates Power 4.9L multi-cylinder engine discussed in previous publications [8, 27], however additional friction improvements are still in development. The current friction breakdown is illustrated in Figure 15. Piston rings are identified as the higher contributor to OP Engine friction, followed by the piston skirt and oil pump and are active areas of research.

Combustion noise was well controlled at or below the guidelines from USCAR [28] at all of the points except one, as shown in Figure 16. The ability of the OP Engine to control scavenging, the high flexibility of the fuel injection system, and the high-pressure fuel injection strategy are all key enablers in controlling the pressure rise rate and combustion noise. The high compression ratio enabled by GCI operation enables more favorable autoignition characteristic from increased cylinder pressure and temperature, stretching out combustion slightly compared to lower compression ratio configurations, further reducing combustion noise. Combustion noise is a calibration parameter and can be adjusted to meet relevant requirements.

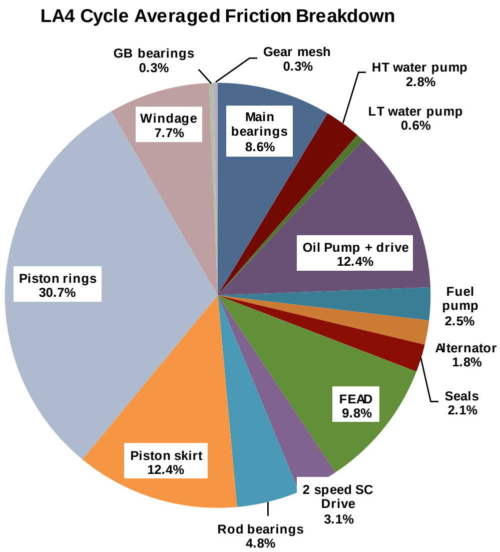
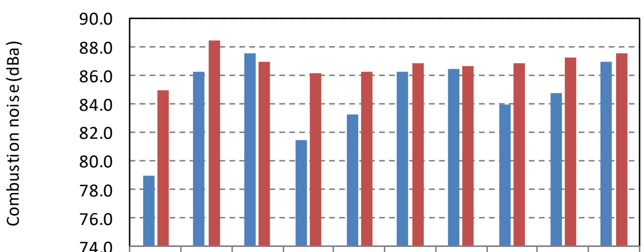


Figure 15: 2.7L OP Engine friction breakdown



	650/14	1700/139	1275/173	1000/29	1000/88	1300/140	1100/87	1200/119	1400/104	1500/158
Noise	79.0	86.3	87.6	81.5	83.2	86.3	86.5	84.0	84.8	87.0
USCAR Limit	85.0	88.5	87.0	86.2	86.3	86.9	86.7	86.9	87.3	87.6

Figure 16: GCI measured combustion noise and USCAR noise limits

The higher compression ratio enabled by GCI operation also achieves high combustion efficiency, yielding gasoline combustion efficiencies that are greater than 98.5% at all points (Figure 17). The combustion efficiencies are very similar to diesel values, however are generated with gasoline fuel. The ability to reduce cylinder scavenging at low-loads, which lowers the pumping work of the engine, also enables high trapped temperatures. The hotter cylinder charge enables better fuel vaporization and higher chemical kinetic rates; leading to more robust, low CoV of IMEP combustion (Figure 18).

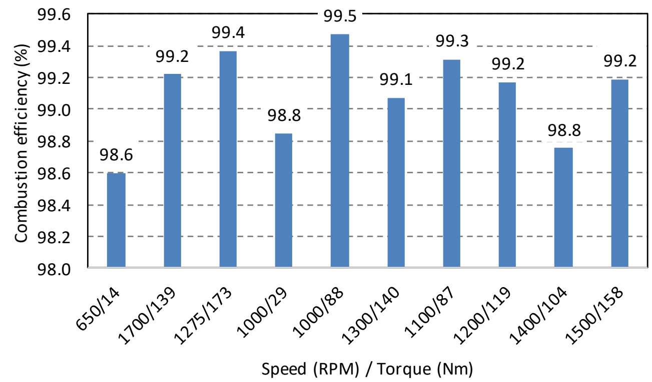


Figure 17: GCI combustion efficiency over 10 modes

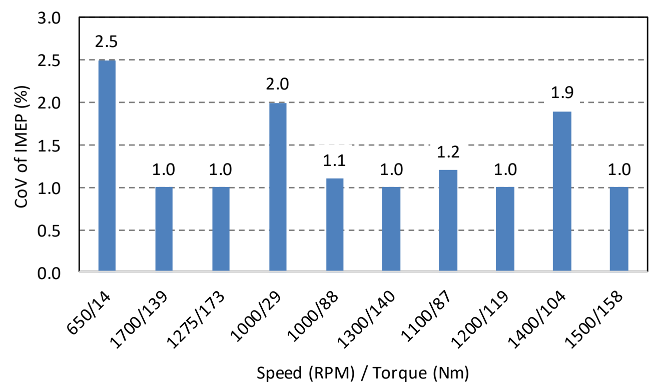


Figure 18: GCI CoV of IMEP over 10 modes

The BSNO$_X$, BSCO, and BSHC values are shown in Figure 19, with BSNO$_X$ as a calibration target. Higher compression ratios tend to increase NO$_X$ emissions, however the combination of lower BMEP operation of the OP Engine combined with lower temperature combustion with gasoline compression ignition keep NO$_X$ formation low. BSCO and BSHC values are low, especially compared to an early injection strategy with GCI [10], due in part to the higher compression ratio of the engine as well as the higher combustion efficiency [29]. Even though higher compression ratios increase NO$_X$ from the higher cylinder temperatures, the decreased HC and CO emissions lowers the overall NMOG + NO$_X$ total emission.

The resulting BSSoot is shown in Figure 20. The partial pre-mixing of the fuel with an early pilot homogenizes the cylinder charge and lower soot formation. The main injection event then controls the rate of heat release and lowers combustion noise as shown previously. The high volatility and partial oxygenation of gasoline fuel promotes better fuel mixing and availability of oxygen, further reducing soot formation, especially during diffusion combustion.

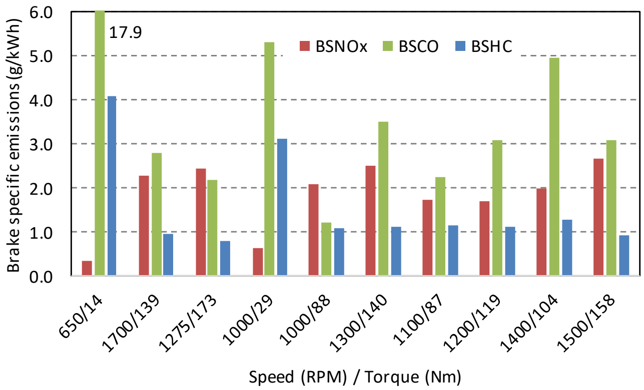


Figure 19: Brake specific NOx, CO, HC over 10 modes

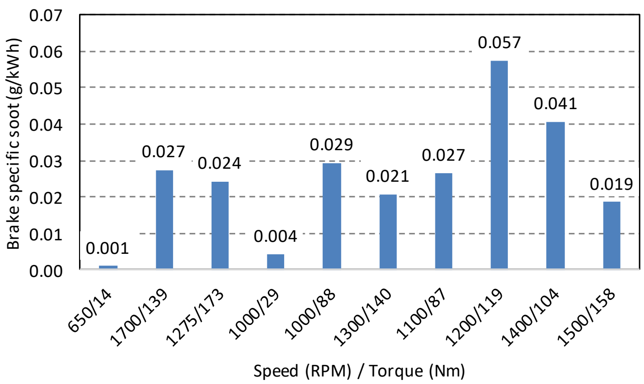


Figure 20: Brake specific soot over 10 modes

<u>Catalyst light-off mode</u>

Previous papers [8, 9] have discussed the unique ability of the OP Engine for rapid after-treatment catalyst light-off and emissions control using diesel fuel. To understand the commercial and emissions potential of GCI, catalyst light-off mode was explored in a single cylinder version of the opposed-piston engine.

Figure 21 highlights the cylinder pressure, rate of heat release, and cumulative heat release representative of an elevated idle condition using gasoline fuel. A similar injection strategy (pilot and main) are used in this condition, however are phased much later in the expansion stroke.

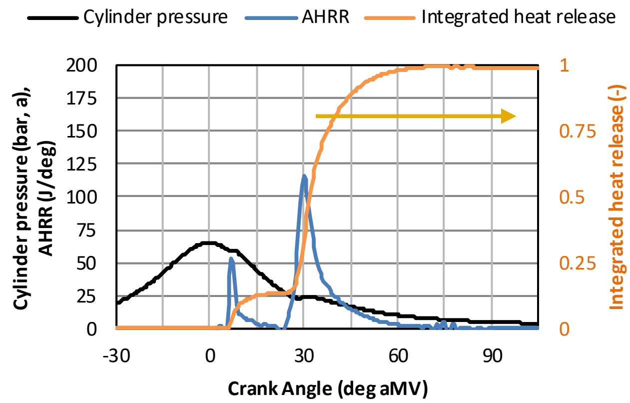


Figure 21: Catalyst light-off mode with gasoline compression ignition

The flexibility of the scavenging and combustion system in the OP engine allows for trapping high temperature residuals, which enables robust and stable gasoline combustion ignition with a 1.1% CoV of IMEP, even with a combustion phasing of 30 degrees after minimum volume. Catalyst light off mode generates high IMEP with low BMEP and results in 365°C exhaust gas temperature while keeping emissions low at 1 g/kWh NO_X and 0.01 g/kWh soot. The hot exhaust gases combined with low emissions during cold start are essential to satisfying stringent emissions requirements.

CONCLUSIONS

A brand new 2.7L multi-cylinder OP Engine was designed and built to integrate into a light-duty pickup truck. The cylinders are tilted 30 degrees from vertical, balanced by the air system on the opposite side and giving the engine the appearance of a Vee shape. The engine uses a high-pressure fuel system capable of generating different rail pressures for the two common rails for combustion flexibility. Engine friction results are encouraging, with piston rings contributing the most, however frictional improvements are an active research area.

Initial results show a cycle average efficiency of 31.7%, which is already greater than 11% conventional engines, after only ten weeks of testing. The final cycle average target of 36.5% is a 30% improvement over conventional four-stroke engines. Combustion noise was well controlled at or below the USCAR limits. For a given NO_X calibration, soot emissions were very low. The cleaner combustion of gasoline fuel enabled the use of higher compression ratio, which increased engine thermal efficiency while reducing low-load CoV of IMEP and combustion noise. The increased compression ratio increased combustion efficiency, reducing HC and CO emissions. Final targets are 36.5% cycle average brake thermal efficiency.

Catalyst light-off mode was explored with GCI. The flexibility of the OP Engine architecture to control scavenging and the controllability of the fuel injection system created stable combustion while generating hot exhaust gas at very low emissions. The combination of hot exhaust gases and low emissions lights off the emissions system quickly, satisfying stringent emissions requirements and enables transition to high efficiency strategies more quickly. After-treatment simulations using initial results and off the shelf diesel components show successful achievement of Tier 3 Bin 160 levels, with an end target of Tier 3 Bin 30.

FUTURE WORK

After only 10 weeks of development, the new 2.7L OP GCI engine is already significantly more efficient than comparable gasoline engines. Considerable efforts are in progress to increase the efficiency from the current 31.7% cycle average BTE to 36.5% BTE with advancements in friction, pumping, and combustion. Friction reduction tasks include reducing piston and liner friction, geartrain windage, and coolant and oil circuits. Tasks related to reducing pumping loss include optimizing scavenging, increasing air handling efficiency, and reducing system restriction. Combustion improvements stem from optimization of the combustion chamber, optimization of fuel injection parameters, and reducing heat transfer losses from the combustion volume. The details of the increase in brake thermal efficiency are proprietary, however a schedule of the anticipated improvements is shown in Figure 22 along with the program target.

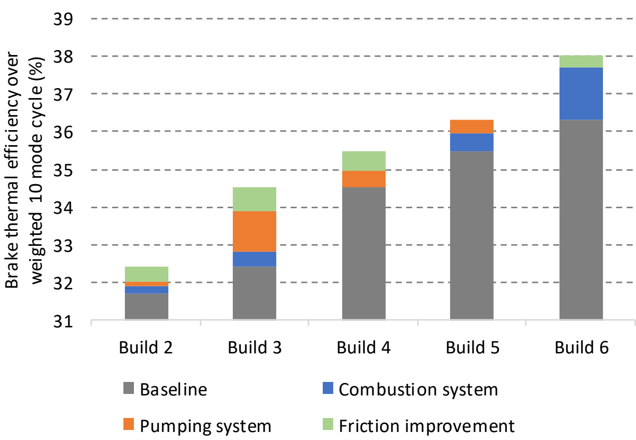


Figure 22: Anticipated cycle brake thermal efficiency improvements with respect to program targets

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