Optimizing Combustion in an Opposed-Piston, Two-Stroke (OP2S) Diesel Engine

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Introduction

First introduced in the late 1800s and once widely used for ground, marine and aviation applications, the opposed-piston, two-stroke (OP2S) engine features an unparalleled combination of power density and fuel efficiency [1] that has yet to be matched by any other powertrain. Despite these advantages, however, OP2S engines—like their two-stroke counterparts—suffered from high soot as well as poor oil control. Unable to meet modern emissions standards, production eventually ceased for OP engines used in on-road applications.

Leveraging today's technologies—such as computational tools, fuel-system advancements (high pressure common rail) and precision manufacturing—the OP engine has been successfully modernized, demonstrating the following when compared to leading, conventional diesel engines:

- 15-24 percent lower cycle-average brake-specific fuel consumption, depending on the application, with similar engine-out emissions levels [2][3]
- Less than 0.1 percent fuel-specific oil consumption [4]
- Reduced cost, weight and complexity due to no cylinder head or valve train

In addition to highlighting the latest Achates Power performance and emissions results, this technical paper will provide a brief overview of the opposed-piston engine's fundamental architectural advantages in:

- Thermodynamics
- Combustion

Also included is a synopsis of the piston thermal management challenges that have affected historical opposed-piston engines and how Achates Power has addressed these challenges by selection of a propriety combustion system to control piston surface temperatures.

For further optimization of the combustion system, with regard to fuel efficiency together with piston thermal management, Achates Power introduced genetic algorithms combined with computational fluid dynamics (CFD).

OP2S Fundamental Advantages

The fundamental advantages of the OP2S have been described in previously published technical papers [2][5].

In addition to these fundamental advantages, the OP2S diesel engine has the following efficiency advantages compared to conventional, four-stroke diesel engines of comparable power and emission standards:

- Lower heat transfer due to favorable surface-area-to-volume ratio
- Higher thermal efficiency due to leaner combustion
- Higher thermal efficiency due to shorter combustion

In the following section, these basic advantages shall be explained in more detail.

Favorable surface-to-volume ratio

The OP engine has favorable surface area/volume ratios during combustion compared to equal displacement four-stroke engines, reducing heat transfer during combustion. Figure 1 shows how to compare a four-stroke to an opposed-piston, two-stroke. Both engines have the same displacement and the same number of pistons. In a thought experiment, an OP2S engine can be envisioned by first eliminating the cylinder head of a six-cylinder, four-stroke engine, then cutting the engine in the middle and flipping it over so that the pistons face each other. The resulting OP2S has a 36 percent reduction in the combustion surface-area-to-volume ratio at the same trapped compression ratio. The right side plot in Figure 1 provides the relationship of surface-area-to-volume ratio to four-stroke engine displacement for OP2S and four-stroke engines. With about a 30 percent[1] lower surface-area-to-volume ratio, the OP2S engine has the heat rejection benefit of a bigger engine with a smaller displacement.

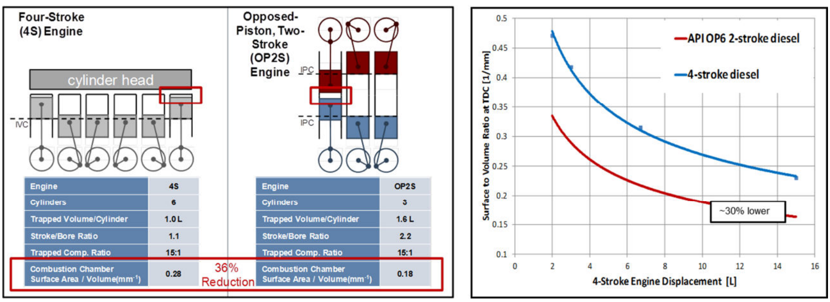


Figure 1: Combustion chamber surface area versus volume comparison between OP2S and conventional four-stroke engines.

[1] 36 percent in the specific example shown.

Leaner Combustion

The two-stroke principle enables a reduction of fuel per combustion event in a larger cylinder for the same power. This results in leaner combustion at the same boost level and, therefore, improves thermal efficiency. This is illustrated in Figure 2 by looking at the last two cylinders of the four-stroke and the far right cylinder of the OP2S. Assuming the same boost condition, the air-fuel ratio of the two-stroke engine is leaner since the trapped volume of the OP2S cylinder is about 60 percent higher. The thermal efficiency of an ideal engine is a function of compression ratio and the ratio of the specific heats. At leaner combustion, the thermal efficiency is higher at the same compression ratio.

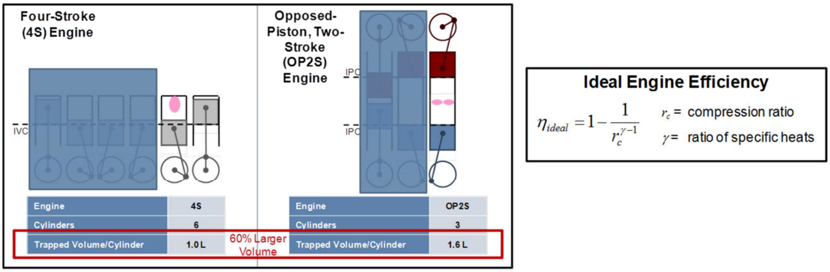
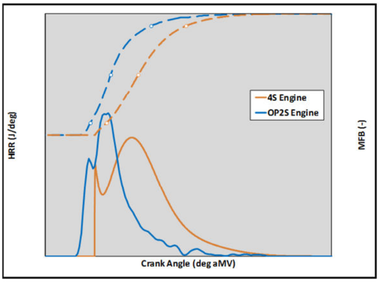


Figure 2: Illustration of leaner combustion in an OP2S.

Shorter Combustion

The larger combustion volume for the given amount of energy released also enables shorter combustion duration while preserving the same maximum pressure rise rate. The faster combustion improves thermal efficiency by reaching closer to constant volume combustion. The first law of thermodynamics is a good way to illustrate this advantage. The plot on the left side of Figure 3 compares a typical four-stroke apparent heat release rate with the heat release rate of an OP2S engine. The OP2S heat release rate has been created by analysis of a measured cylinder pressure trace. The four-stroke heat release rate is generated to get the same power and peak cylinder pressure as the OP2S engine. The OP2S shows shorter and earlier combustion with a higher peak energy release rate. The first law of thermodynamics in a simplified version for four-stroke and OP2S engines can easily explain it. The cylinder volume before combustion starts is about 60 percent bigger for the OP2S. When assuming the same compression end pressure due to the same compression ratio and ignoring the difference in the ratio of specific heat in a first order approach, the apparent heat release rate for the OP2S has to be higher for the same pressure rise rate. Therefore, the energy release can be faster and shorter. Thanks to the reduced heat transfer, the combustion can happen earlier to achieve an increased effective expansion ratio and higher indicated efficiency.



1st Law of Thermodynamics:
4S Engine

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

OP2S Engine

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

Figure 3: Difference in heat release rates between the four-stroke and OP2S engine.

The aforementioned fundamental OP2S thermal efficiency advantages are further amplified by:

- Lower heat loss because the reduced cylinder counts lead to larger cylinders
- Higher wall temperatures of the combustion chamber since piston crowns can sustain higher temperatures than cylinder heads
- Further reduction in the surface area/volume ratio and better scavenging by using a greater than 2.0 stroke-to-bore ratio[2] [5][6]
- Reduced pumping work thanks to uniflow scavenging with the OP architecture also giving a higher effective flow area than comparable four-strokes or a single piston, two-stroke uniflow or loop scavenged engine [7]
- A decoupled pumping process from the piston motion thanks to the two-stroke architecture, which allows alignment of the engine operation with a maximum compressor efficiency line [8]
- Lower NOx characteristics as a result of lower BMEP requirements because of the two-stroke cycle operation [9]

[2] Since two pistons combine to compress the air/fuel mixture, mean piston speeds are still moderate even at stroke-to-bore ratios that exceed 2.0.

Combustion System of the OP2S Engine

Features of the Combustion System

Achates Power has developed many proprietary combustion systems composed of two pistons coming together to form an elongated ellipsoidal combustion volume where the injectors are located at the end of the long axis [10] (Figure 4).

This combustion system allows:

- High turbulence, mixing and air utilization with both swirl and tumble charge motion
- An ellipsoidal combustion chamber resulting in air entrainment into the spray plumes from two sides
- Inter-digitated, mid-cylinder penetration of fuel plumes enabling larger λ=1 iso-surfaces
- Excellent control at lower fuel flow rates because of two small injectors instead of one large one
- Multiple injection events and optimization flexibility with strategies such as injector staggering and rate-shaping [10]

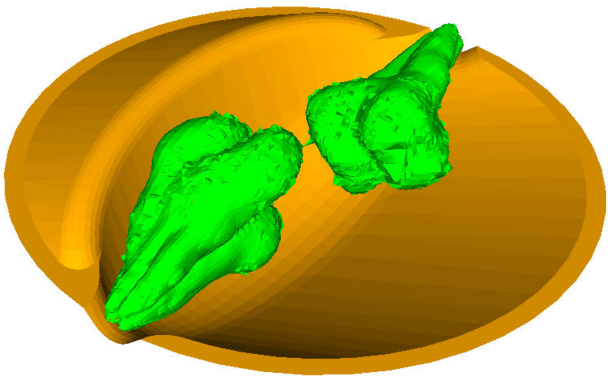


Figure 4: Schematic of the combustion system with plumes coming out of two side-mounted injectors.

The result is no fuel spray impingement on the piston walls and minimal flame-wall interaction during combustion. This improves performance and emissions [2] with fewer hot spots on the piston surfaces, reducing piston thermal loads and increasing engine durability [10].

The shape of the piston crown forces a change in charge motion during the compression stroke. The dominating swirl motion at the end of the scavenging cycle is partially transformed into tumble motion in the elongated combustion chamber. This improves the air entrainment into the spray and enables a short combustion profile.

Figure 5 shows this transformation of swirl into tumble motion during compression. A further factor for good combustion is to have a high turbulent kinetic energy (TKE) at the start of fuel injection, which is a result of the swirl-to-tumble transformation by the piston crown shape.

Since a combustion event occurs in every revolution, the thermal energy input into the piston crown can be significant. The control of the piston crown temperature is an historical challenge of the opposed-piston engine and Achates Power has spent significant development time to solve this challenge.

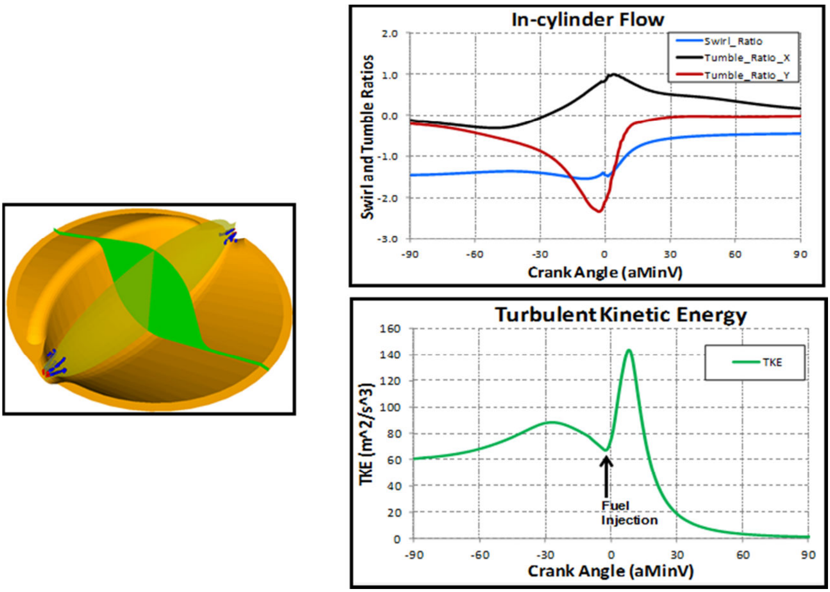


Figure 5: In-cylinder charge motion in an OP2S engine.

Piston Thermal Management

Piston thermal management addresses the main failure modes caused by the combustion heat input into the piston. Those are:

- Piston crown oxidation and erosion
- Ring coking leading to a scuffing condition
- Under-crown oil coking – into runaway condition
- Top land "mushrooming" – growing into liner

It is essential to know the local piston temperature for the development process in regards to piston thermal management. Therefore, a proprietary telemetry system developed in-house is employed to measure temperatures at various locations on the piston. Up to seven locations are monitored per piston in real time, allowing for the study of multiple variables that affect piston temperature. Changes in injector orientation can quickly be evaluated and adjusted to give the optimal temperature profile for combustion, heat transfer and piston durability. The thermocouples are mounted in strategic locations on the piston bowl, such as the bowl rim, injector trench and top ring land. The remote telemetry is achieved by radio frequency (RF). In a signal-conditioning module mounted on the piston, thermocouple (TC) signals are offset by "cold junction" voltage from a sensor and gained up to 0 to 3.0 volts. Amplified and corrected TC signals along with a reference voltage are then time multiplexed and sent to a voltage-to-frequency (V to F) converter. Converter output modulates the transmitter. At the receiver, the demodulated signal is sent to a frequency-to-voltage (F to V) converter. F to V output is de-multiplexed by a micro controller, which also checks the system for proper signal timing. Overall system gain is set using the reference voltage from the signal conditioning module. Individual temperature signals are switched to sample and hold circuits, which then relay the signals to the test cell data acquisition system for display and recording. Power is supplied to the piston-mounted components of the system by inductive coupling within ±6 degrees centigrade (C) from bottom dead center. The data acquisition rate is one data point per second, and the measurements are accurate within ± 3 degrees C.

The thermocouples measure the temperature approximately 2 mm below the combustion surface. A thermal finite element analysis (FEA) with temperature-dependent material properties is used to extrapolate to the surface to get a cycle-averaged surface temperature. This surface temperature must remain below a set limit of 520 degrees C to avoid oxidation and fatigue failure due to compromised material properties at the combustion chamber surface. On the under crown, this temperature must not exceed 285 degrees C to avoid oil coking. The FEA model is validated by comparing against the thermocouple measurements at various locations on the piston. The description of the FEA model and results is outside the scope of this work, but the trends from the thermocouple measurements for different injection patterns are directly compared with predictions from the combustion-CFD model described in the subsequent sections. It is shown that the combustion-CFD results consistently predict the measured trends with respect to injection pattern variation, and help identify the favorable hardware combinations and the governing mechanisms for improved piston thermal management.

The following counter measures were applied to control piston thermal management [11]:

- Appropriate power densities consistent with durability requirements
- Management of the "hot side" – that is, combustion strategy:
    - Piston bowl geometry
    - Injection –spray pattern, number of holes, hole size
    - Calibration – start of injection, duration, air/fuel ratio
    - Liner and manifold – swirl
- Management of the "cold side"
    - Cooling gallery – geometry and fill ratio
    - Oil jets – number and flow rate
    - Oil flow through the connecting rod
- Material selections
    - Crown and skirt
    - Coatings

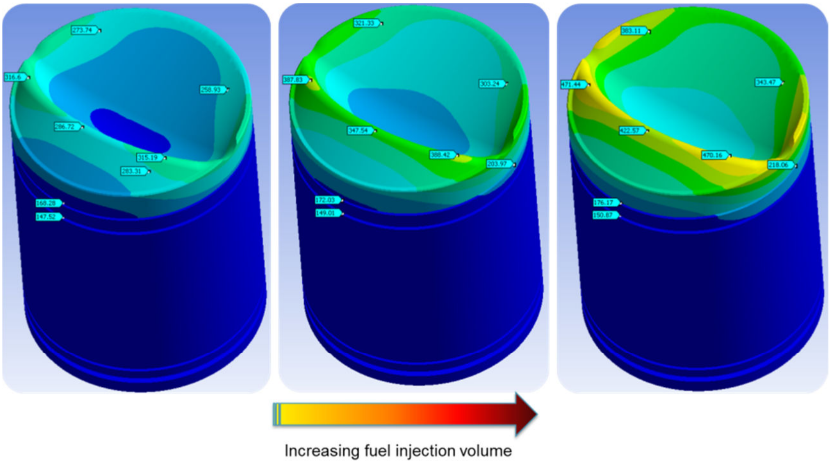


Figure 6: Effect of engine load on piston crown temperatures.

By increasing the engine load, the heat input into the piston crown increases exponentially. The maximum piston crown temperatures are rising at the same time as the heat input. Figure 6 shows the temperature distribution over the piston crown as the result of a FEA simulation at different fuel levels. Due to the shape of the crown the lip, temperature is rising faster than the rest of the surface. Managing the lip temperature is essential for increasing the engine load and power.

The interaction of combustion with the bowl lip area is influenced by the spray pattern and charge motion in the combustion chamber. The pictures in Figure 7 are results from combustion CFD analyses of different spray patterns and show the near-wall fluid temperatures. All these different spray patterns are proprietary to Achates Power. This illustrates the impact of spray shapes on the temperature hot spots on the piston crown surface. Reducing the near-wall fluid temperature at the lip provides an additional margin from the temperature limits.

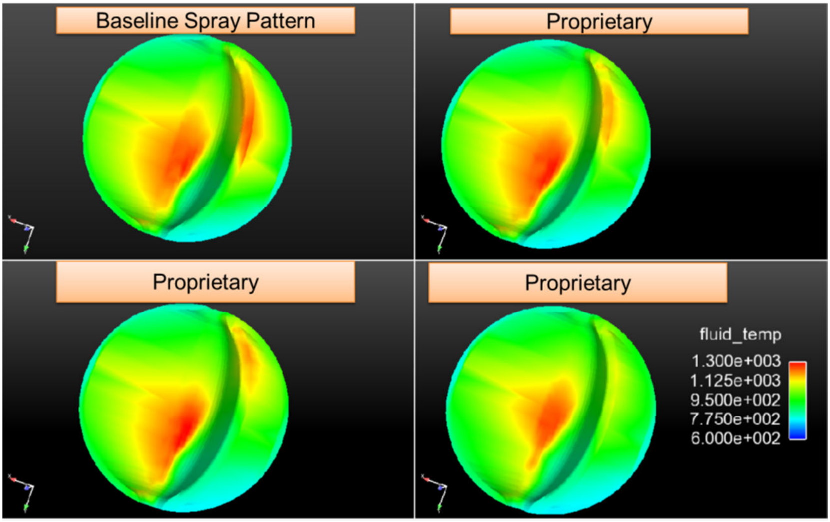


Figure 7: Effect of spray patterns on near-wall fluid temperatures.

The different spray pattern can be built up by changing the spray angle, number of holes and clocking of injectors relative to the piston bowl shape. Change in bowl shape also affects the maximum piston crown temperature. Figure 8 summarizes an experimental investigation of different combustion systems with different piston bowl shapes and their impact on the maximum measured piston temperature on the lip of the piston crown. The plot shows three different load points for three different combustion systems. The calibration for each load point was performed for the same peak cylinder pressure limits and the same engine-out emissions requirements. Combustion system #1 and #2 are reaching at A100 and B100 the maximum piston temperature and a calibration measure had to be put in place to get to the target IMEP without exceeding the piston temperature limits. Of course, the calibration measures decreased the indicated efficiency for those points that ended up in adding additional fuel to reach the target IMEP, which also increases the heat input to the piston and the peak cylinder pressure. C100 combustion system #1 did not get to the target IMEP without exceeding the PCP limit so it was not calibrated to the piston limit.

Combustion system #3 on the other side did not reach the temperature limit at all to achieve target IMEP at the peak cylinder pressure limits. The combustion for system #3 is more advanced than for the other two systems and indicated efficiency is significantly improved by several percentage points.

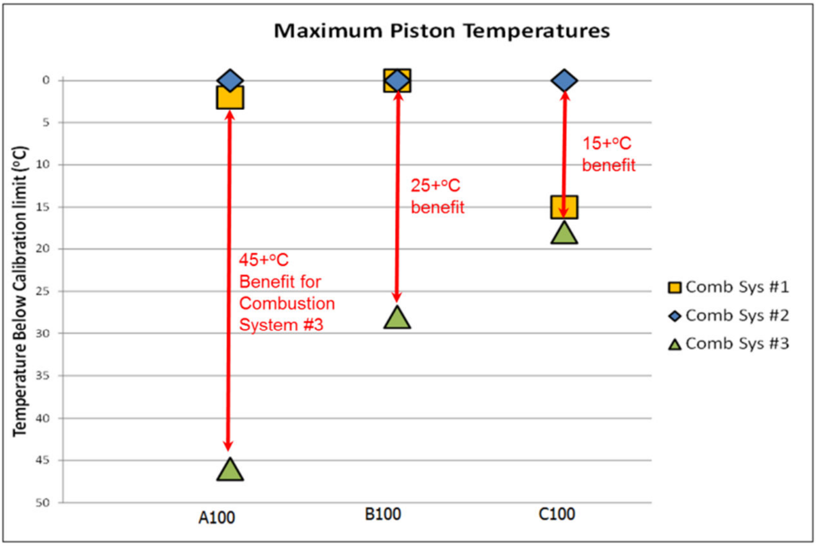


Figure 8: Maximum measured piston crown temperature for different combustion systems.

The choice of the combustion system for the OP2S engine is influenced by the piston temperature limits in order not to be limited in performance.

In addition to the combustion side of the piston crown, there are measurements available for the under crown. The baseline system has a gallery cooled piston with two oil cooling jets per piston to reject the heat out of the hot zone. In order to improve the cooling of the under crown, Achates Power has investigated mechanisms to reduce the material temperature of the pistons further. In Figure 9, there are two add-on cooling measures shown by their reduction of piston temperature at the thermocouples close to the lip of the piston crown. An additional 8 to 10 degrees C reduction in material temperature leads to increased power potential of the engine or enables calibration measures to increase the indicated thermal efficiency further.

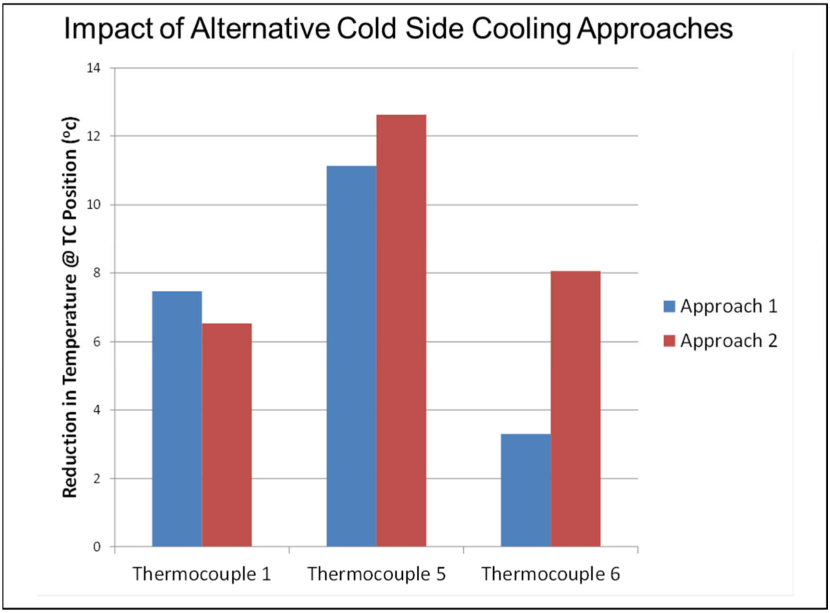


Figure 9: Impact of alternatives on piston under-crown cooling.

Optimizing Spray Patterns with CFD

It is natural to investigate optimization tools to search for the best combination of spray pattern and combustion bowl shapes in order to provide the best indicated thermal efficiency by not exceeding piston temperatures and peak cylinder pressure limits. The Achates Power preferred optimization tool for this kind of optimization is the genetic algorithm method (GA), which is incorporated in chemically reactive CFD. Genetic algorithms work on the "survival of the fittest" principle through tournament selection between individual designs across several generations of runs (Figure 10). GAs are very useful to automatically explore a wide design space and understand trade-offs and response surfaces of key variables. As opposed to traditional DOE methods, GAs explore and fine-tune only the most favorable designs. Only the strongest survive the tournament in each generation and a merit function defines which combinations move on to the next generation. The development of the merit function is dependent on the expected outcome of the optimization process. In the specific case shown in this paper, the goal was to optimize the heat input into the piston crown and improve the indicated efficiency at the same time.

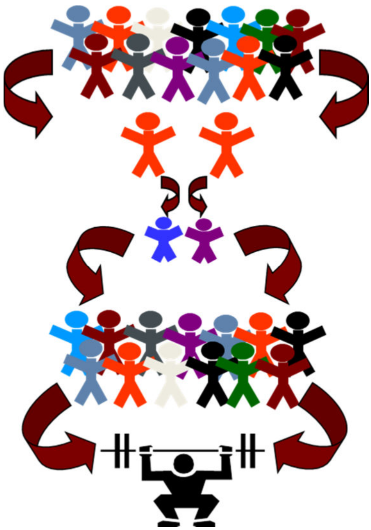


Figure 10: Schematic of the genetic algorithm optimization process.

$$Merit = \underbrace{C_1\,(CCW)}_{\textit{Performance term}} - \underbrace{C_2\left(fn(Piston\_hot\_spot\_area)\right)}_{\textit{Piston thermal management term}}$$

Figure 11: Definition of the merit function for the GA.

The indicated efficiency is represented in the merit function as integral under the resulting pressure curve over volume from the start to the end of the closed cycle. The access to piston temperature in a CFD GA is more complicated. To keep simulation times under control, a conjugate heat transfer analysis for every parameter combination has to be avoided. As a surrogate in the optimization process, the near-wall temperatures at the piston crown have been distributed over bins of different temperature ranges. The amount of near-wall cells in the higher temperature bins can be integrated and expressed as the area of hot spots on the piston surface along critical areas like the lip. This piston hot spot area has been found to correlate very well to the measured piston temperature. This has been shown in previous publications and will be elaborated further in this paper [10]. Any reduction in hot-spot area is directly reflected by a reduction in piston heat input and, therefore, in piston temperature.

In Figure 11, CCW represents closed-cycle (PdV) work computed over the combustion cycle. The piston hot-spot area represents heat input into the pistons due to interaction of the diffusion flame plumes with piston walls. Coefficients have been determined via measured trends for a wide range of operating conditions. Trends from the merit function are validated over a number of conditions using a well-correlated combustion model. The merit function was applied for a GA optimization focused on enhancing performance and thermal management under high power density conditions. Merit function can be modified to include emissions trade-offs (NOx, soot, CO) in addition to performance and piston thermal management.

Independent variable
Hole size for top injector
Hole size for bottom injector
Spray angle for top injector
Spray angle for bottom injector
Number of holes for top injector
Number of holes for bottom injector
Swirl ratio

Each injector of the twin injector configuration is uniquely optimized

- Injected fuel mass from each injector remains constant
- 70 HP/L conditions @ 2400 RPM.
- Piston bowl shape is fixed.
- In-cylinder trapped conditions fixed.

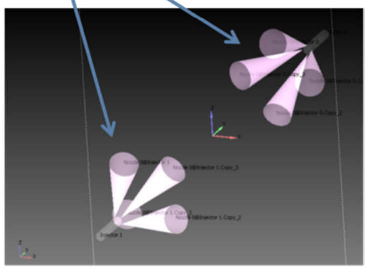


Figure 12: Variation parameters for the spray pattern GA.

As independent variation parameters for the GA study, the number of holes, the hole size and the spray angle for each injector were selected. To get the effect of the charge motion also, the swirl was added as a parameter. Injected fuel mass, piston bowl shape and trapped conditions have been kept constant.

After about thirty generations, the GA converged on an optimized solution that offered a significant reduction in hot spots and, at the same time, increased the closed-cycle work. Figure 13 shows all the results of all the combinations the GA came up with. The x-axis illustrates the difference in closed-cycle work (CCW) and the y-axis illustrates the difference in hot spot areas to the correlated baseline case.

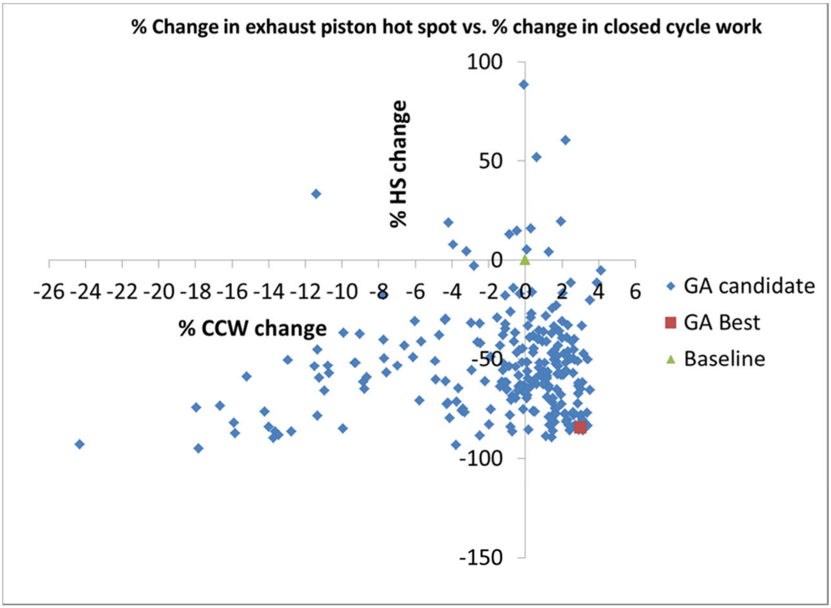


Figure 13: Overview of GA results for CCW and hot spot areas.

The distribution of the points also indicates that the GA did start in every corner of the design space but quickly converged into the bottom right corner, which represents a decrease in hot spot areas and an increase in closed-cycle work at the same time. This result was not expected at the beginning. It was expected that every measure to reduce the hot spot area causes a decrease in indicated efficiency. With the use of the GA in this optimization process, Achates Power has a tool that allows the design of combustion systems for opposed-piston, two-stroke diesel engines of all sizes and applications to meet performance requirements and long-term durability by managing the piston temperatures and the efficiency.

A snapshot during combustion is shown in Figure 14. The left side represents the baseline configuration and the right side shows the GA-optimized spray pattern. The timing is 23 degrees after minimum volume. The color contours show the near-wall fluid temperatures and the iso-surface represents the diffusion flame front. For the baseline case, the combustion front progresses over the piston lip and has increased the near-wall temperature. The optimized solution keeps the flame front away from the lip and, therefore, shows no increased temperature of the fluid in this region.

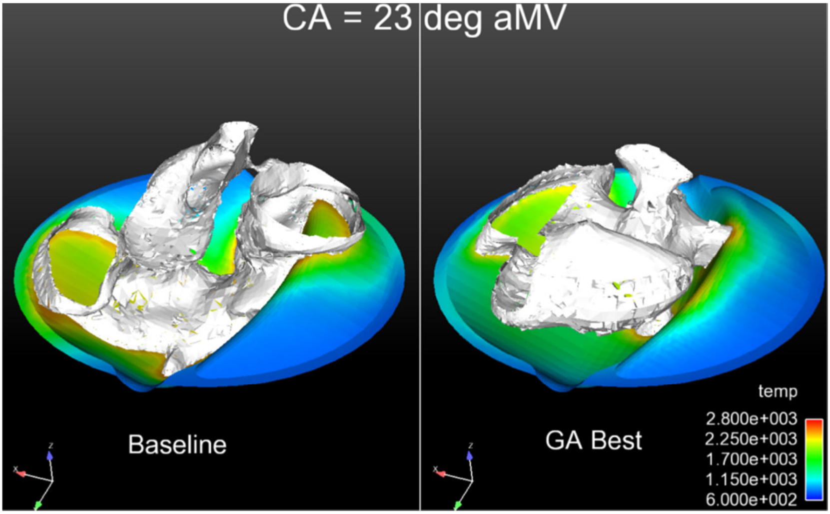


Figure 14: Comparison between GA best and baseline.

Engine Performance Results

The remaining question is what the optimized combustion system does to the overall engine performance. In order to answer this question, an engine needs to first be specified.

Engine Specifications

For the best implementation of all the inherited advantages of the opposed-piston, two-stroke diesel engine, the number of cylinders was selected to be three [12]. The baseline power cylinder has a 1.6 L displacement and a stroke-to-bore ratio of 2.2. The bore diameter is 98.425 mm. These are the dimensions of the single-cylinder prototype Achates Power has tested since 2012. All data in this paper are based on measurements from this single-cylinder engine.

Table 1: Medium-Duty Engine Specifications

Dimensions	
Bore	98.425 mm
Stroke	215.9 mm
Stroke-to-bore ratio	2.2
Displacement per cylinder	1.64 L
Number of cylinders	3
Engine displacement	4.9 L
Performance Specifications	
Rated power	205 kW
Rated power speed	2200 rpm
Maximum torque	1100 Nm
Max. torque speed range	1200-1600 rpm
Max. BMEP (2S engine)	14 bar
Emission legislation	US2010/ EURO 6
Aftertreatment	DOC/DPF/SCR
Engine-out NOx (cycle avg.)	5 g/kWh
Engine-out soot (cycle avg.)	0.03 g/kWh

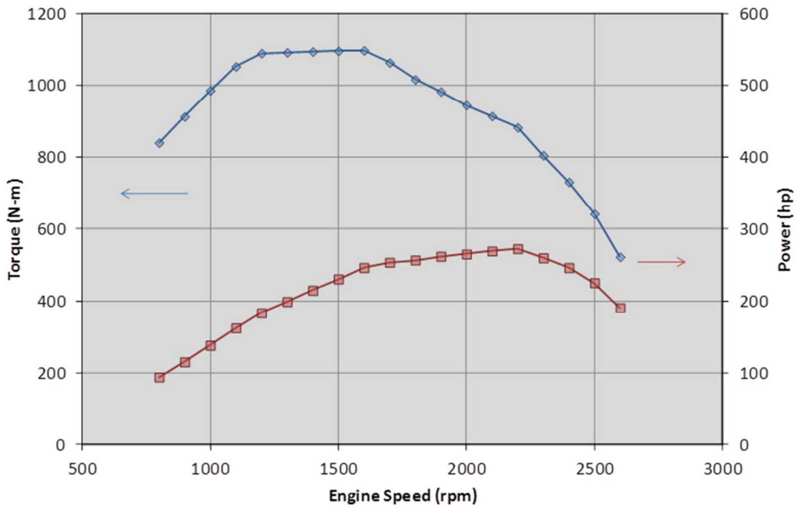


Figure 15: Torque and power curve for the 4.9 L three-cylinder engine.

Table 1 summarizes the main dimensions and performance specifications. The target application is for medium-duty commercial trucks. The power and torque requirement is suited for this market segment. The engine-out NOx target is where the industry is heading in the near future, since the SCR system performance has improved significantly. Figure 15 shows the torque and power curves versus the engine speed.

Benchmark Engine

Finding a benchmark engine comes down to the availability of public data. In 2010, Ford Motor Co. published their latest diesel engine product for full size pick-up trucks above 8500 lbs. GVW. Figure 16 shows a picture of the engine and the published BSFC and BSNOx maps. The engine is a 6.7 L V8 diesel engine with 300 hp rated power and 1000 Nm maximum torque.

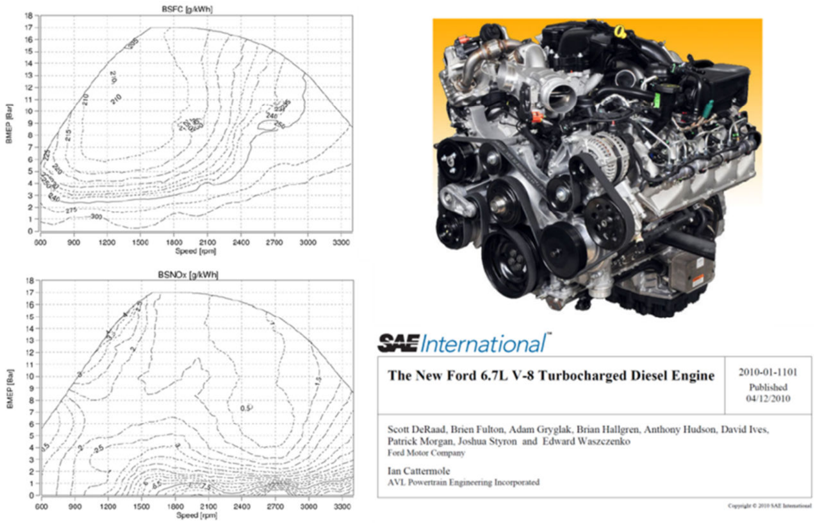


Figure 16: Benchmark engine BSFC and BSNOx map.

BSFC Map of the OP2S Diesel Engine

The BSFC map was created via a GT-Power model. The following assumptions have been made:

Combustion:

- Rate of heat release profiles have been measured on the single-cylinder engine using an optimized combustion system to specify piston temperature limits.
- Peak cylinder pressure limit: 200 bar

Air System:

- The air system consists of a fixed geometry turbocharger, a first CAC downstream turbo compressor, a supercharger, a second CAC downstream supercharger, an EGR system with pick-up upstream from the turbine, EGR coolers and mixing with the air stream upstream of the supercharger.
- Turbocharger maps are provided from a supplier for a prototype device under development showing high efficiencies for the compressor and turbine.
- A supercharger map was provided by a supplier. The map was adjusted to allow room for future development in efficiency by optimizing the performance to the needs of the opposed-piston, two-stroke diesel engine.
- CAC pressure drop and effectiveness are based on current production units.
- Exhaust back pressure counts for the DOC/DPF/SCR aftertreatment system and is derived from transient measurements.

Friction Model

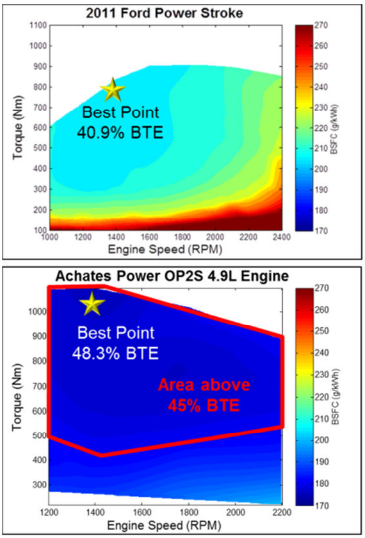
- Mechanical friction is based on measurements on current hardware and extrapolated for three cylinders. The engine is under the assumption of a clean-sheet design, incorporating the future friction improvement potential.
- Next-generation common rail fuel system with extra-low leakage, high-efficient high pressure pump and 250 MPa maximum rail pressure. The parasitic losses are mapped into the GT-Power model.
- Supercharger transmission mechanical efficiency of 97 percent, according to supplier information

Scavenging

- Achates Power has developed a method to measure scavenging performance by measuring in-cylinder CO_2 concentrations before and after combustion.
- The measured scavenging efficiencies match the CFD correlation of measured single-cylinder load points very well.

- Also three-cylinder, open-cycle CFD simulations for comparable load points show matching scavenging efficiencies. The GT-Power model uses this open-cycle scavenging result as model input.

Figure 17 shows the BSFC map generated in the lower left corner. The best brake thermal efficiency is 48.3 percent at around 1400 rpm and 1000 Nm. Compared to the Ford engine, this is an increase of more than seven percentage points of efficiency. But, more important than the increase in brake thermal efficiency is the flat fuel map. A huge portion of the map is above a brake thermal efficiency of 45 percent. The table compares the 12 power modes from the 13-mode steady-state test cycle used for emissions certification. Averaging the BSFC over this cycle with the emission-weighting factors leads to a 24 percent BSFC reduction compared to the V8 engine.



Operating Condition	API MD 2017 BSFC (g/kWh)	Ford 6.7L V8 BSFC (g/kWh)	Difference %
A25	186	230	-19%
A50	178	217	-18%
A75	178	208	-15%
A100	175	211	-17%
B25	192	265	-27%
B50	180	240	-25%
B75	177	230	-23%
B100	180	230	-22%
C25	200	330	-39%
C50	184	270	-32%
C75	179	252	-29%
C100	180	252	-28%
Weighted Average	**180**	**238**	**-24%**

24% lower cycle average BSFC

Figure 17: BSFC comparison with benchmark engine.

The flat BSFC map enables:

- Better in-use fuel economy
- Less driver-to-driver variation
- Fewer number of shifts and/or simpler, less expensive transmissions
- No need for downspeeding and associated drivetrain upgrades

- Less application-specific development effort
- Further improvement is possible by applying new technologies like waste-heat recovery

Summary

The opposed-piston, two-stroke engine has fundamental advantages in producing low emissions and low fuel consumption at low manufacturing costs [3]. The performance and emissions of the architecture has been optimized and the mechanical design and durability challenges have been solved with analytical tools and innovative approaches. The OP2S engines are being developed for light-, medium- and heavy-duty vehicles for automotive applications as well as for the military. The technology is also being readied for generator sets and large ship engines.

From the beginning, Achates Power has addressed the historical challenge with piston heat load that applies to all two-stroke engines. The piston thermal management covers the heat input due to combustion on the piston crown as well as the rejection of the heat from the under crown of the piston to the cooling oil. Optimizing the combustion system under consideration of piston temperature limits is essential in the development process. Complex optimization tools like genetic algorithms—in combination with combustion CFD simulations—allow the reduction of the heat input into the piston with an increase in indicated efficiency at the same time.

The latest results show a brake thermal efficiency of 48.3 percent for the best point for a medium-duty engine. The performance is based on prototype parts of turbochargers, superchargers and today’s production parts of the air system and aftertreatment components. Compared to a typical four-stroke commercial vehicle engine, the efficiency islands of the engine map for the Achates Power opposed-piston, two-stroke diesel engine are very flat over a substantial region of the map. This enables the optimization of the drivetrain using existing components for transmissions, rear axles, etc.

The Achates Power opposed-piston, two-stroke diesel engine demonstrates significantly higher brake thermal efficiencies than comparable four-stroke diesel engines with the same power. Implementing unique and proprietary design solutions illustrates not only the efficiency potential, but also the durability and low oil consumption needed for a successful future generation of commercial vehicle engines.

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