MODERNIZING THE OPPOSED-PISTON, TWO-STROKE DIESEL ENGINE FOR MORE EFFICIENT COMMERCIAL VEHICLE APPLICATIONS

Dr. Gerhard Regner
Director, Applications Engineering
Achates Power, Inc.
4060 Sorrento Valley Boulevard
San Diego, CA 92121

Co-Authors:

John Koszewnik	Fabien Redon	Eric Dion
Chief Technical Officer	VP, Performance & Emissions	VP, Mechanical Systems

INTRODUCTION

Opposed-piston engines have a long history, starting with the Junkers Jumo 205 [1] diesel aviation engine from the 1930s and continuing today in marine diesel engines. Each cylinder has two facing pistons that come together at top dead center and move outward upon combustion. Since the pistons cyclically expose or occlude the exhaust and intake ports, there is no valve train. Camshafts, pushrods, rocker arms, valves, valve springs, valve keepers, etc., are not required. Because the intake and exhaust ports are at the opposite ends of the cylinder, opposed-piston engines have efficient, uniflow scavenging.

Opposed-piston (OP) engines, when properly sized and configured, have inherent thermodynamic advantages [2]. Due to the high thermal efficiency and lack of cylinder heads, OP engines have lower heat rejection to coolant. Moreover, as a two-stroke engine, they have inherently high specific power output. By eliminating cylinder heads and the valve train, OP engines cost less than conventional four-stroke engines.

Achates Power was formed in 2004 to modernize the opposed-piston engine, and has demonstrated breakthroughs in combustion and thermal efficiency, achieved through more than 3,000 hours of dynamometer testing. While much of the development by Achates Power is directed at commercial and passenger vehicle markets, high thermal efficiency, high specific power and low heat rejection make the Achates Power opposed-piston engine ideally suited for all kinds of applications.

DESIGN ATTRIBUTES

Several design attributes of the Achates Power opposed-piston, two-stroke engine optimize its performance.

Opposed-Piston Architecture

The advantages of opposed-piston engines are described in the book, Opposed Piston Engines: Evolution, Use, and Future Applications [3]:

"OP engines evolved because of their ease of manufacture, excellent balance, and competitive performance and fuel-efficiency relative to comparable four-stroke engines….With the progressive development of OP engines…other significant advantages also emerged…Among these advantages were cutting-edge specific output, high specific torque, very high power density, and very high power-to-bulk ratio….Other OP two-stroke advantages, compared to the four-stroke engine, were relatively low heat-to-coolant ratios, high reliability and low maintenance, relative ease of servicing, excellent multi-fuel tolerance, low injection pressures, and simple fuel injection nozzles."

The source and magnitude of the thermal efficiency advantage has been described by Achates Power [2]. Eliminating the cylinder heads has two advantages. The surface area/volume ratio during combustion is reduced compared to conventional engines, decreasing heat transfer, which increases indicated thermal efficiency. And since cylinder heads are cooled to lower maximum temperatures than pistons, the heat transfer benefits of the opposed-piston design are even greater. Also, the stroke is split between two pistons, enabling high stroke/bore ratios without excessive mean piston speeds. Since intake and exhaust ports are on opposite sides of the cylinder, efficient uniflow scavenging is utilized. Lastly, the convergence of two piston crowns prior to combustion in lieu of a conventional cylinder head/piston arrangement enables a uniquely shaped clearance volume with excellent charge motion and very rapid burn duration.

Two-Stroke Operation

Two-stroke engines have fuel efficiency advantages compared to four-stroke engines [2]. The amount of fuel per cylinder displacement injected for each combustion event can be reduced to roughly half for the same power as a four-stroke engine. This provides two thermal efficiency advantages for the two-stroke engine:

- A leaner operating condition at the same boost pressure maintains a higher ratio of specific heats during combustion.
- A reduced energy release density at the same power level allows for shorter combustion duration without exceeding a maximum rate of pressure rise constraints.

Moreover, as a two-stroke, the Achates Power engine allows for optimized pumping losses by only partially scavenging the exhaust from the combustion chamber. This is accomplished by controlling the supercharger, a feature that avoids the complex and expensive design of the variable valve train mechanisms used for the same purpose in four-stroke engines.

The double firing frequency also provides specific power advantages. Since each cylinder fires every revolution in a two-stroke design, engine displacement can be reduced for the same power as a four-stroke engine without exceeding peak cylinder pressures or other design constraints.

Power Cylinder System

The heart of the engine is the power cylinder system. It includes the pistons, cylinders, cylinder liners, port designs and fuel system. It has been designed, manufactured, tested and refined through iterative versions on a series of single-cylinder test engines to prove the performance and emissions capability as well as the durability of the engine.

Using computational fluid dynamic studies correlated with single-cylinder engine test results, a very clean and efficient combustion system for opposed-piston engines has been developed. Of critical importance, the design results in large stoichiometric iso-surfaces with excellent mixing and charge motion at the point of auto-ignition. These combustion system attributes are accomplished with a unique set of intake ports that provide swirl, coupled with mating converging-crown pistons that induce tumble motion at top dead center. Additionally, a unique and proprietary nozzle design provides interdigitated fuel plumes with the appropriate flow rates and penetration. Together, this hardware combination provides for very fast burn rates, contributing to both power and fuel efficiency.

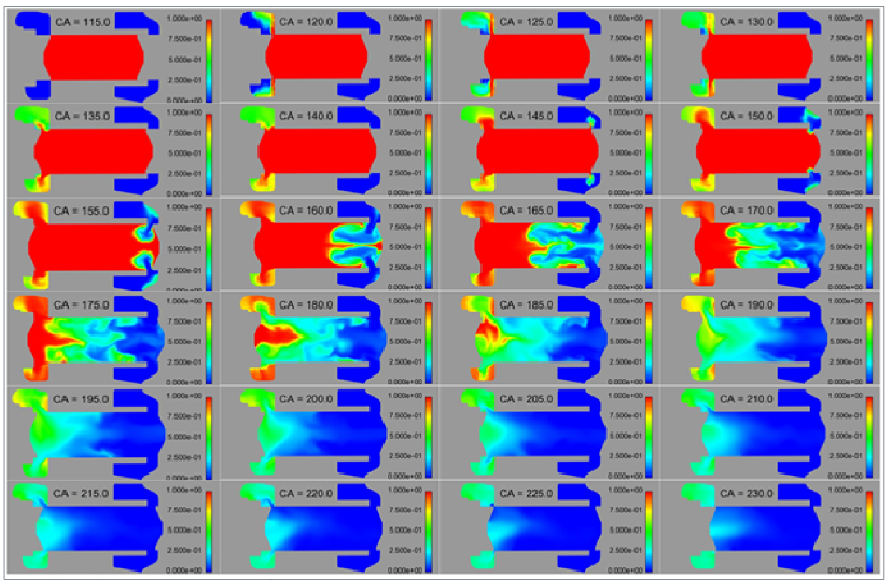

Figure 1: Full Load CFD Simulation Results of In-cylinder Gas Exchange

Lastly, port timing has been optimized over multiple CFD and physical testing iterations of the cylinder liner to provide the optimal blow-down, uniflow scavenging and supercharging characteristics. An example of CFD simulation results is shown in Figure 1. In the first frame, in the upper left, the expansion cycle after combustion is well underway. In the next frame, to the right, the exhaust (left side) ports open as the pistons continue their outward travel, and blow down begins. During blow down, the intake is closed so only exhaust gas leaves the cylinder via the exhaust ports. The intake ports in this example open at a crank angle of 145° ATDC. The pressure in the cylinder falls below the pressure level of the intake manifold and fresh charge starts to enter the cylinder. This is the start of the scavenging phase where intake and exhaust ports are open at the same time. Scavenging continues until a 230° crank angle, at which time the exhaust ports close. The final frame in the diagram shows a small amount of residual gas left in the cylinder. Careful design and analysis of the ports, manifolds and gas exchange system balances good scavenging efficiency while minimizing pumping losses.

In addition to serving as the foundation for all combustion development, the single-cylinder test engine was also used to validate the durability of the power cylinder components. This includes successful completion of a 50-hour full load durability test, which further demonstrates the mechanical integrity of the power cylinder system.

Cranktrain System

Several mechanical design arrangements of crankshafts and connecting rods to articulate the pistons have been analyzed, developed and tested. The dual-crankshaft, Junkers Jumo-style arrangement has several advantages. A cut-away of this arrangement and a 3-cylinder representation is shown in Figure 2. This mechanical design is referred to by Achates Power as the "A48" design, and its advantages include robustness, compactness and mechanical simplicity.

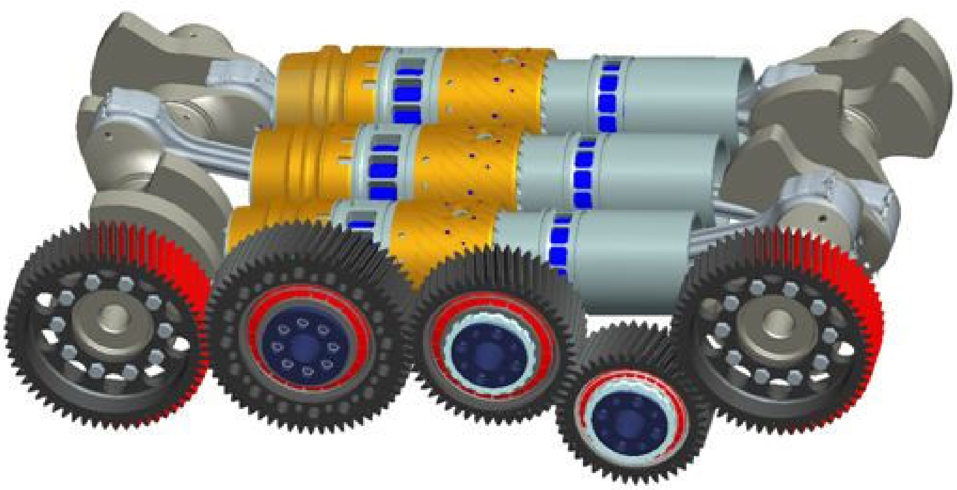

Figure 2: Cutaway of Power Module Concept, Showing Internal Gear Train

Most of the successful opposed-piston engines throughout history utilize the dual-crankshaft architecture that is similar to the A48, including Junkers Jumo 205 and 207, Fairbanks Morse 38D, Rolls Royce K60, Leyland L60, Kharkov Morozov and Coventry Climax H30. A number of improvements have been made to the architecture to address concerns about piston thermal management, cylinder thermal management, wrist pin durability and piston ring durability. These improvements have been tested and are discussed in more detail in the section on design solutions and durability demonstration.

The virtues of the Achates Power A48 engine architecture include:

Conventional crank-slider mechanism

The A48 architecture uses conventionally designed crankshafts and connecting rods, which are commonplace in the industry. It does not require any unconventional and unproven engine mechanisms.

Integrated internal gear train

The A48 mechanism has an internal gear train, which allows the flexibility to have the power taken off of any crankshaft or idler gear. A depiction of a generic gear train is shown in Figure 2. The engine characteristics as seen by the transmission input can be varied by altering the gear ratios of the internal gear train if power is taken off one of the idler gears. If an application requires high torque, a drive ratio can be chosen to reduce speed of the engine output shaft. Other applications might benefit from increased output speed at reduced torque to drive, for example, a high-speed generator in hybrid drive applications. The engine can serve a wide range of applications from a core engine design by altering the gear ratio of the internal gear train.

Compact shape

Compared to other opposed-piston architectures, the A48 design does not suffer from excessive width or volume, yet still maintains a thermally-efficient stroke/bore ratio. Indeed, the compact flat shape of the A48 architecture (similar to the familiar boxer-style engine) lends itself to enhanced packaging options. Moreover, because of the compact size of the architecture, A48 variants have been designed that package into existing passenger and commercial vehicles.

Optimal Stroke-to-Bore Ratio

Extensive analysis has been performed to determine the optimal stroke/bore ratio needed to maximize engine thermal efficiency. There are three main effects to consider:

- In-cylinder heat transfer decreases as the stroke/bore ratio increases due to a decreased combustion chamber surface area/volume ratio during combustion. The decreased heat transfer directly leads to higher indicated thermal efficiency and reduced heat rejection to the coolant. Figure 3 shows simulated indicated thermal efficiency loss calculated at the same operating conditions for a range of cylinder stroke/bore ratios. The effect is non-linear as the change in indicated thermal efficiency gets progressively worse as the stroke/bore ratio decreases.

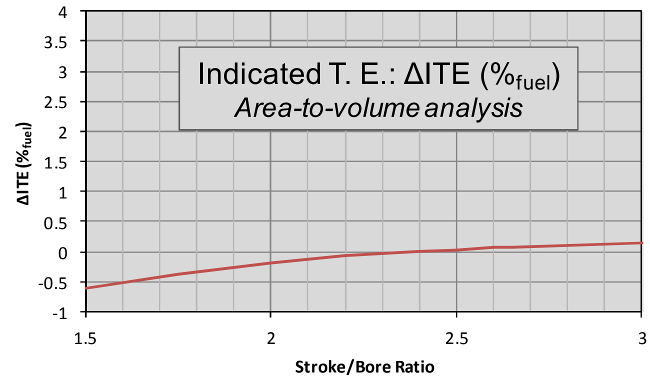

Figure 3: Effect of Stroke/Bore Ratio on Indicated Thermal Efficiency

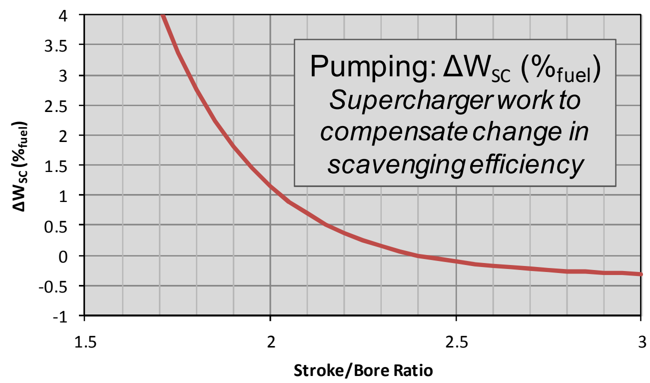

Figure 4: Effect of Stroke/Bore Ratio on Pumping Work

- The scavenging efficiency increases as the stroke/bore ratio is increased. To quantify the impact this effect has on engine efficiency, Achates Power combined 3D CFD and 1D simulations to calculate the change in pumping work (measured as a percent of fuel energy) required to maintain a constant scavenging efficiency with changes in the stroke/bore ratio. Results of these simulations are provided in Figure 4, which shows the change in pumping work relative to that obtained with a stroke/bore ratio of 2.4. The pumping work increases rapidly after the stroke/bore ratio decreases below 2.2. These analytical results have also been confirmed with scavenging studies conducted on test engines with varying S/B ratios.

- Engine friction was found to have a non-linear dependence on stroke/bore ratio, as displayed in Figure 5. Using a published friction model [4], Achates Power found that for a fixed PCP, the crankshaft bearing friction decreases as the stroke/bore ratio increases, while the power-cylinder friction has the opposite effect. The net effect is that the friction increases when the stroke/bore ratio exceeds a value of about 2.3, although the magnitude of the effect is much smaller than the heat transfer and pumping effects.

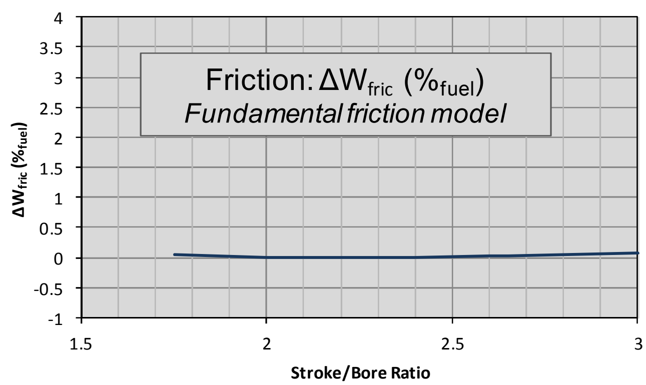

Figure 5: Effect of Stroke/Bore Ratio on Friction Work

Combining these factors, indicated thermal efficiency and pumping work benefit from a longer stroke/bore ratio. Friction work decreases until a stroke/bore value of about 2.3 and then increases as the stroke/bore is increased further. Any opposed-piston engine with a stroke/bore below 2.0 will be compromised from an efficiency and heat rejection-to-coolant basis. Figure 6 shows the combined change in brake thermal efficiency over stroke to bore ratio.

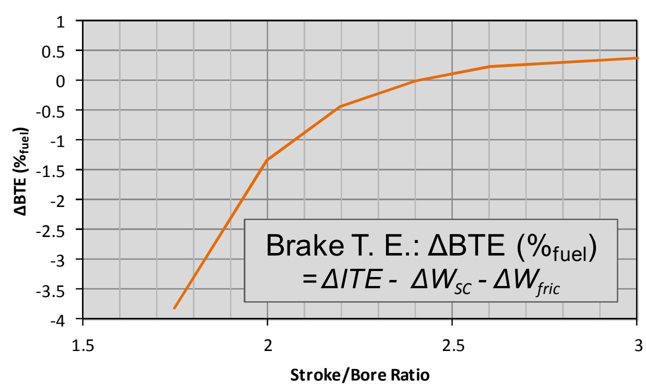

Figure 6: Combined Impact of Stroke/Bore Ratio on BTE

In general this curve is valid for all 2-stroke engines not only the opposed-piston 2-stroke engine. Conventional 2-stroke engines (with one piston per cylinder) are usually designed with stroke/bore ratio below 1.5 to avoid excessive piston speed. Any theoretical advantages of the 2-stroke combustion process are overpowered by scavenging and heat transfer losses due to the low stroke/bore ratio.

Three-Cylinder Configuration

Extensive analysis has determined that a 3-cylinder, opposed-piston engine is optimal from a gas exchange perspective compared to 2-cylinder or 4-cylinder versions, primarily due to gas dynamic effects. Figure 7 shows the intake and exhaust port open areas versus crank angle for 2-cylinder, 3-cylinder and 4-cylinder engines. The gas exchange duration in a two-stroke uniflow scavenged engine is about 120° crank angle. In a three-cylinder design, the scavenging events are aligned in a way that they have minimal interference with each other and still keep enough mass flow going over the cycle to provide adequate energy to the turbocharger so it operates most efficiently to compress the intake air.

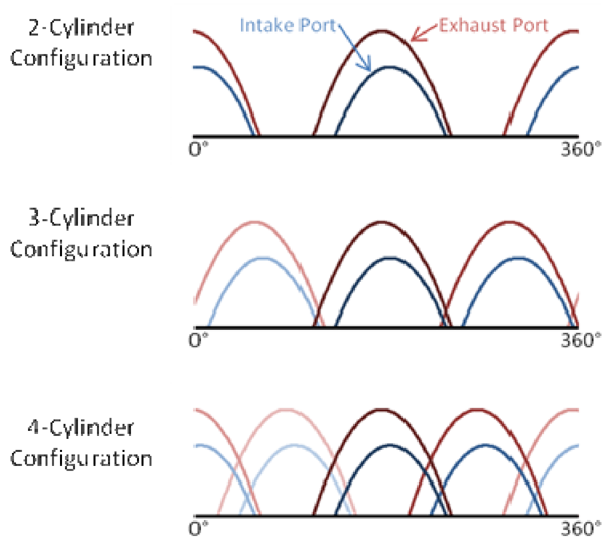

Figure 7: Cross-charging in a 2-stroke Engine

In a two-cylinder configuration, however, the gas-exchange events are too far separated in time. This separation causes the turbocharger to lose energy over the cycle, which has a negative effect on the turbine's efficiency—especially at lower loads and engine speeds. The loss on turbocharger energy has to be compensated by the crank-driven supercharger, which causes a reduction in brake thermal efficiency.

Conversely, in a four-cylinder configuration, the gas-exchange events overlap too much. This causes cross charging to occur at a point in time when hot exhaust gases are leaving the cylinder. The interruption of exhaust gas flow causes an increase in residual gas content and, therefore, a lower scavenging efficiency—leading to a reduction in thermal efficiency. Even with a complex design of the exhaust manifold to separate the pulses, there will still be gas communication even with a twin scroll turbine housing. Separating the exhaust system into two turbochargers leads back to the two-cylinder problem with the energy flow interruption over the cycle.

Two-, 4-, and 5- cylinder options are all viable as part of a family of engines—and indeed speak to the modularity of the design—but a 3-cylinder design is optimal.

Air-Handling System

Figure 8 shows the air handling system layout for a commercial vehicle engine. Both a current production supercharger and fixed geometry turbocharger are used. The turbocharger utilizes exhaust energy to compress air downstream of the compressor wheel. The supercharger is required to provide the initial airflow used for starting, and the supercharger and its associated recirculation valve are used to adjust boost pressure in the intake manifold to manage the gas exchange process while minimizing pumping losses. The supercharger drive system is optimized to maintain high thermal efficiency over the entire engine map, increase low-speed torque, and enhance the cold-start capability of the engine. In addition, the supercharger improves transient performance and acts as an EGR pump for efficient exhaust gas recirculation.

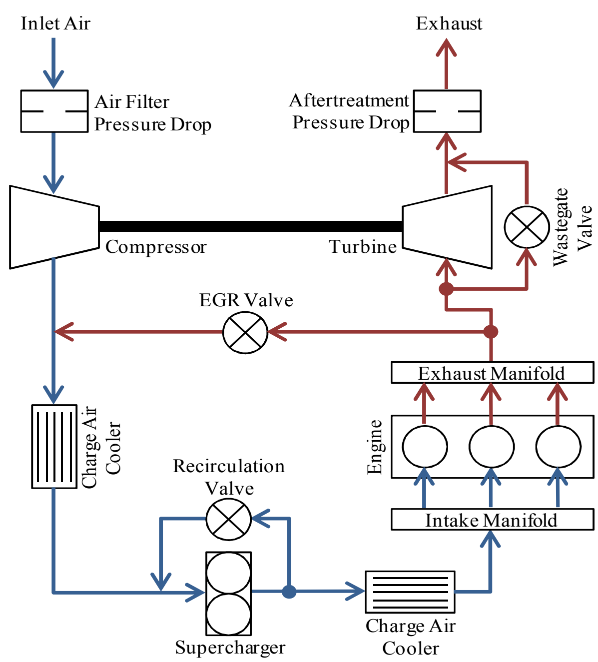

Figure 8: Air Handling System Layout

Fuel Injection

The single-cylinder test engine has a 2000 bar capable, common-rail, fuel-injection system using production Bosch injectors. Unique nozzles were created to account for the different combustion chamber geometry inherent in the opposed-piston engine. Each cylinder contains two injectors facing each other. This proprietary design has been extensively tested.

Higher fuel rail pressures and injectors are being studied to assess whether the atomization improvements result in a favorable fuel efficiency trade-off versus the additional parasitic losses.

Cooling System

The consequence of high specific power is higher thermal loading into the engine components, especially the pistons. In order to cope with the heat going into the pistons, a gallery-cooled design is used, with dual oil cooling jets per piston providing the necessary oil flow to maintain acceptable piston temperatures. The gallery cooling has been developed and tested on the single-cylinder research engine.

The cylinder liner contains a water jacket to reject the combustion heat out of the liner. A patented cylinder liner design has been developed in order to manage the high thermal loading at the center of the liner and at the exhaust port bridges. The cooling arrangement for the liner has been developed and tested on the single-cylinder research engine.

Summary of Architectural and Design Advantages

The Achates Power opposed-piston engine has been designed to meet emissions and durability requirements while minimizing fuel consumption. The architectural and design advantages, which combined make for a superior solution in a commercial vehicle engine, are shown in Table 1: Design Advantages of Achates Power Engine.

The Achates Power engine has been configured to optimize fuel consumption and heat rejection characteristics. Given large vehicle space requirements for fuel systems and cooling systems, the approach of minimizing the fuel consumption and heat rejection reduces the overall power pack volume for a given power output. Additionally, by reducing the heat rejected to coolant, the frontal area of the radiator can be decreased, offering aerodynamic efficiency advantages.

Table 1: Design Advantages of Achates Power Engine

Feature	Optimal	Achates Engine	Rationale
Cycle	2-Stroke	2-Stroke	▪ Combustion event in every cylinder every revolution [1] ▪ High indicated thermal efficiency from leaner combustion [2,3] ▪ Low pumping loss by only partially scavenging cylinder at times [2,3]
Architecture	Opposed Piston	Opposed Piston	▪ Favorable surface area to volume increases indicated thermal efficiency and reduces heat rejection to coolant [2,3]
Stroke/Bore	>2.4	2.4	▪ Improved scavenging reduces pumping loss [2,3] ▪ Reduced heat transfer increases indicated thermal efficiency and reduces heat rejection to coolant [2,3]
Cylinder Count	3	3	▪ 3-cylinder design maintains turbocharger energy while eliminating negative cross-charging effects [1,2,3]
Combustion System	Efficiency optimized	Efficiency optimized	▪ The patented Achates Power combustion system, including injector design and configuration, piston bowl design, and port design, is proven to have extremely high indicated thermal efficiency and low pumping losses [2,3]

Notes:1: high power density, 2: low fuel consumption, 3: low heat rejection

DEMONSTRATED PERFORMANCE

Single-Cylinder Research Engine

The custom single-cylinder research engine, shown in Figure 9, has been manufactured in-house and is tested on two in-house dynamometers. The engine has a bore of 98.425 mm, and a stroke of 215.9 mm, resulting in a displaced volume of 1.64 L. The liner geometry creates fixed port timing, and the piston geometry and injection spray pattern have been specified based on combustion simulations. The common-rail fuel injection system is capable of injection pressures up to 2000 bar and can produce multiple injection events per engine cycle.

The conditioned combustion air and EGR are delivered to the intake manifold of the single-cylinder test engine via the development system shown in Figure 10. An external air compressor feeds compressed air to the conditioning unit where it is mixed with exhaust gas taken from the exhaust side of the engine. An EGR pump, necessary because of the required pressure difference across the cylinder, pulls the exhaust through a gas-to-water heat exchanger before delivering exhaust gas to the intake stream. The EGR rate delivered to the engine is controlled by the EGR pump speed and a ball valve located downstream of the pump. After the air and exhaust gas are mixed, the intake gas flows through a second heat exchanger followed by a heater to precisely control the intake manifold temperature. The exhaust manifold pressure is set with a back pressure valve in the exhaust system.

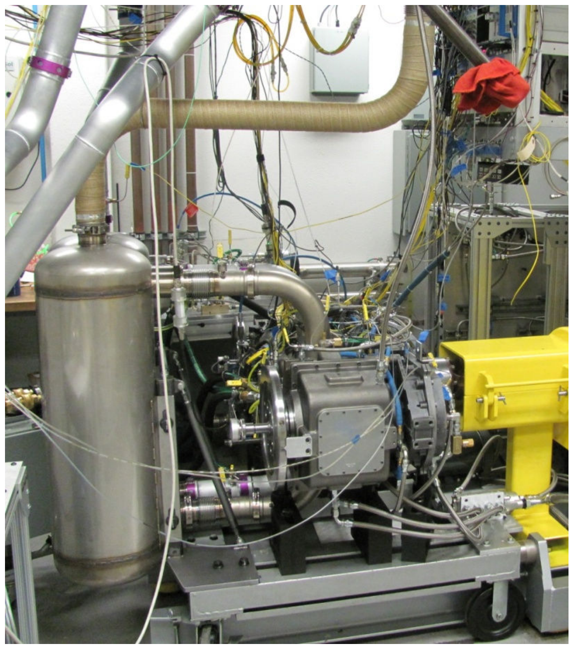

Figure 9: Single-Cylinder Research Engine Installed in Test Cell

In-cylinder pressure is measured at 0.5° crank-angle intervals with a Kistler 6052C piezoelectric pressure transducer coupled to a Kistler 5064 charge amplifier. The cylinder pressure signal is pegged to an average of the intake and exhaust manifold pressures during scavenging, measured with Kistler 4005B and 4049A high-speed pressure transducers, respectively. Custom in-house software is used to acquire and process the data. A California Analytical Instruments (CAI) emissions analyzer is used to measure the steady-state concentration of five exhaust species (CO2, CO, O2, HC, NOx) and intake CO2. An AVL 415s Smoke Meter provides a measure of exhaust soot content. A measurement of the combustion noise is provided by an AVL FlexIFEM Advanced Noise Meter.

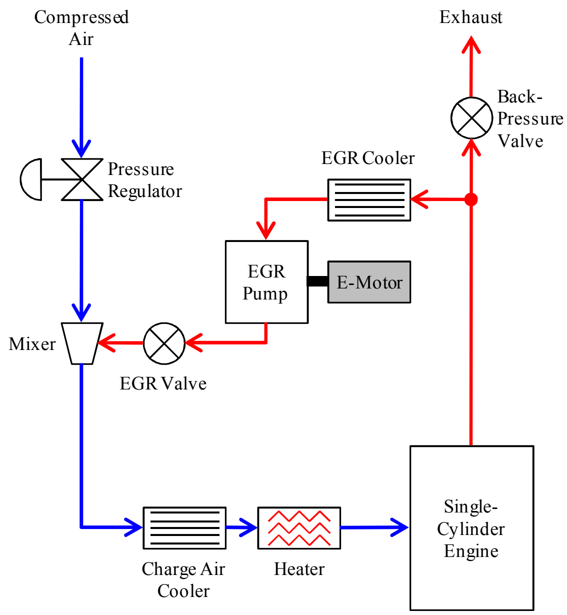

Figure 10: Schematic of the Test Cell Air and EGR Conditioning System

Interface Model

Friction and pumping energy losses, which represent the difference between indicated work and brake work, are specific for each engine configuration and do not translate from a single-cylinder to a multi-cylinder engine by simple multiplication. In order to predict the brake-specific performance of a multi-cylinder engine based on single-cylinder combustion results, an “interface model” has been created in 1D engine system simulation software. This model is correlated to the in-cylinder pressure trace to provide multi-cylinder-based predictions of the friction and pumping work required at the operating point measured on the dynamometer. The results from the interface model, therefore, provide predictions of multi-cylinder brake-specific performance and emissions parameters based on measured single-cylinder results.

Figure 11 shows the schematic of the input data and assumptions of the interface model. The combustion chamber geometry, the piston motion, and the porting profiles are identical to what exists in the single-cylinder engine, while the number of cylinders and associated manifold configurations are application specific. Engine speed, fuel flow rate, air flow rate, EGR percentage, cylinder pressures at intake port closing (IPC), and intake manifold temperatures match the measured values. The rate of heat release is derived from the measured cylinder pressure and is input directly into the combustion submodel. Assumptions for the air-handling equipment, charge cooling components, and aftertreatment system are used in the pumping loss prediction. The Chen-Flynn mechanical friction model is based on the mechanism design and analysis and is correlated to experimental friction results. The work needed to drive all accessories, including the supercharger, is also taken into account.

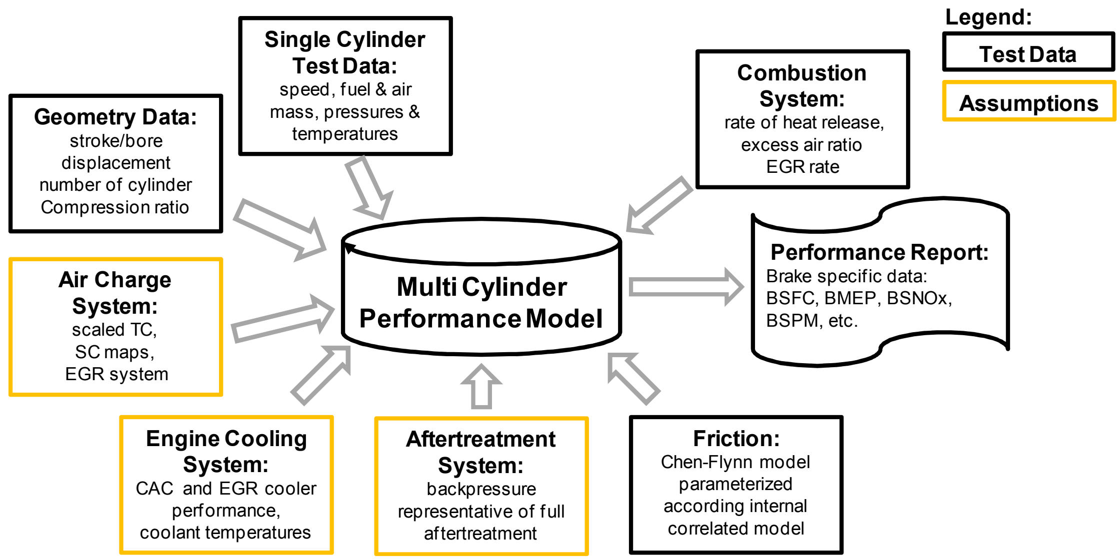

Figure 11: Multi-Cylinder Interface Model Input Data Flow

The interface model air-handling system (Figure 8) consists of a supercharger, a turbocharger, and a charge air cooler after each compression stage. The size and characteristics of the air-handling system components are application specific. The compressor and turbine is modeled using map data provided by a turbocharger supplier, and the supercharger model uses a full map obtained from a supercharger supplier. A dual-drive-ratio mechanism is assumed for the engine-supercharger connection. The two drive ratios for the supercharger are useful for maintaining high thermal efficiency over the entire engine map, for increasing low speed torque, and for enhancing the cold start capability of the engine. A supercharger recirculation loop and valve are included to control the inlet manifold pressure, and a turbine waste-gate valve is modeled for over-boost and over-speed protection, although at the conditions provided here the waste-gate valve is not needed.

EGR is introduced into the intake system after the compressor and before the first charge air cooler. It is assumed that both charge air coolers are of the air-to-water type and are located on a secondary low temperature coolant circuit. The charge air coolers' effectiveness values are set to 90%, which assumes a typical degree of cooler fouling. Charge air cooler fouling with this arrangement is expected to be less pronounced than in four-stroke engines. The hot EGR mixes with cooler compressor outlet air prior to entering the charge air cooler, which significantly reduces the inlet charge temperature and the soot concentration thereby decreasing the likelihood of fouling [8]. The second charge air cooler is assumed to be mounted close to the intake manifold in a high position to avoid condensate build-up in the cooler and the associated corrosion.

The interface model requires a detailed characterization of the scavenging process because it is important to arrive at the correct concentrations of fresh air and residual gas in the cylinder prior to the start of the closed-cycle portion of the simulation. For this reason, the scavenging efficiency was measured in the test engine using an in-cylinder CO2 sampling method [9], and the scavenging efficiency versus delivery ratio relationship was used in the interface model correlation process.

The interface model was exercised by first setting a turbine effective diameter and the supercharger mechanical drive ratios. Then for each operating condition, the compressor and turbine efficiencies were specified based on supplier data for off-the-shelf production hardware, and the two-stroke scavenging schedule was set to match measured results at the given speed and delivery ratio. Finally, the cylinder pressure trace, intake air flow per cylinder, and EGR percentage were matched to experimental results when using the measured rate of heat release by adjusting the following parameters: supercharger drive ratio (to one of the two possible values), supercharger recirculation valve, and EGR valve position. If a sufficient match to the experimental results could not be achieved, a new set of boundary conditions was provided to the single cylinder engine, the experiment was re-run with the new operating condition, and the interface model was re-matched to the updated experimental results. This iterative process typically succeeded within two to three iterations. The results of this correlation exercise provided a prediction of the brake specific parameters assuming a fixed turbine size and a dual-drive-ratio supercharger.

Engine Performance

The process of measuring single-cylinder combustion results and then using the interface model to predict multi-cylinder engine performance has been exercised for an operating range typical of an engine in a commercial vehicle. The specifications of this engine are provided in Table 2. The target full load torque curve, expressed in BMEP is shown in Figure 12. It has a significant torque backup as is very common for heavy-duty truck engines. It should be noted that although the total engine power output for a three-cylinder of 160 kW would be underpowered for a typical heavy-duty truck application, scaling this engine to a larger displacement per cylinder, based on the target full load BMEP curve would not only increase the power but further improve the thermal efficiency because of the more favorable surface area to volume ratio.

Table 2: Medium-Duty Engine Specifications

Maximum Power	53 kW/cylinder @ 2160 rpm
Maximum Torque	334 Nm/cylinder @ 1390 rpm
Number of Cylinders	3
Displaced Volume	1.64 L/cylinder
Stroke	215.9 mm
Bore	98.425 mm
Maximum BMEP	12.8 bar
Trapped Compression Ratio	15.6:1

The engine operating conditions, designated as A25, A75, A100, B50, B75, B100, C25, C75 and C100 are derived from the steady-state supplemental certification cycle adopted by the U.S. and Europe [10]. Only nine of the 13 engine modes are considered as a representative subset for measuring fuel consumption and emissions in order to reduce total testing time. The same weighting factors as specified by the regulations are used to calculate the cycle-average fuel consumption and emissions values.

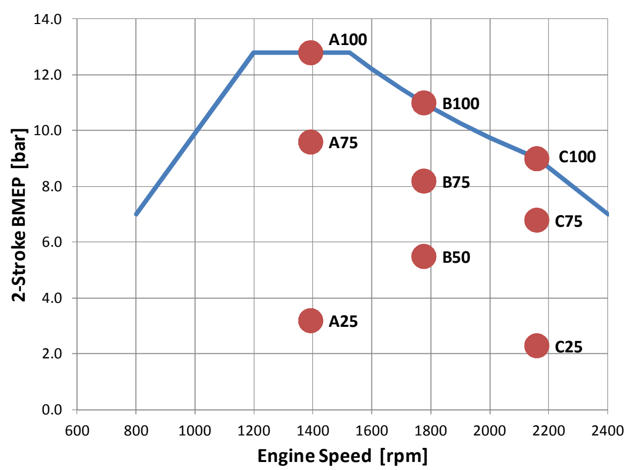

Figure 12: 4.9L 3 Cylinder Engine – Target Full Load BMEP Curve

Table 3 provides performance and emissions results for the Achates Power opposed-piston, two-stroke 4.9L three cylinder engine, where the indicated results were measured directly in the single-cylinder research engine, and the brake-specific performance values were based on the multi-cylinder interface-model predictions for friction and pumping losses. The operating conditions were selected based on the assumption that a Vanadium SCR catalyst, which typically has a NOx conversion efficiency of over 90% in a temperature range between 300 and 400 °C, was used as the NOx aftertreatment device.

Table 3: Achates Power Opposed-Piston, Two-Stroke Engine
Performance and Emissions Results

Engine Condition		**A25**	**A75**	**A100**	**B50**	**B75**	**B100**	**C25**	**C75**	**C100**
Engine Speed	rpm	1391	1391	1391	1775	1775	1775	2158	2158	2158
IMEP	bar	3.8	10.8	14.5	6.4	9.8	13.4	3.0	8.1	11.6
BMEP	bar	3.2	9.6	12.8	5.5	8.2	11.0	2.3	6.8	9.0
Indicated Power	kW	42.9	123.6	165.5	93.1	143.3	195.7	52.7	144.4	205.9
Brake Power	kW	36.6	109.7	146.2	80.2	119.6	160.4	40.8	120.5	159.5
Indicated Thermal Efficiency	%fuel	52.3	51.3	51.8	51.1	52.5	51.4	52.1	51.5	51.9
Brake Thermal Efficiency	%fuel	44.9	45.8	46.0	44.3	44.1	42.4	40.6	43.2	40.5
Friction Loss	%fuel	6.1	4.2	3.9	5.6	5.3	4.9	8.9	5.9	5.7
Pumping Loss	%fuel	1.3	1.3	1.8	1.2	3.2	4.1	2.6	2.3	5.7
ISFC (Engine)	g/kWh	159.5	162.8	161.2	163.4	158.8	162.4	160.1	162.1	160.8
BSFC (Engine)	g/kWh	187.0	183.4	182.5	189.6	190.3	198.1	206.9	194.3	207.5
BSNOx	g/kWh	1.595	4.015	4.325	2.199	2.421	4.489	2.021	2.732	3.823
BSSOOT	g/kWh	0.006	0.02	0.014	0.019	0.015	0.022	0.016	0.028	0.032
BSCO	g/kWh	0.593	0.225	0.122	0.149	0.097	0.118	0.462	0.156	0.128
BSHC	g/kWh	0.455	0.214	0.186	0.264	0.242	0.223	0.477	0.274	0.278
Peak Cylinder Pressure	bar	79	148	188	111	153	200	79	148	198
50% Mass Burned Fraction	deg aMV	1.6	2.2	4.0	-0.2	2.8	4.5	-0.7	1.2	3.7
Burn Duration 10-90%	deg	10.1	15.9	16.0	16.3	17.8	19.8	12.2	18.8	18.6
AVL Noise	(dB)	91.9	90.4	89.8	94.0	91.8	89.0	97.1	92.2	90.6
Air/Fuel Ratio	-	29.1	22.6	23.5	22.8	25.5	24.8	31.6	24.7	24.5
External EGR Rate	%	32.6	29.3	28.2	31.7	38.3	29.0	29.5	31.4	33.9
Intake Manifold Pressure	bar	1.264	2.063	2.733	1.678	2.383	3.355	1.399	2.411	3.405
Intake Manifold Temperature	degC	38	51	54	48	54	61	42	56	67
Turbine Outlet Temperature	degC	277	404	389	363	299	339	261	323	308
Fuel Spec. Oil Consumption	%fuel	0.072	0.103	0.123	0.103	0.105	0.128	0.171	0.126	0.14
Brake Spec. Oil Consumption	g/kWh	0.13	0.19	0.22	0.20	0.20	0.25	0.35	0.24	0.29

The peak brake thermal efficiency of 46.0$\%_{fuel}$ occurs at the A100 operating condition and is equivalent to achieving a brake-specific fuel consumption of 182.5 g/kWh. All the considered operating conditions are also highly efficient, with brake thermal efficiencies in excess of 40%$_{fuel}$. For all load points the 50% mass burned fraction is earlier than 4.5 deg crank angle after minimum volume (aMV) for optimal efficiency for the 1.6 L power cylinder unit. This is very advanced compared to 4-stroke engines and has to do with the reduced heat transfer during combustion and shorter combustion. The combustion event is not only advanced it is also shorter than comparable 4-stroke engines. Since the engine operates in a 2-stroke cycle, the energy release per time can be faster for the same combustion noise level and peak cylinder pressures.

Table 3 also shows air/fuel ratios, external EGR rates and intake manifold conditions. The minimum exhaust temperature downstream the turbine is 261 C and the maximum is 404 C. This is a relative small spread which enables the use of a catalyst coating optimized to higher conversion efficiencies for a smaller temperature range. The oil consumption is measured with the Da Vinci sulfur trace method.

DESIGN SOLUTIONS AND DURABILITY DEMONSTRATION

While opposed-piston engines eliminate many engine components that are among the most common to fail in conventional engines - cylinder head, cylinder head gasket, exhaust valves, cams, etc. - they introduce new design features that have to be fully validated.

Wrist Pin Durability

Because wrist pins for two-stroke engines are primarily under continuous compressive load, it is challenging to adequately lubricate the bearing. Without a force reversal on the wrist pin, lubricating oil fails to migrate to all surfaces of the pin, leading to premature wear.

To address this failure mode, a biaxial bearing, which is illustrated in Figure 13, has been designed and developed. This bearing design uses two distinct, non-concentric journals to carry the load. The motion and geometry of the pin and carrier alternately load and unload different portions of the bearing so that the full bearing is squeeze film lubricated in each engine cycle.

Similar designs have been used successfully in production on crosshead two-stroke marine engines and two-stroke locomotive engines. The type of wrist pin is well-understood [5].

Achates Power has designed, built and tested its own biaxial bearing to operate up to 200 bar peak cylinder pressures in an A48 engine configuration. Additionally, experimentally-calibrated and proprietary analytical models have been developed to determine minimum oil film thickness, density and film pressure of different bearing design alternatives which enable rapid design evolution. To separate the wrist pin from combustion bowl development, a bearing carrier was designed and assembled with the piston. Figure 14 shows photographs of the intake and exhaust carriers and pins after a runtime of 50 hours of full load operation at rated power (C100). The pictures show no sign of scuffing or cavitation, which demonstrates the potential of the design for heavy-duty durability requirements.

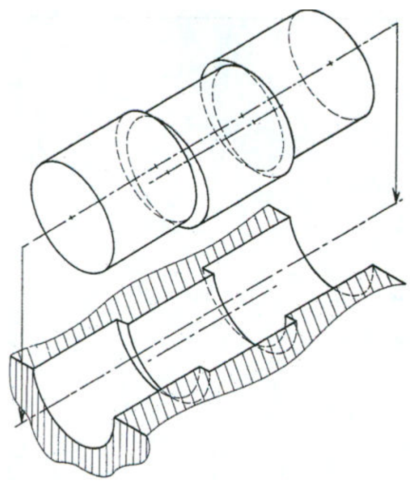

Figure 13: Bi-axial Wrist Pin Illustration

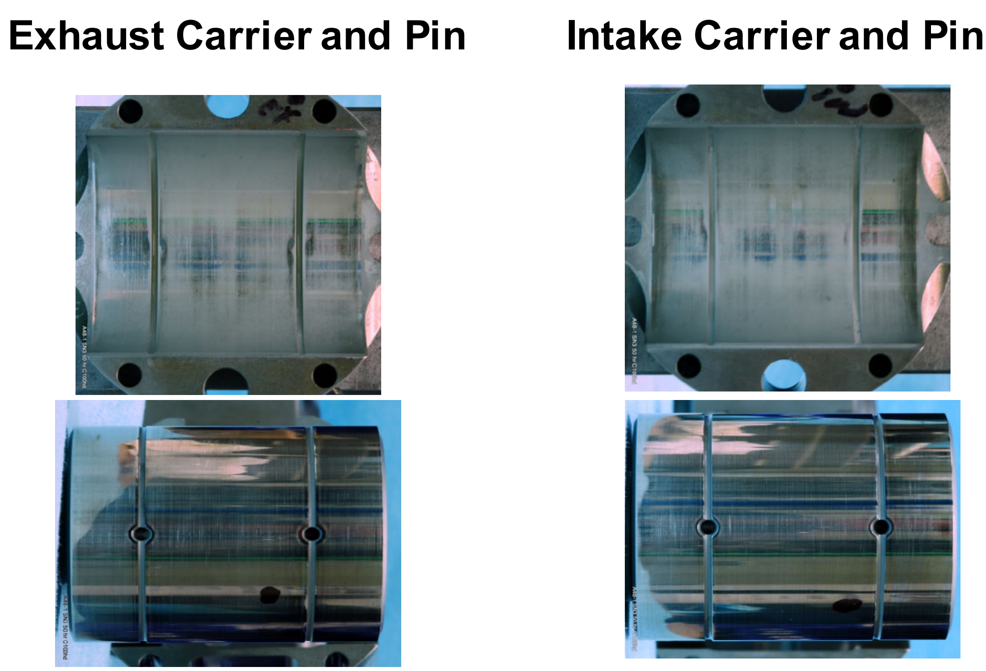

Figure 14: A48 1.6L SCE: Wrist Pins after 50 Hrs. Full Load Durability

Piston and Cylinder Thermal Management

Since two-stroke engines fire in every cylinder during every engine revolution, they tend to have high thermal loads on the piston and cylinders. The A48 power module design has several advantages over many other two-stroke engines:

- The very high thermal efficiency, described previously, results in less heat being transferred to the combustion chamber surface than in other 2-stroke engines.
- The Achates Power engine has moderate BMEP and gas temperatures compared to some two-stroke engines.

Achates Power has extensive experience in managing the piston and cylinder thermal loads. Proven impingement cooling solutions for both pistons and cylinders have been developed, which are covered by multiple U.S. and foreign patents and patent applications (Lemke, Hoffman, Wahl, & Lee, 2008), (Lemke, McHargue, Wahl, & Lee, 2009).

Beyond this, management of piston and liner temperatures requires a total systems approach, including:

- Selection of the appropriate injector spray patterns and piston bowl geometries to reduce heat flux into the piston crown.
- Port timing selections that manage trapped air charge temperatures.
- Use of appropriate calibration settings, such as air/fuel ratios and beginning of injection timing.
- Optimal flow rates of the piston cooling jets and fill ratios for the galleries.

One of the key cylinder cooling objectives is to maintain a uniform temperature axially along the bore to minimize bore distortion and allow use of lower friction piston rings. This is accomplished by introducing the impingement cooling along the outside circumference of the middle of the cylinder, which is the area of highest heat transfer. This also allows the coolant flows along the exhaust side and the intake side of the cylinder to be metered separately, recognizing that there will be higher heat transfer on the exhaust side.

Achates Power has published a technical paper SAE 2012-01-1215 “Cylinder Cooling for Improved Durability on an Opposed-Piston Engine” on the topic at the 2012 SAE World Congress [6]. The paper describes the analytic methods to design and analyze cylinder cooling solutions. The method includes performing conjugate heat transfer analysis using computational fluid dynamics, accounting for the dynamic effects of swirl and piston motion. The paper also describes how alternative cooling jacket designs were analyzed and refined to find a robust solution.

Technical goals for maximum piston temperatures have been developed to prevent oil degradation, ring jacking and carbon build-up. Experimentally-calibrated and proprietary analytical models are used to evaluate thermal loading and thermal management of piston and cylinder design alternatives, which enables rapid design evolution.

Oil Consumption and Cylinder Durability

Ported engines have historically been known to have problems with excessive oil consumption, driven by a tradeoff between low oil consumption and acceptable cylinder and piston durability. The oil consumption goal of Achates Power for commercial engines is 0.1% of fuel (fuel specific oil consumption = 0.1%), well within range of commercial acceptance.

A Da Vinci sulfur trace system is in use at Achates Power that enables the measurement of oil consumption in real time [7].

Mitigation techniques to reduce oil consumption, as needed, include:

- Modifying oil ring tension.
- Modifying scraper element conformability.
- Modifying ring end gaps, end chamfers and land chamfers.
- Modifying ring groove tilt, pinch, keystone angle, texture and flatness.
- Modifying ring side clearance, cross sealing and side sealing.
- Modifying volume behind ring and volume between rings.
- Modifying bore texture and form after honing and form at operating temperature.
- Modifying cylinder cooling design and operation to change liner temperature.
- Unique honing approaches

By using the Da Vinci sulfur trace system, the measurement of an oil consumption map for the engine can be done within a few hours. Figure 15 shows the map covered by the 13 modes of the steady-state emission test cycle. The cycle averaged fuel specific oil consumption is measured to 0.114 %. The maximum measured showed 0.171 % and the minimum is at 0.072 %. These are exceptional values for a 2-stroke engine and approach those of best-in-class four-stroke engines.

In addition, poorly designed ported engines have been known to have problems with piston ring clipping, where the ring makes metal-to-metal contact with a port timing edge. This contact abrades the material and eventually leads to scuffing or excessive wear of the liner. Achates Power has developed and tested a number of design solutions to mitigate this problem.

To mitigate this failure mode, experimentally calibrated proprietary analytical models have been developed to evaluate the ring clipping potential of different design alternatives, which enable rapid design evolution. Current mitigation actions have included:

- Reducing port widths to limit ring excursion
- Designing unique port geometries that limit contact stress and radial acceleration of the ring as it traverses the ports.
- Design of the free ends of the ring to guide the rings back into the port without clipping

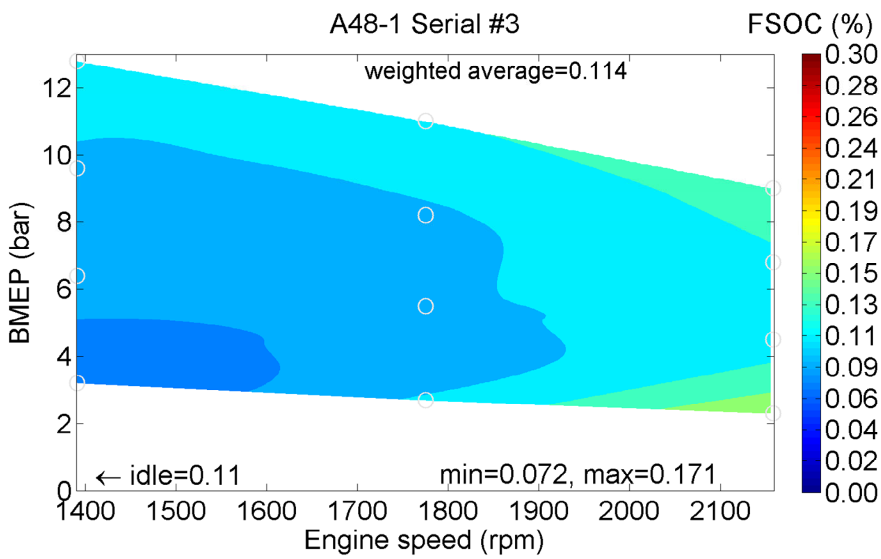

Figure 15: Fuel-Specific Oil Consumption Map of the A48 1.6L SCE Engine

Full Load Durability Screening

In order to screen the potential for HD durability requirements a 50 hrs. rated output (C100) test was performed. For assessment of degradation in performance and oil consumption map data have been measured before and after the 50 hrs. full load using the same hardware configuration. Including oil consumption maps, hot starts and daily cold starts, the operating time was split up into:

- 50.7 hrs @ rated power (C100)

- 14.5 hrs @ B50
- 8.7 hrs @ A50
- 21.4 hrs. transients and other load points

The overall runtime before tear down of the engine was 95.3 hrs.

The fuel consumption and power at C100 before and after the test was identical. The oil consumption as a weighted average over the 13 steady-state modes changed only 0.012% points of fuel from 0.133% to 0.145%.

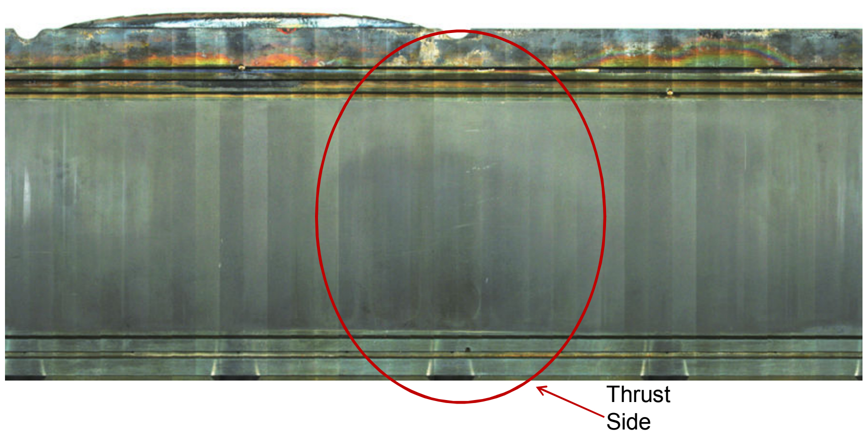

Figure 16: Intake Piston Unrolled (Color Enhanced) – Post Test

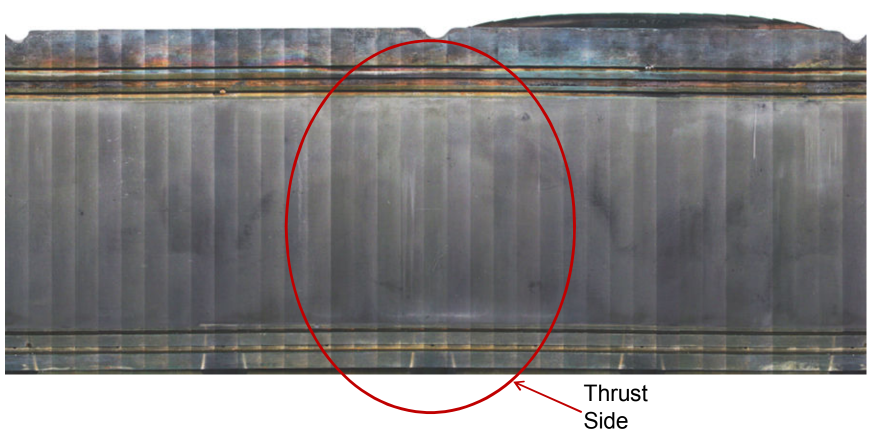

Figure 17: Exhaust Piston Unrolled (Color Enhanced) – Post Test

After more than 90 hrs. run time the engine was torn down and inspected for enhanced wear or signs of scuffing. Figure 16 and Figure 17 display the unrolled intake and exhaust pistons, respectively. The pictures are color enhanced for a better assessment. No signs of scuffing or even the potential to scuff can be seen on either piston. No sign of ring jacking in the grooves has been discovered.

No measurable wear was found at the top and second compression ring. The effacement was centered on both rings on both intake and exhaust sides. The oil control rings show excellent signs of cross-sealing on top and bottom faces and a smooth uniform effacement on the lower scraper face. No measurable wear was detected on the oil control rings.

Conclusion of the 50 hrs. full load screening test:

- No signs of scuffing or high wear have been found for the power cylinder unit.
- Also the rest of the engine was in excellent condition and was ready to continue running.
- All power cylinder parts showed the potential to fulfill heavy-duty durability requirements.

SUMMARY

The Achates Power opposed-piston engine is the most thermally efficient internal combustion engine known, combining advantageous surface area/volume ratio, short combustion duration, higher ratio of specific heat and efficient scavenging. The architecture also has benefits of high-specific power, light weight and low cost. Achates Power has modernized the architecture by mitigating the design challenges and designing a proprietary and patented combustion system that is clean and efficient. Achates Power has validated its engine designs through more than 3,000 hours of dynamometer testing.

Latest test results show a brake thermal efficiency of 46% for the best point for a medium-duty engine size. The demonstrated performance is based on today’s production parts of turbochargers, superchargers as well as air system and aftertreatment components. Compared to a typical 4-stroke commercial vehicle engine, the efficiency islands of the engine map for the Achates Power opposed-piston, 2-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.

Robust and reliable design solutions for wrist pin, piston and cylinder thermal management have been developed, tested and validated. The cycle averaged measured oil consumption of 0.114 % of fuel is exceptional for a 2-stroke engine and comes close to best in class oil consumption data of 4-stroke diesel engines. A 50 hrs. full load durability screening test did not show any sign of wear or scuffing of the power cylinder unit. Performance and oil consumption before and after the test have been measured and did not show any degradation.

MODERNIZING THE OPPOSED-PISTON, TWO-STROKE DIESEL ENGINE FOR MORE EFFICIENT COMMERCIAL VEHICLE APPLICATIONS

The Achates Power opposed-piston, 2-stroke diesel engine architecture demonstrated significantly higher brake thermal efficiencies than a 4-stroke diesel engine with the same power. Implementing unique and proprietary design solutions demonstrated not only the efficiency potential, but also durability and low oil consumption needed for a successful future generation of commercial vehicle engines.

REFERENCES

[1] Aircraft Diesels by P.H. Wilkinson, pub. Pitman Publishing Corporation, 1940.

[2] Foster, D., Herold, R., Lemke, J., Regner, G., Wahl, M., "Thermodynamic Benefits of Opposed-Piston Two-Stroke Engines", SAE International Technical Paper 2011-01-2216, 2011, doi:10.4271/2011-01-2216.

[3] Flint, M. and Pirault, J.P., "Opposed Piston Engines: Evolution, Use, and Future Applications", SAE International, Warrendale, PA ISBN 978-0-7680-1800-4, 2009.

[4] Sandoval, D. and Heywood, J., "An Improved Friction Model for Spark-Ignition Engines", SAE International Technical Paper 2003-01-0725, 2003, doi:10.4271/2003-01-0725.

[5] Wakuri, Y., Ono, S., & Soejima, M. (1982). On the Lubrication of Crosshead-pin Bearing with Eccentric Journal. Bulletin of the JSME, Vol. 25, No. 206.

[6] Lee, P., Wahl, M., "Cylinder Cooling for Improved Durability on an Opposed-Piston Engine", SAE International Technical Paper 2012-01-1215, 2012, doi:10.4271/2012-01-1215.

[7] Callahan, B., Froelund, K., Wahl, M., "Oil Consumption Measurements for a Modern Opposed-Piston Two-Stroke Diesel Engine", ASME Technical Paper, ICEF2011-60140, 2011.

[8] Teng, H. and Regner, G., "Characteristics of Soot Deposits in EGR Coolers", SAE International Journal of Fuels and Lubricants, Vol. 2, No. 2, pp. 81-90, 2010. Also published as SAE Technical Paper 2009-01-2671, 2009.

[9] Foudray, H.Z. and Ghandhi, J.B., "Scavenging Measurements in a Direct-Injection Two-Stroke Engine", SAE Technical Paper 2003-32-0081, 2001.

[10] Heavy-Duty Supplemental Emissions Test (SET), Retrieved from http://www.dieselnet.com/standards/cycles/set.php, 2010.

