

# Opposed - Piston Two - Stroke Diesel Engine - A Renaissance

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## ABSTRACT

Historically, the opposed-piston two-stroke diesel engine set combined records for fuel efficiency and power density that have yet to be met by any other engine type. In the latter half of the twentieth century, the advent of modern emissions regulations stopped the wide-spread development of two-stroke engine for on-highway use. At Achates Power, modern analytical tools, materials, and engineering methods have been applied to the development process of an opposed-piston two-stroke engine, resulting in an engine design that has demonstrated 13% fuel consumption improvement compared to a state-of-the-art 2010 medium-duty diesel engine at the same engine-out emissions levels while achieving specific oil consumption less than 0.1% of fuel over the majority of the operating range. Further benefits of the opposed-piston two-stroke diesel engine over a conventional four-stroke design are a reduced parts count and lower cost.

## INTRODUCTION

Opposed-piston two-stroke engines were conceived in the late 1800's in Germany and subsequently developed in multiple countries for a wide variety of applications [1-3]. An excellent summary of the history of opposed-piston engines can be found in [1]. Produced initially for their manufacturability, high power density, and competitive fuel efficiency, opposed-piston two-stroke engines demonstrated their versatility in a variety of applications including aircraft, ships, tanks, trucks, and locomotives and maintained their presence throughout most of the twentieth century. Historically, all types of engines have faced a number of technical challenges related to emissions, fuel efficiency, cost and durability - to name a few - and these challenges have been more easily met by four-stroke engines, as demonstrated by their widespread use. However, the limited availability of fossil fuels and the corresponding rise in fuel cost has led to a re-examination of the fundamental limits of fuel efficiency in internal combustion (IC) engines, and opposed-piston engines, with their inherent thermodynamic advantage, have emerged as a promising alternative. This paper discusses the

potential of opposed piston engines in light of today's market and regulatory requirements by applying state-of-the-art tools and methods to the engine development process.

## Opposed-Piston Two-Stroke Engine Advantages

A number of fundamental advantages of opposed-piston two-stroke engines make them attractive alternatives to common four-stroke engines. The opposed-piston (OP) arrangement, characterized by two pistons reciprocating opposite to each other in a common cylinder, has inherent heat transfer benefits compared to a standard crank-slider arrangement with a single piston and a cylinder head, and these benefits can be realized without sacrifices to engine friction or mechanical durability. First, with two pistons moving, a larger cylinder displacement can be realized for a given cylinder bore leading to a reduction in the number of cylinders compared to an engine with a standard crank-slider/cylinder head arrangement. A reduced number of cylinders decreases the surface area available for in-cylinder heat transfer. Second, an effective stroke-to-bore ratio in the range of 2:1 to 3:1 can be realized without increasing the piston speed, leading to more favorable surface-area-to-volume ratios and a further reduction of in-cylinder heat transfer. Third, the OP arrangement eliminates the cylinder head and replaces it with a second piston that can be maintained at a higher metal temperature, reducing the thermal losses to that surface of the combustion chamber.

The OP engine also has mechanical advantages compared to a standard four-stroke engine. The two moving pistons allow for piston porting, whereby the pistons uncover intake and exhaust ports at opposite ends of the cylinder and expose the combustion chamber to the intake and exhaust manifolds, respectively. The piston porting eliminates the need for poppet valves and the valve actuation mechanism, increasing the simplicity and decreasing the cost of the engine while eliminating the friction and durability concerns associated with the engine valvetrain. The near symmetric movement of the opposing pistons lead to excellent engine balance,

even for single-cylinder configurations, and thereby reduce the loading of the crankshaft bearings.

The two-stroke cycle and its double firing frequency gives engine designers the choice of decreasing brake mean effective pressure (BMEP) levels and increasing power density compared to four-stroke engines of equivalent power output. The lower BMEP levels can be accomplished with lower peak cylinder pressures and therefore lower peak cylinder temperatures, both of which lead to design advantages. The lower cylinder pressures result in lower mechanical stress on engine components and therefore can be designed to be of lighter weight. The lower cylinder temperatures result in decreased NO<sub>x</sub> formation during combustion, lowering the requirements for exhaust gas recirculation (EGR) and/or NO<sub>x</sub> aftertreatment devices. The increased power density leads directly to smaller engine package size and weight, both of which are beneficial to increasing overall vehicle fuel economy and to decreasing manufacturing costs.

## Modern Solutions to Opposed-Piston Two-Stroke Engine Challenges

The challenge posed by the emissions regulations in the latter half of the twentieth century was difficult to overcome for two-stroke engines of any architecture and led engine manufacturers to generally favor four-stroke engine development. As demonstrated by the results reported in this paper, the emissions challenge - when revisited with modern analytical tools, materials, and engineering methods - is no longer limiting the successful design of a clean and efficient OP two-stroke engine.

Since the OP arrangement has no cylinder head, the fuel injector must be installed in the cylinder liner. Historically, this has created a technical challenge compared to the common crank-slider arrangement with the fuel injector located in the center of the cylinder head. The large distances in the fuel-spray direction (i.e. across the diameter of the cylinder) combined with low fuel injection pressures made accessing all of the available air in the combustion chamber difficult, resulting in inefficient combustion with relatively high NO<sub>x</sub> and soot formation. Additionally, the interaction between the fuel spray and the in-cylinder fresh-charge motion with traditionally high swirl resulted in combustion occurring near the combustion-chamber surfaces causing increased thermal loading of the piston, piston rings, and cylinder liner, and leading to reduced thermal efficiency and increased cooling requirements.

Thanks to modern development tools and advanced fuel systems, the OP arrangement with a liner-mounted injector has turned from a technical challenge into a unique opportunity. The availability of fuel systems with high injection pressures and the greater ease of manufacturing asymmetric injector nozzle hole directions have enabled the fuel spray of liner-mounted injectors to better utilize the air within the combustion chamber with little-to-no wall impingement. Additionally, the ability to quickly and

accurately model the fuel spray, in-cylinder gas motion, and combustion using computational fluid dynamic (CFD) software packages (e.g. [4,5]) has allowed the engineering of the combustion chamber geometry and nozzle configuration to achieve clean and efficient combustion. The ability to shape two combustion chamber surfaces (the two pistons crowns) and incorporate multiple fuel injection locations on the liner has provided a larger design space than is available in common four-stroke engines.

The higher thermal loading of mechanical components compared to four-stroke engines that results from the lack of the thermally relieving exhaust and intake cycles is also addressed with modern tools and materials. As discussed previously, the combustion is designed to occur away from the piston and liner surfaces, which reduces cooling requirements of these components. Additionally, conjugate heat transfer (CHT) simulations are used to analyze the cooling circuits of the engine and engineer an effective cooling system that protects components and prevents oil degradation.

Cylinder scavenging, which is a primary difference between two- and four-stroke engines, is a technical challenge for all two-stroke engines. In order to achieve a once-per-revolution firing frequency, the two-stroke engine must accomplish cylinder scavenging in roughly one-third of an engine revolution, as opposed to a full revolution in four-stroke engines, and must do so without the aid of a direct mechanical displacement pump (i.e. the piston). Instead, the two-stroke engine requires an external pressure differential between the intake and exhaust ports to induce flow through the cylinder that allows the fresh charge to replace the exhaust products. Compared to other two-stroke architectures that use loop-scavenging, the opposed-piston engine with intake and exhaust ports being located at opposite ends of the cylinder employs the more efficient uniflow scavenging [6]. The development goal for scavenging is to minimize the external pumping required to purge the exhaust residual from the cylinder while creating the charge motion for subsequent fuel/air mixing during injection. Complete characterization of the scavenging process is possible with the use of efficient and accurate CFD software packages available today (e.g. [5], [7]). Optimization of port, cylinder, and piston geometries and their effects on the developed in-cylinder flow field can be accomplished in software without time-consuming and costly hardware fabrication and testing.

With regards to emissions, one of the opportunities afforded by the two-stroke scavenging process is the ability to retain some portion of the charge in the cylinder after combustion ("internal" EGR) as a means to control NO<sub>x</sub> by simply reducing the pumping work applied by the aircharge system. This helps improve fuel efficiency at part load. For high rates of EGR, the use of cooled external EGR is still required.

High oil consumption is a traditional challenge for two-stroke engines and is problematic for two reasons. First, free oil

is a significant source of particulate emissions, and second, additives in the oil create high ash residue that tends to contaminate after-treatment devices. The engineering goal is to develop a piston-liner and ring-liner interface that uses as little oil as possible without compromising durability. Fortunately, many of the technologies developed to reduce oil consumption in four-stroke engines apply directly here: improvements in cylinder bore materials, cylinder bore finishing, piston ring technologies, crankcase breathing systems, management of cylinder bore oil impingement, synthetic oils, and low ash and phosphorus oils are all technologies that have been applied to four-stroke engines and are equally applicable to two-stroke engines. For the work reported here, an advanced oil consumption analyzer has been used [8], making research and development of these oil-control technologies quicker and much more focused.

The lubrication of wrist pin bearings is another well-known two-stroke challenge as the oil replenishing of the continuously loaded wrist pin bearing is difficult. Several techniques have evolved utilizing bearings with substantially greater surface area and ladder grooves as well as special coatings that aid boundary lubrication.

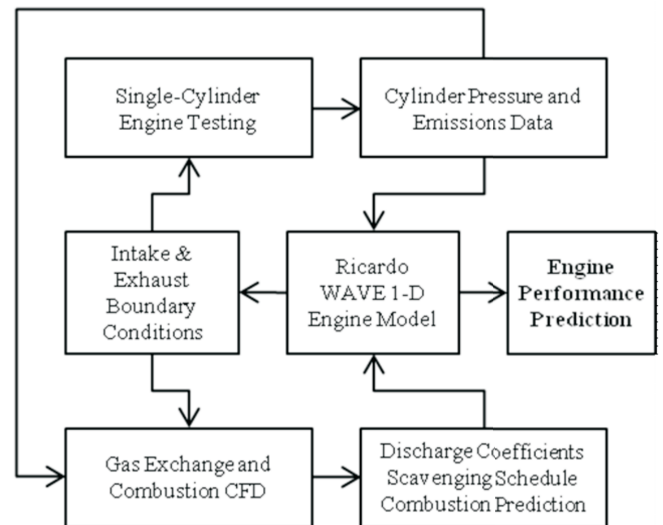
## The Renaissance of the Opposed-Piston Two-Stroke Engine

The renaissance of the opposed-piston two-stroke engine has been aided by three circumstances: the increasing demand and regulatory requirements for highly fuel-efficient and clean internal combustion engines, the thermal efficiency benefit of OP engines that is not found with other engine architectures, and the development of designs that have overcome the challenges and limitations of previous implementations. The fundamental thermal efficiency benefits of this engine along with its low emissions, small package size and weight, and low cost relative to current four-stroke engines make it an attractive alternative for future commercial and passenger vehicles. The following sections summarize the current status of the Achates Power opposed-piston two-stroke engine that is able to meet low engine-out emissions levels with acceptable oil consumption while achieving fuel consumption levels that exceed those of the current state-of-the-art four-stroke engine.

## ENGINE DEVELOPMENT PROCESS

Due to the complex interaction between scavenging, charge composition, combustion, and external air-charge system components, a detailed and thorough development process is needed in order to realize the architectural benefits of an OP two-stroke engine. The development process followed in this work combines both analytical tools and experimental measurements (see Figure 1). At the center of the development process is a correlated Ricardo WAVE 1-D engine model that takes input from various sources. The two-stroke scavenging schedule and port discharge coefficients are supplied by 3 D CFD simulations, the friction

model is based on engine friction measurements, and the heat release rate is measured by a single-cylinder research engine. In turn, the 1 D engine model supplies intake and exhaust boundary conditions for the single-cylinder research engine and performance predictions for the multi-cylinder target engine. Furthermore, emission results from the single-cylinder research engine and measured fuel spray data are used for correlation with the 3 D combustion simulations that are aimed at optimizing the combustion chamber geometry.



**Figure 1. Performance and Emissions Development Process Flow Chart**

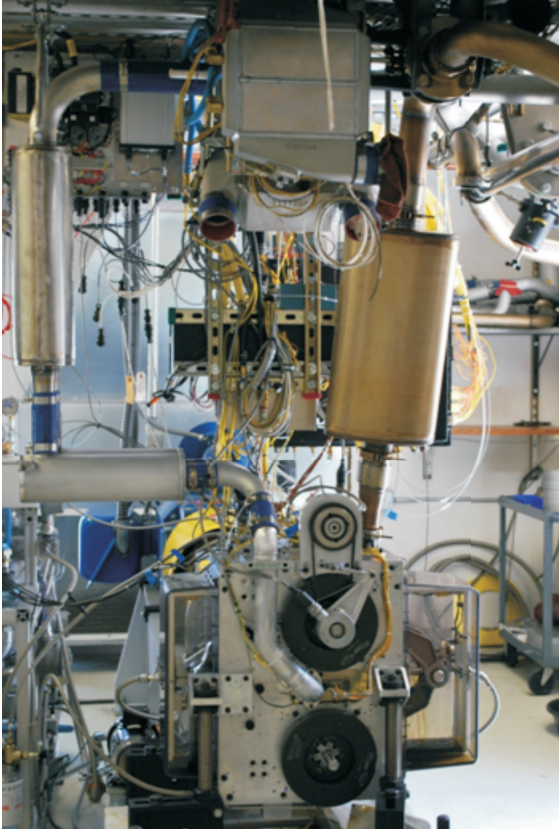
## Single Cylinder Research Engine

The custom single-cylinder research engine, shown in Figure 2, has been manufactured in-house and is tested on a 300 hp AC dynamometer. The engine has a trapped compression ratio of 17.4, a bore of 80 mm, and a stroke of 212.8 mm, resulting in a displaced volume of 1.06 l. The fixed liner geometry creates fixed swirl ratio and port timings, and the piston geometry and injection spray pattern has been specified based on analytical combustion simulation results. A common-rail fuel injection system is capable of creating injection pressures up to 2000 bar and can produce multiple injection events per engine cycle. The maximum cylinder pressure is limited to 160 bar, and the maximum liner temperature is limited to 200°C.

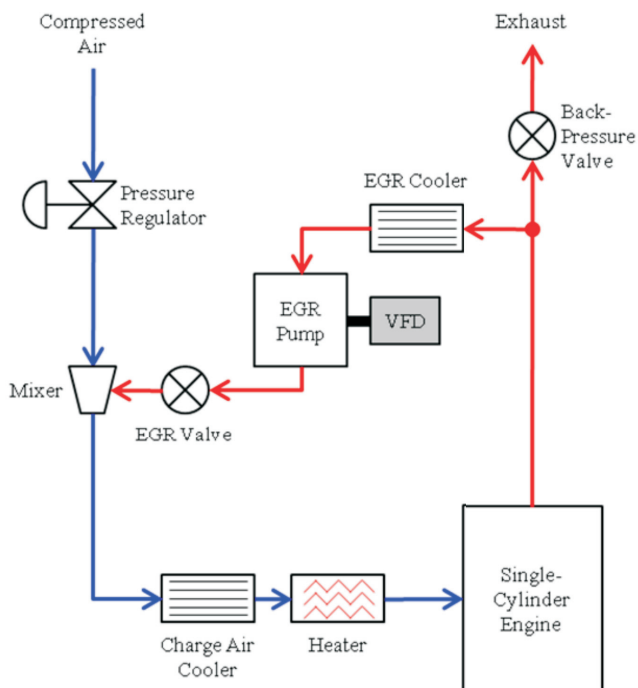
The conditioned combustion air and EGR are delivered to the intake manifold of the single-cylinder engine via the system shown in Figure 3. An external air compressor feeds compressed air to the conditioning unit where it is mixed with exhaust gas taken from the exhaust side of the engine. An EGR pump, required because of the necessary 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 2. Single Cylinder Research Engine Installed in Test Cell**



**Figure 3. Schematic of the Air and EGR Conditioning System**

In-cylinder pressure was measured at 0.5 crank-angle resolution with a Kistler 6052C piezoelectric pressure transducer coupled with a Kistler 5064 charge amplifier. The cylinder pressure signal is pegged to the intake and exhaust manifold pressures, 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 ( $\text{CO}_2$ ,  $\text{CO}$ ,  $\text{O}_2$ ,  $\text{HC}$ ,  $\text{NO}_x$ ) and intake  $\text{CO}_2$ . A Dekati DMM-230A Mass Monitor provided real-time particulate matter values, and an AVL 415s Smoke Meter provides a measure of exhaust soot content.

## 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 Ricardo WAVE. This model is exercised (as described below) to match the experimental boundary conditions and measured in-cylinder pressure trace so as 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 4 shows the schematic of the input data and assumptions of the interface model. The combustion chamber geometry, the piston motions, 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, and intake pressures and temperatures match the measured values. The rate of heat release is derived from the measured cylinder pressure and is input directly into the combustion sub-model. 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.

The interface model air-handling system (Fig. 5) consists mainly 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 supercharger model uses a full map obtained from a supplier and a dual-drive mechanism was assumed to provide two drive ratios for the supercharger, which is useful for maintaining high thermal efficiency

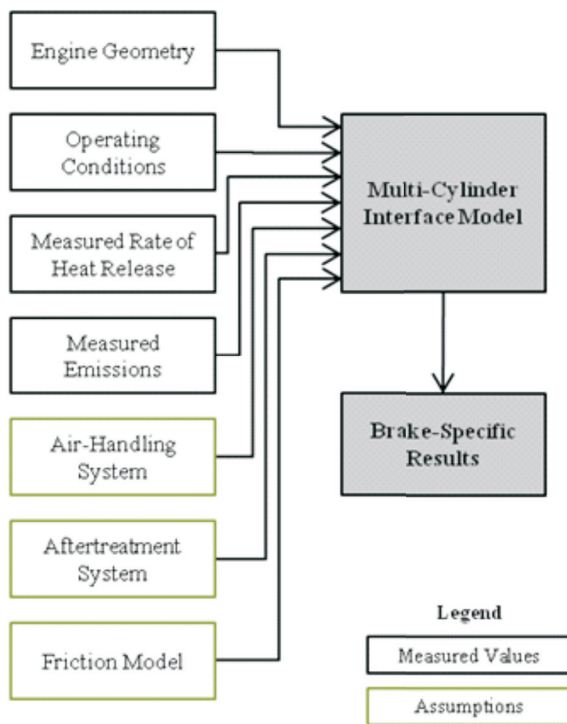


Figure 4. Multi-Cylinder Interface Model Input Data Flow

over the entire engine map, for producing excellent low speed torque, and enhancing the cold start capability of the engine. The compressor and turbine are modeled as 'mapless' components with user-specified efficiencies that are consistent with the operating point and available turbocharger supplier data. A supercharger recirculation loop and valve were included to control the inlet manifold pressure, and a turbine waste-gate was modeled for over-boost and over-speed protection.

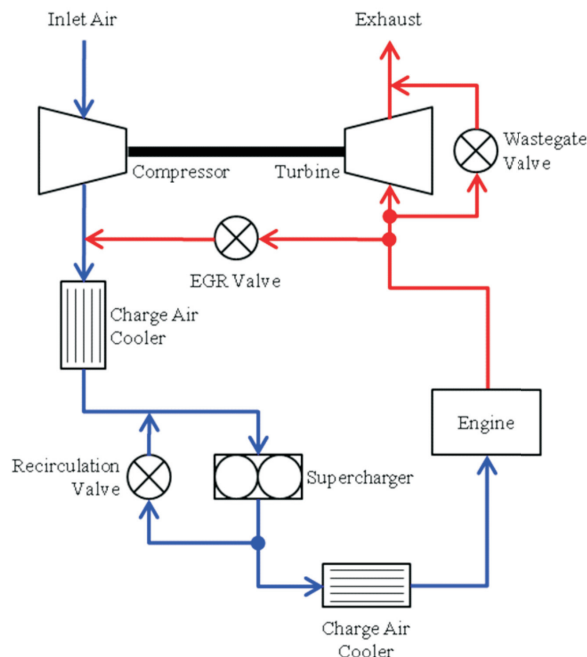


Figure 5. Air Handling System Configuration

EGR is introduced into the intake system after the turbo 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 cooler's effectiveness values are set to 90%, which is a valid assumption even with a certain 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 to the charge air cooler, which significantly reduces the inlet charge temperature and decreases the thermophoretic force, a significant source of charge air cooler fouling [9]. 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. Concerns about hydrolock associated with condensate build-up are decreased with an opposed-piston two-stroke engine because, in configurations of three cylinders or more, at least one of the cylinders will always be open to both manifolds, allowing the condensate to flow through the engine.

The interface model is exercised by first setting a turbine effective diameter and specifying the two supercharger mechanical drive ratios that will be considered for a given engine application. Then for each operating condition, the compressor and turbine efficiencies are specified based on supplier data, and the two-stroke scavenging schedule is set to match CFD simulation results at the given speed and engine load. Finally, the cylinder pressure trace, intake air flow per cylinder, and EGR percentage are 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 prescribed values), supercharger bypass valve position, EGR valve position, and waste-gate valve position. If a sufficient match to the experimental results cannot be achieved with the assumed supercharger drive ratios and turbine effective diameter, a new set of boundary conditions is provided to the single cylinder engine, the experiment is re-run with the new operating condition, and the interface model is re-matched to the updated experimental results. This iterative process typically succeeds within two to three iterations.

## RESULTS

### Medium-Duty Engine Performance

The process of measuring single-cylinder combustion results and then using the interface model to predict multi-cylinder engine performance has been applied to an engine size typically found in a medium-duty commercial vehicle. The specifications of this medium-duty engine are provided in Table 1. The engine is operated at the corner points (C100, C25, A100, A25) and center point (B50) of the steady-state supplemental certification cycle adopted by the US and Europe [10]. Only 5 of the 13 engine modes are considered to reduce total testing time, although the weighting factors

specified by the legislation are used to calculate the cycle-average fuel consumption and emissions values.

**Table 1. Medium-Duty Engine Specifications**

Maximum Power	210 kW @ 2400 rpm
Maximum Torque	920 N-m @ 1500 rpm
Number of Cylinders	4
Displaced Volume	4.26L
Stroke	212 mm
Bore	80 mm
Maximum BMEP	13.6 bar
Trapped Compression Ratio	17.4:1

Operating results for the proposed medium-duty engine are provided in Table 2. Results are given for each of the five operating conditions considered as well as the cycle-averaged fuel consumption and emissions values. Also provided in Table 3 are EURO IV emissions standards for the European Steady-State Cycle (ESC) and European Transient Cycle (ETC), which are the relevant emissions levels for commercial vehicles in India [11]. The engine operation has been optimized to achieve the EURO IV NOx standards in-cylinder, thereby eliminating the need for NOx aftertreatment devices.

The results shown in Table 2 provide indicated performance values from the single-cylinder engine and brake-specific

performance values based on the multi-cylinder interface model that includes predictions of friction and pumping losses. As shown, the brake thermal efficiencies at C100 and B50 exceed 42%, which is equivalent to a brake-specific fuel consumption of less than 200 g/kWh. The cycle-averaged fuel consumption of 203.2 g/kWh is achieved at a cycle-averaged NOx level lower than the EURO IV NOx emissions standard of 3.5 g/kWh. The cycle-averaged brake-specific PM, CO, and HC levels are higher than the EURO IV standards, but state-of-the-art DOC/DPF aftertreatment devices provide sufficient reduction of these species to bring them within acceptable limits. It should be noted that the interface model assumed a turbine outlet pressure based on a complete DOC, DPF and SCR aftertreatment model. Eliminating the SCR would reduce the engine back pressure and would further decrease the fuel consumption.

This engine was also operated at an engine-out NOx level capable of achieving the more stringent US 2010 emissions standards assuming use of an SCR aftertreatment device. The brake-specific fuel consumption and NOx emissions results are provided in Table 3. For comparison, the fuel consumption for this engine calibration is 13% lower than a state-of-the-art 2010 medium-duty diesel engine at the same engine-out NOx levels [12].

## Oil Consumption

As previously discussed, opposed-piston two-stroke engines breathe through ports on both ends of the cylinder liner. This implies that the compression rings must traverse the

**Table 2 – EURO IV Compliant Performance and “Engine Out” Emissions Results**

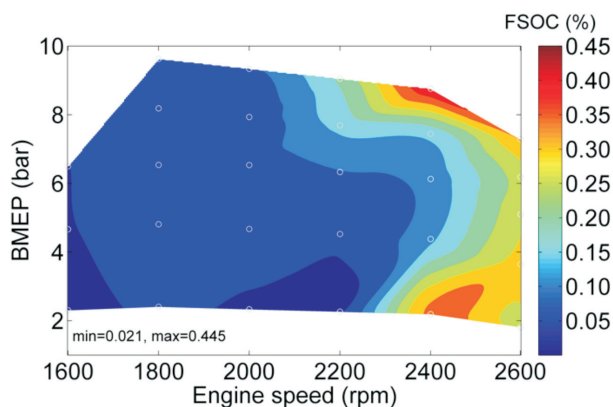
Engine Condition		C100	C25	B50	A100	A25	Avg	ES C	ETC
Engine Speed	rpm	2400	2400	2000	1600	1600	203.2	3.5	3.5
IMEP	bar	14.7	4	6.8	15.4	4.1			
BMEP	bar	12.8	3	5.7	13.6	3.4			
Indicated Power	kW	244.7	69.1	96.1	175.5	46.5			
Brake Power	kW	217.4	51.5	81.3	154	38.2			
Ind. Therm. Eff	%	49.5	47.7	50.1	45.4	48.6			
Brake Therm. Eff	%	42.9	35.3	42.4	39.8	40.1			
Friction Losses	% fuel	3.7	10.8	6.1	2.3	7.4			
Pumping Losses	% fuel	2.9	1.6	1.5	3.2	1.1			
Exhaust + Heat Loss	% fuel	50.5	52.3	49.9	54.6	51.4			
ISFC	g/kWh	169.1	176.2	167.3	184.6	172.4			
BSFC	g/kWh	195.1	237.0	197.6	210.3	209.0	203.2	3.44	0.20
BSNOx	g/kWh	3.75	2.97	2.98	3.54	2.52	3.44		
BSPM	g/kWh	0.34	0.048	0.023	0.16	0.048	0.20		
BSCO	g/kWh	3.21	0.77	0.37	4.63	0.47	2.81		
BSHC	g/kWh	1.83	0.79	0.95	0.97	0.91	1.31		



Engine Condition		C100	C25	B50	A100	A25	Avg
BSFC	g/kWh	200.8	237.9	199.7	217.5	208.7	208.1
BSNOx	g/kWh	1.24	0.89	0.98	1.12	0.83	1.12

**Table 3. US 2010 Compliant Fuel-Consumption and NOx Emissions Results**

ports. In order to achieve a low level of oil consumption, the oil control rings in the Achates Power A40 engine are not mounted on the piston but instead are stationary in the liner and wipe the piston skirt. Figure 6 shows a complete oil consumption map that was obtained from tracing sulfur in the exhaust stream [8]. This method requires the use of sulfur free fuel and lubricating oil with known sulfur content. Any oil reaching the exhaust carries a trace amount of sulfur that is picked up by the instrument. Each of the 30 engine operating condition making up the performance map is sampled for 60 sec at one sample per second. The measurement method is very repeatable and is far less time consuming than gravimetric methods. The fuel-specific oil consumption shown in Figure 6 is below 0.1% across a large portion of the operating map. Only at high speed and/or high power conditions does the oil consumption increase slightly which may have been a contributing factor for the somewhat elevated BSPM values at the C100 load point in Table 2.

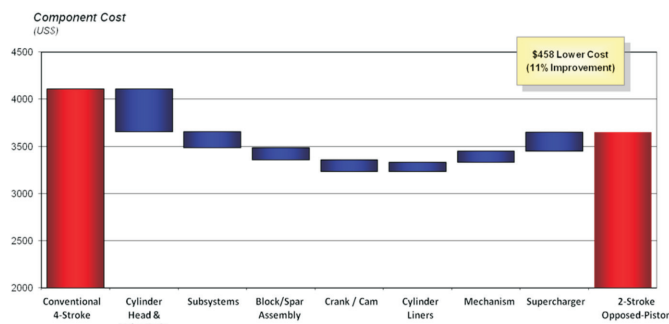


**Figure 6. Fuel Specific Oil Consumption (FSOC) for Single Cylinder Research Engine**

## ENGINE COST ANALYSIS

As discussed previously and consistent with observations by Flint and Pirault [1], the opposed-piston architecture benefits from a low part count that results in a low engine cost. This is particularly advantageous in high volume applications in which a \$500 savings per engine can yield significant savings to the manufacturer. To estimate this cost advantage, a study by a cost analysis expert was performed that compared the component cost of the Achates Power A40 engine to a conventional diesel engine meeting similar emissions standards and equivalent manufacturing volumes.

A 2007 6.7 liter inline-six engine was purchased and deconstructed to the individual component level. Each component was analyzed for its material composition, formation process, machining time and complexity plus other processes such as plating or coating. The major components were then compared directly with a similarly functioning component on the Achates Power engine, and material composition, mass, machining time and rate were compared. Only engine components and major subsystems such as the turbocharger, fuel system and lubrication system were analyzed. For the purposes of this study, other emissions-related equipments such as diesel particulate filter and NOx after-treatment were excluded. Figure 7 represents a summary of the study.



**Figure 7. Cost Comparison of OP Engine to Conventional 4-stroke Medium Duty Commercial Vehicle Engine**

The major cost advantage of the opposed-piston architecture is due to not requiring a cylinder head and corresponding valvetrain system. The smaller, more compact engine requires smaller subsystems (such as cooling and lubrication) and accounts for additional cost savings. Similarly, the structural and crankshaft components are also downsized and less costly. A slight cost increase of the opposed-piston engine is found with the cylinder liners, mechanism and supercharger, the latter being required for two-stroke operation. The overall result of this analysis showed an 11% cost improvement for the OP engine compared to a four-stroke engine of equivalent power.

## CONCLUSIONS

The results reported in this work have shown that the opposed-piston two-stroke engine architecture is a suitable platform for a highly efficient and clean internal combustion engine featuring low oil consumption. EURO IV NOx emissions are achievable in-cylinder obviating the need for NOx aftertreatment in certain markets. A SCR after-treatment can be fitted with further fuel efficiency benefits and even lower NOx levels at the tailpipe. The architectural benefit of a reduced parts count (no cylinder head) reduces cost, weight and complexity compared to conventional four-stroke engine designs.

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## DEFINITIONS/ABBREVIATIONS

OP Engine	Opposed Piston Engine
IC Engine	Internal Combustion Engine
DOC	Diesel Oxidation Catalyst
DPF	Diesel Particulate Filter
SCR	Selective Catalytic Reduction
NO <sub>x</sub>	Nitric Oxides
IMEP	Indicated Mean Effective Pressure
BMEP	Brake Mean Effective Pressure
ISFC	Indicated Specific Fuel Consumption
BSFC	Brake Specific Fuel Consumption
BSPM	Brake Specific Particulate Matter
BSHC	Brake Specific Hydrocarbons
BSCO	Brake Specific Carbon Monoxide
ESC	European Steady-State Cycle
ETC	European Transient Cycle