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The Achates Power Opposed-Piston Two-Stroke Engine: Performance and Emissions Results in a Medium-Duty Application

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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 a 15.5% fuel consumption improvement compared to a state-of-the-art 2010 medium-duty diesel engine at similar engine-out emissions levels. Furthermore, oil consumption has been measured to be less than 0.1% of fuel over the majority of the operating range. Additional 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 1800s in Europe and subsequently developed in multiple countries for a wide variety of applications [1,2,3,4,5]. 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 two-stroke engines in light of today's market and regulatory requirements, the methodology used by Achates Power in applying state-of-the-art tools and methods to the opposed-piston two-stroke engine development process, and the performance and emissions results obtained at operating conditions consistent with a medium-duty application.

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, the OP architecture creates a larger cylinder displacement for a given cylinder bore diameter, 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 [6]. 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 nearly symmetric movement of the opposing pistons leads to excellent engine balance, even for single-cylinder configurations, and thereby reduces the loading of the crankshaft bearings for OP architectures with folded single crank or rhombic dual crank drive arrangements.

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_X formation during combustion, lowering the requirements for exhaust gas recirculation (EGR) and/or NO_X 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 and resulted in inefficient combustion with relatively high NO_X and soot formation. Under some conditions, however, the fuel-spray would over-penetrate and wet the cylinder walls thereby destroying the lubricant film leading to wear, fuel dilution of the oil and high oil consumption. Additionally, the interaction between the fuel spray and the traditionally high-swirl in-cylinder fresh-charge motion resulted in combustion occurring near the combustion-chamber surfaces. The near-surface combustion led to reduced thermal efficiency and increased cooling requirements because of the higher thermal loading of the piston, piston rings, and cylinder liner.

Thanks to modern development tools and advanced fuel systems, the OP architecture 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.* [7], [8]) 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 a shorter, thermally relieving gas exchange process 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 OP two-stroke engine must accomplish cylinder charging and 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 blower-scavenged 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 far apart at opposite ends of the cylinder employs the more efficient uniflow scavenging [9]. 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.* [8], [10]). 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 burnt charge in the cylinder after combustion ("internal" EGR) as a means to control NO_X 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, but the engine's supercharger provides an efficient method to pump the EGR from the exhaust to the intake.

High oil consumption is a traditional challenge for two-stroke engines and can be problematic for two reasons. First, aerosolized oil is a significant source of particulate emissions in compression-ignition two-strokes, and second, additives in the oil create 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 equally to two-stroke engines: 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 developed for four-stroke engines but are equally applicable to two-stroke engines. As a side benefit, the lack of a valvetrain in two-stroke engines obviates the need for anti-scuffing oil additives such as zinc dialkyl dithiophosphates (ZDDP). For the work reported here, an advanced oil consumption analyzer has been used [11], 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 [6] 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 performance and emissions results of an Achates Power opposed-piston two-stroke engine that meets 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.

DATA ACQUISITION AND CORRELATION

SINGLE CYLINDER RESEARCH ENGINE

The custom single-cylinder research engine, shown in Figure 1, has been manufactured in-house and is tested on a 300 hp AC dynamometer. The engine has a trapped compression ratio of 16.7, 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 a fixed swirl ratio and port timing, 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.

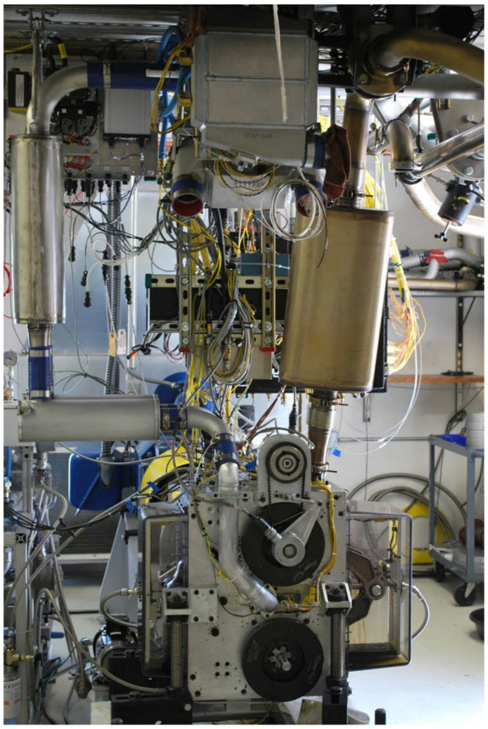

Figure 1. Single cylinder research engine installed in the test cell.

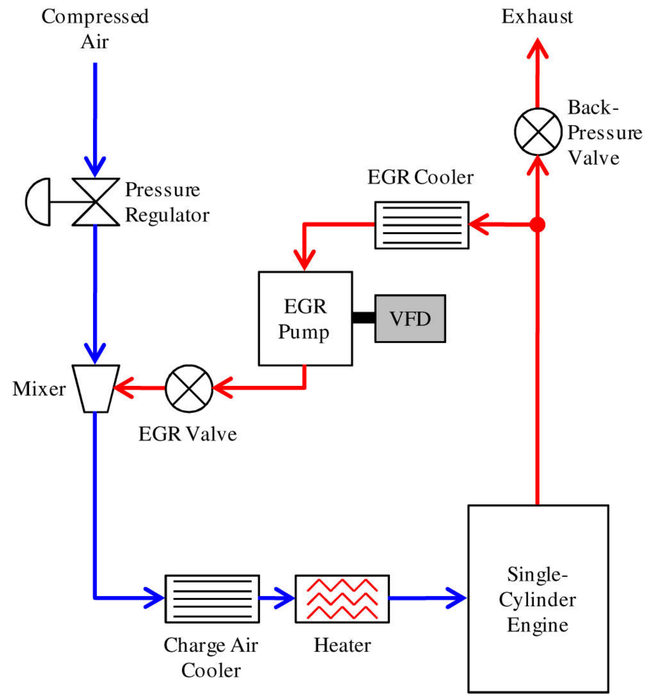

Figure 2. Schematic of the air and EGR conditioning system.

The conditioned combustion air and EGR are delivered to the intake manifold of the single-cylinder engine via the system shown in Figure 2. 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.

In-cylinder pressure was measured at 0.5° crank-angle resolution 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, 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 (CO_2, CO, O_2, HC, NO_x) and intake 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 1D engine system simulation software. This model is correlated to 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 3 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.

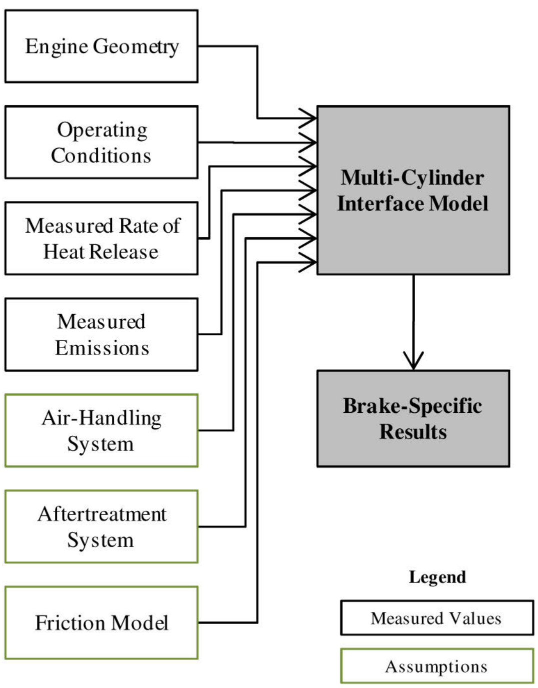

Figure 3. Multi-cylinder interface model input data flow.

The interface model air-handling system (Figure 4) 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 are modeled as 'mapless' components with user-specified efficiencies that are consistent with the operating point and available turbocharger supplier data, and the supercharger model uses a full map obtained from a supplier. A dual-drive mechanism was 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 were included to control the inlet manifold pressure, and a turbine waste-gate valve was modeled for over-boost and over-speed protection, although at the conditions provided here the waste-gate valve was not needed.

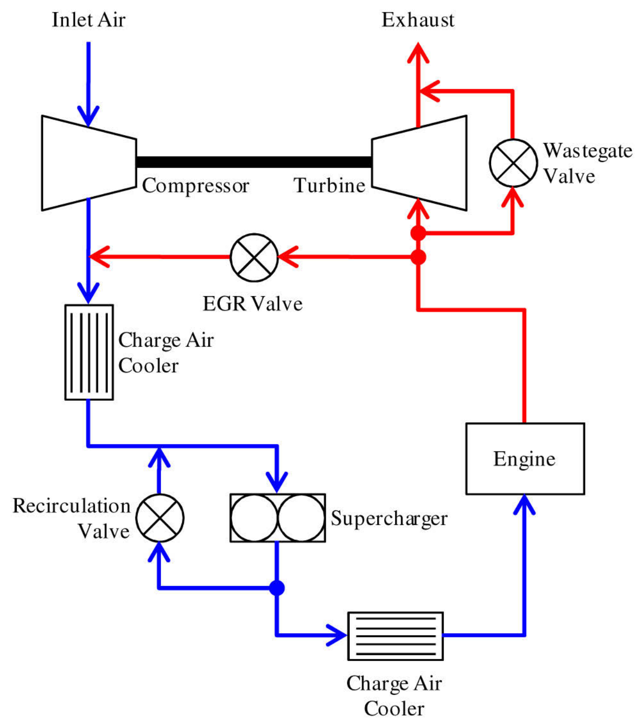

Figure 4. Air handling system configuration.

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 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 the charge air cooler, which significantly reduces the inlet charge temperature and decreases the thermophoretic force, a significant source of charge air cooler fouling [12]. 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 was exercised by first setting a turbine effective diameter and specifying the two supercharger mechanical drive ratios that were considered for a given engine application. Then for each operating condition, the compressor and turbine efficiencies were specified based on supplier data, and the two-stroke scavenging schedule was set to match measured results at the given speed and engine load. 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 prescribed values), supercharger bypass valve position, and EGR valve position. If a sufficient match to the experimental results could not be achieved with the assumed supercharger drive ratios and turbine effective diameter, 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.

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 exercised for an operating range typical of an engine in a medium-duty commercial vehicle. The specifications of this medium-duty engine are provided in Table 1. It should be noted that although the total engine power output for a three-cylinder, 1.06 L per cylinder engine would be slightly underpowered for a typical medium-duty application, the three-cylinder engine is the preferred configuration for thermal efficiency considerations and therefore was used in this study. Scaling this engine to a larger displacement per cylinder would not only increase the power but further improve the thermal efficiency. The engine operating conditions, designated as A25, A75, B50, B75, C25, and C75 are derived from the steady-state supplemental certification cycle adopted by the US and Europe [13]. Only 6 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 legislation are used to calculate the cycle-average fuel consumption and emissions values.

Table 1. Medium-duty engine specifications.

Maximum Power	46.6 kW/cylinder @ 2400 rpm
Maximum Torque	240 N-m/cylinder @ 1600 rpm
Number of Cylinders	3
Displaced Volume	1.06 L/cylinder
Stroke	212.8 mm
Bore	80 mm
Maximum BMEP	13.6 bar
Trapped Compression Ratio	16.7:1

Table 2 provides performance and emissions results for the Achates Power opposed-piston two-stroke medium-duty 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 peak brake thermal efficiency of $42.9\%_{fuel}$ occurs at the B75 operating condition and is equivalent to achieving a brake-specific fuel consumption of 195.5 g/kWh. The A75, B50, and C75 operating conditions are also highly efficient, with brake thermal efficiencies in excess of $40\%_{fuel}$. The low-load A25 and C25 conditions are less efficient because of higher relative frictional losses.

The cycle-averaged brake-specific fuel consumption (BSFC) and emissions values are provided in Table 3. The cycle-averaged BSFC of 202.7 g/kWh is achieved at a cycle-averaged NO_x level of 1.88 g/hph and cycle-averaged soot level of 0.052 g/hph. The engine-out emissions are in a range that allows 2010 US tailpipe emission requirements [14] to be met with typical aftertreatment (DOC, DPF, SCR) performance. Also included in Table 3 are cycle-averaged BSFC and NO_x emission values from a state-of-the-art, medium-duty four-stroke engine [15]. The cycle-averaged values from the four-stroke engine were averaged over the same six operating conditions as the Achates Power engine. The Achates Power engine has 15.5% lower BSFC than the reference four-stroke engine, albeit at a higher engine-out NO_x emission level. Because the reference four-stroke engine did not provide soot emission numbers, it is impossible to determine how much of the fuel consumption advantage would be sacrificed to achieve the same NO_x emissions.

Table 2. Achates Power opposed-piston two-stroke engine performance and emissions results.

Engine Condition		A25	A75	B50	B75	C25	C75
Engine Speed	rpm	1600	1600	2000	2000	2000	2400
IMEP	bar	4.0	10.7	6.9	10.3	4.1	9.8
BMEP	bar	3.2	9.4	5.8	8.8	2.9	7.8
Indicated Power	kW	34.5	91.6	73.3	110.3	52.5	126.0
Brake Power	kW	27.5	80.6	61.9	94.2	36.6	100.6
Indicated Thermal Efficiency	$\%_{fuel}$	48.3	47.3	49.3	50.2	52.0	51.2
Brake Thermal Efficiency	$\%_{fuel}$	38.4	41.6	41.7	42.9	36.2	40.8
Friction Losses	$\%_{fuel}$	7.9	3.5	6.3	4.6	12.5	5.9
Pumping Losses	$\%_{fuel}$	2.0	2.2	1.3	2.8	3.2	4.5
Exhaust + Heat Losses	$\%_{fuel}$	51.7	52.7	50.7	49.8	48.0	48.8
ISFC	g/kWh	173.6	177.3	169.9	166.8	161.3	163.7
BSFC	g/kWh	218.0	201.3	200.8	195.5	231.3	205.3
$BSNO_x$	g/hph	1.71	2.01	2.05	1.80	1.94	1.73
BSSoot	g/hph	0.008	0.067	0.065	0.038	0.011	0.077
BSCO	g/hph	0.22	1.19	0.46	0.36	0.25	0.30
BSHC	g/hph	0.32	0.25	0.43	0.34	0.54	0.46

Table 3. Cycle-averaged brake-specific fuel consumption and emissions values for the Achates Power engine and a state-of-the-art medium-duty four-stroke engine [15]. Note that only NO_X emission were provided for the reference engine.

Engine Condition		Achates Power	Ref. [12]
BSFC	g/kWh	202.7	239.9
$BSNO_x$	g/hph	1.88	0.97
BSSoot	g/hph	0.052	---
BSCO	g/hph	0.47	---
BSHC	g/hph	0.38	---

OIL CONSUMPTION

As previously discussed, opposed-piston two-stroke engines breathe through ports on both ends of the cylinder liner, which implies that the compression rings must traverse the ports. Figure 5 shows a complete oil consumption map that was obtained from tracing sulfur in the exhaust stream [11]. 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 then detected by the instrument. Each of the 29 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 5 is below 0.1% of fuel across a large portion of the operating map, and only at high speed and/or high power conditions does the oil consumption increase slightly.

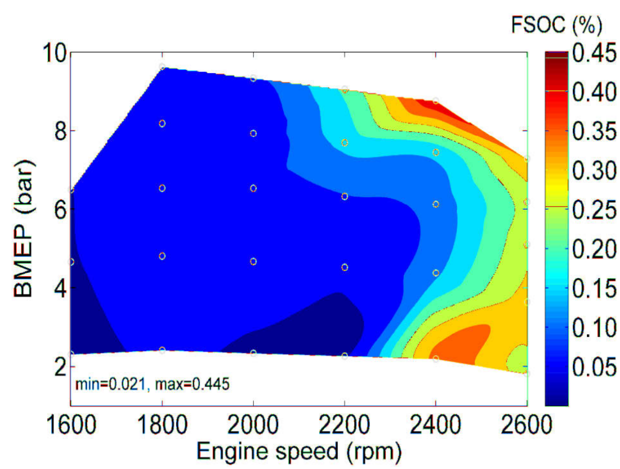

Figure 5. Fuel-specific oil consumption (FSOC) for single cylinder research engine.

ENGINE COMPLEXITY, WEIGHT AND COST ANALYSIS

As discussed previously and consistent with observations by Flint and Pirault [1], the opposed-piston architecture benefits from low weight and a low part count that results in a low engine cost. This is particularly advantageous in high volume applications in which small per engine savings can yield significant savings to the manufacturer and end-user. To estimate these advantages, a benchmarking study was conducted to compare the component complexity, weight and cost of the Achates Power opposed-piston engine to a conventional diesel engine meeting similar emissions standards and equivalent manufacturing volumes.

A 2007 6.7 liter inline-six medium-duty four-stroke engine was purchased and deconstructed to the individual component level. The first activity was to count each discrete component, which confirmed a 40% part count reduction with the opposed-piston engine as shown in Figure 6.

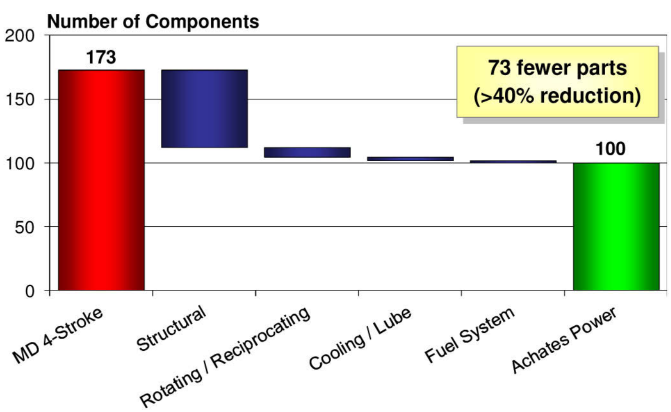

Figure 6. Component complexity comparison of OP engine to conventional four-stroke medium-duty commercial vehicle engine.

Next, each component was weighed on the medium-duty four-stroke engine and compared to the opposed-piston engine prototype. This particular engine configuration had been designed for an aviation application, and as a result optimized for low weight. Forecasts confirmed an engine weight advantage of approximately 30% for the opposed-piston engine versus the same medium-duty four-stroke engine as shown in Figure 7.

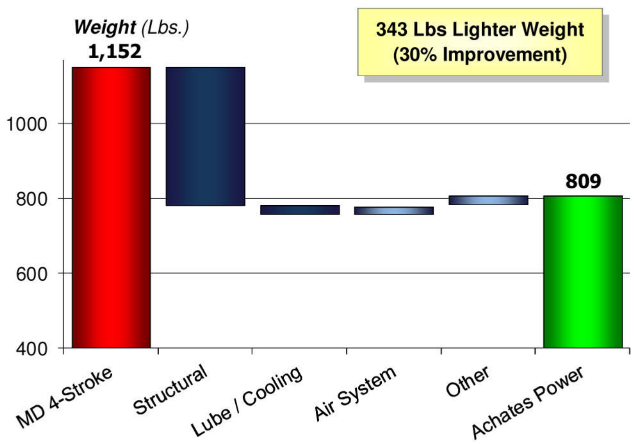

Figure 7. Weight comparison of OP engine to conventional four-stroke medium duty commercial vehicle engine.

Lastly, each component on the medium-duty four-stroke engine 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 latest generation 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 equipment such as diesel particulate filter and NO_x after-treatment were excluded. Figure 8 represents a summary of the study.

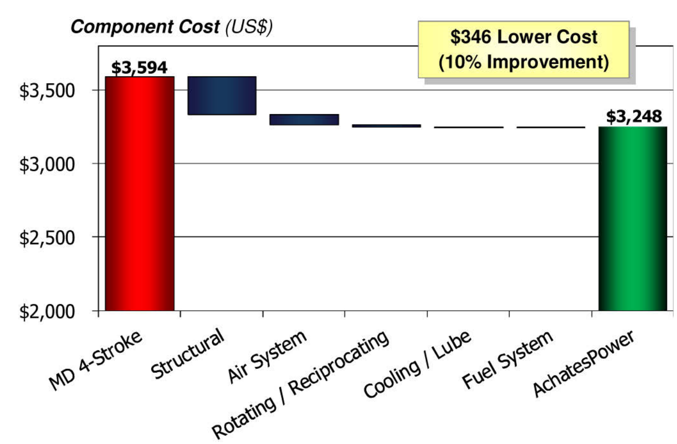

Figure 8. Cost comparison of OP engine to conventional four-stroke medium-duty commercial vehicle engine.

The major cost, weight and complexity advantages of the opposed-piston architecture result from 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 and rotating / reciprocating mechanism. The overall result of this analysis showed a 10% 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. A cycle-averaged brake-specific fuel consumption (BSFC) value of 202.7 g/kW-hr is achieved with engine-out emission levels that, which paired with reasonable aftertreatment devices, would be expected to achieve the stringent 2010 US heavy-duty emission standards. This BSFC value is 15.5% lower than a state-of-the- art four-stroke engine designed for the same medium-duty application. Oil consumption, a historical difficulty for opposed-piston two-stroke engine, was measured to be less than 0.1% of fuel for a majority of the engine speed/load map. The architectural benefit of a reduced parts count (no cylinder head) reduces cost, weight and complexity compared to conventional four-stroke engine designs.

ACKNOWLEDGEMENTS

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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_x
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

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