Meeting Stringent 2025 Emissions and Fuel Efficiency Regulations with an Opposed-Piston, Light-Duty Diesel Engine

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Abstract

With current and pending regulations-including Corporate Average Fuel Economy (CAFE) 2025 and Tier 3 or LEV III-automakers are under tremendous pressure to reduce fuel consumption while meeting more stringent NOx, PM, HC and CO standards. To meet these standards, many are investing in expensive technologies-to enhance conventional, four-stroke powertrains-and in significant vehicle improvements. However, others are evaluating alternative concepts like the opposed-piston, two-stroke engine.

First manufactured in the 1890s-and once widely used for ground, marine and aviation applications-the historic opposed-piston, two-stroke (OP2S) engine suffered from poor emissions and oil control. This meant that its use in on-highway applications ceased with the passage of modern emissions standards.

Since then, Achates Power has enhanced the opposed-piston engine and resolved its historic challenges: wrist pin and power cylinder durability, piston and cylinder thermal management, piston ring integrity and oil consumption [1].

An in-depth study on opposed-piston, two-stroke diesel engine performance and emissions in a light-duty truck application is presented here for the first time in a technical paper. The paper includes a:

- Brief review of the opposed-piston, two-stroke engine's architectural advantages (thermodynamics, pumping work and combustion)
- Comprehensive overview of the engine's performance and emissions results, including indicated thermal efficiency, fuel consumption and emissions
- Comparison of fuel economy and emissions to the published benchmark, the Cummins 2.8L ATLAS Diesel Engine [2]
- Discussion of an exhaust temperature control strategy that is used to meet the aggressive catalyst light-off requirements of light-duty applications by achieving rapid catalyst light-off after a cold start
- Comparison of engine balance of the light-duty truck concept engine and a state-of-the-art gasoline V6 engine
- Examination of the packaging options for an opposed-piston, two-stroke engine in a light-duty truck application

The results of this study show that the Achates Power opposed-piston engine benefits-high efficiency, low emissions and reduced cost, mass and complexity-already demonstrated for medium-duty commercial vehicles [1] are also available for light-duty applications. In fact, to an even greater extent: over 30% fuel economy improvement when compared to an equivalent four-stroke diesel engine.

Moreover, this study shows that the final 2025 light-truck CAFE fuel economy regulation not only has the potential to be met but also the potential to be exceeded with a full-size 5,500 lb. pick-up truck by simply applying the Achates Power technology without any hybridization or vehicle improvements.

Introduction

With the arrival of stricter emissions and fuel efficiency regulations such as Tier 3 or LEV III and CAFE requirements, substantial changes in engine technology are required to reach the targets. As a brief summary, Tier 3 or LEV III standards will be introduced in phases between 2017 and 2025. By 2025, the fleet average emissions level will be SULEV, 0.030 g/mi NMOG+NOx for all light-duty vehicles (below 8,500 lbs. GVW). It should be noted that the majority of tailpipe emissions during certification as well as many real-world driving conditions occur just after starting the engine before the catalysts reach operating temperature. In many applications, greater than 50% of the tailpipe emissions on a diesel FTP-75 test occur in the cold-start phase [3]. In fact, with a well-designed aftertreatment system, it can be shown that greater than 50% of the tailpipe emissions occur during the first 200 seconds of the test [3].

Along with these emissions reductions, CAFE standards are increasing from the original 2010 requirements of 27.5 MPG for cars and 23.5 MPG for light trucks to targets of 37.8 MPG for cars and 28.8 MPG for light trucks in 2016. For 2017-2025, passenger cars would be required to achieve 5% annual improvements, and light trucks 3.5% annual improvements. Eventually, the regulation takes CAFE to 54.5 MPG (without credits) for cars and light-duty trucks in 2025. Figure 1 below shows the evolution of the combined passenger and truck CO2 equivalent EPA fuel economy regulation. These standards on emissions and CAFE create significant challenges.

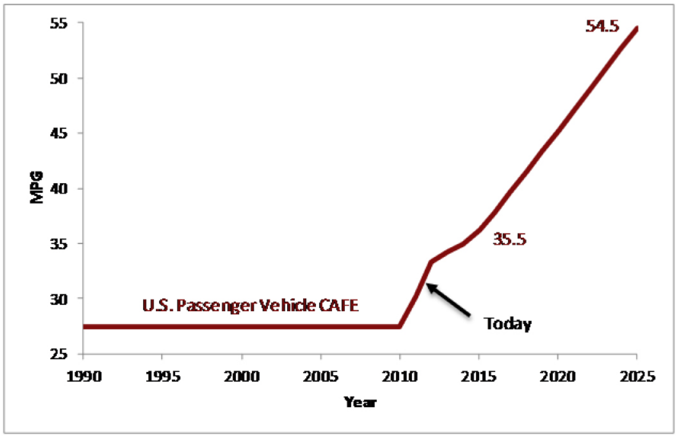

Figure 1. CAFE targets.

In response to these latest regulatory changes, it is imperative for diesel engines to reduce cold-start emissions by fast catalyst light-off as well as lower fuel consumption, all without sacrificing drivability. As will be shown in this paper, the Achates Power opposed-piston, two-stroke diesel engine configured to meet the requirements of a light-duty truck can provide a significant fuel economy advantage when compared to a four-stroke engine of similar performance.

Opposed-Piston Engine Architectural Advantages

Opposed-piston, two-stroke engines were conceived in the 1800s in Europe and subsequently developed in multiple countries for a wide variety of applications, including aircraft, ships, tanks, trucks and locomotives. They maintained their presence throughout most of the twentieth century. An excellent summary of the history of opposed-piston engines can be found in the SAE book, *Opposed-Piston Engines: Evolution, Use, and Future Applications* by M. Flint and J.P. Pirault [4]. Produced initially for their manufacturability and high power density, opposed-piston, two-stroke engines have demonstrated superior fuel efficiency compared to their four-stroke counterparts. This section examines the underlying reasons for the superior fuel efficiency and emissions.

The OP2S diesel engine has the following efficiency advantages compared to a conventional, four-stroke diesel engine:

1. Reduced Heat Losses

The Achates Power opposed-piston engine, which includes two pistons facing each other in the same cylinder, offers the opportunity to combine the stroke of both pistons to increase the effective stroke-to-bore ratio of the cylinder working volume.

For example, when coupling two piston trains from a conventional, single-piston engine with a stroke-to-bore ratio of 1.1, the resulting opposed-piston engine bore-to-stroke ratio is twice or 2.2. This can be accomplished while preserving the engine speed capability of the base design.

To achieve the same stroke-to-bore ratio with a single-piston engine, the mean piston speed would double for the same engine speed. This would severely limit the engine speed range and, therefore, the power output.

The increase in stroke-to-bore ratio has a direct mathematical relationship to the area-to-volume ratio of the combustion space. For example, when comparing a single-piston engine to an opposed-piston engine with the same piston slider dimensions, the following outcome can be seen:

Table 1. OP2S compared to a single-piston engine.

	Single Piston	OP2S
Trapped Volume/Cyl.	1.0L	1.6L
Bore	102.6 mm	102.6 mm
Total Stroke	112.9 mm	224.2 mm
Stroke-to-Bore Ratio	1.1	2.2
Compression Ratio	15:1	15:1
Surface Area (Min Vol.)	20 cm^2	20 cm^2
Volume (Min Vol.)	71 cm^3	114 cm^3
Area-to-Volume Ratio	0.28	0.18

In this example, the reduction in the surface area top volume ratio is a very significant 36%. The lower surface area directly leads to a reduction in heat transfer.

The plot on the following page shows that the area-to-volume ratio of a six-liter, opposed-piston engine is equivalent to a 15-liter, conventional diesel engine.

This reduction in area-to-volume ratio is one of the main reasons why larger displacement engines are more efficient than smaller ones. With the Achates Power opposed-piston architecture, there is the opportunity to achieve the efficiency of much larger engines.

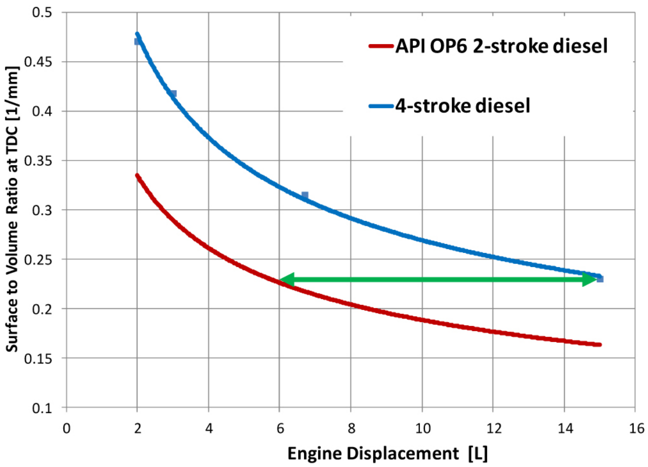

Figure 2. Surface-to-volume ratio versus engine displacement for an OP2S and conventional engine.

2. Leaner Combustion

When configuring an opposed-piston, two-stroke engine of the same displacement as a four-stroke engine-for example, converting a six-cylinder, conventional engine into a three-cylinder, opposed-piston engine-the power that each cylinder has to deliver is the same. The opposed-piston engine fires each of the three cylinders at each revolution while the four-stroke engine fires each of its six cylinders one out of two revolutions.

Therefore, the amount of fuel injected for each combustion event is similar, but the cylinder volume is twice as much for the Achates Power opposed-piston engine. So for the same boost conditions, the opposed-piston engine will achieve leaner combustion, which increases the ratio of specific heat. Increasing the ratio of specific heat increases the pressure rise during combustion and increases the work extraction per unit of volume expansion during the expansion stroke.

Ideal Engine Efficiency

$$\eta_{ideal} = 1 - \frac{1}{r_c^{\gamma-1}}$$

r_c = compression ratio

γ = ratio of specific heats

3. Faster and Earlier Combustion at the Same Pressure Rise Rate

The larger combustion volume for the given amount of energy released also enables shorter combustion duration while preserving the same maximum pressure rise rate. The faster combustion improves thermal efficiency by reaching a condition closer to constant volume combustion. The lower heat losses as described above lead to a 50% burn location closer to the minimum volume. The plot below illustrates how the heat release rate compares between a four-stroke engine and the Achates Power opposed-piston engine.

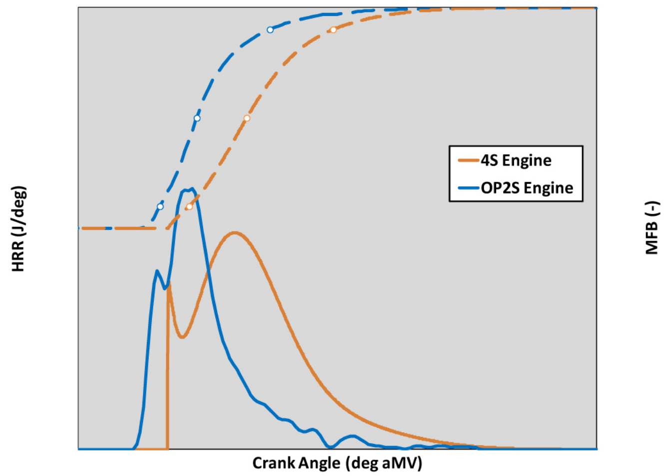

Figure 3. Heat release rate comparison between a four stroke and the OP2S.

The ideal combustion should occur at the minimum volume and be instantaneous. The opposed-piston engine is much closer to this ideal condition at the same pressure rise rate.

The aforementioned fundamental OP2S thermal efficiency advantages [5] are further amplified by:

- Lower heat loss due to higher wall temperature of the two piston crowns compared to a cylinder head. (Reduced temperature delta).
- Reduced pumping work thanks to uniflow scavenging with the OP2S architecture giving higher effective flow area than a comparable four-stroke or a single-piston, two-stroke uniflow or loop-scavenged engine [6].
- A decoupled pumping process from the piston motion due to the two-stroke architecture allows alignment of the engine operation with a maximum compressor efficiency line [7].
- Lower NOx characteristics as a result of lower BMEP requirements because of the two-stroke cycle operation [8].

Efficiency and Emissions Enablers

Combustion System

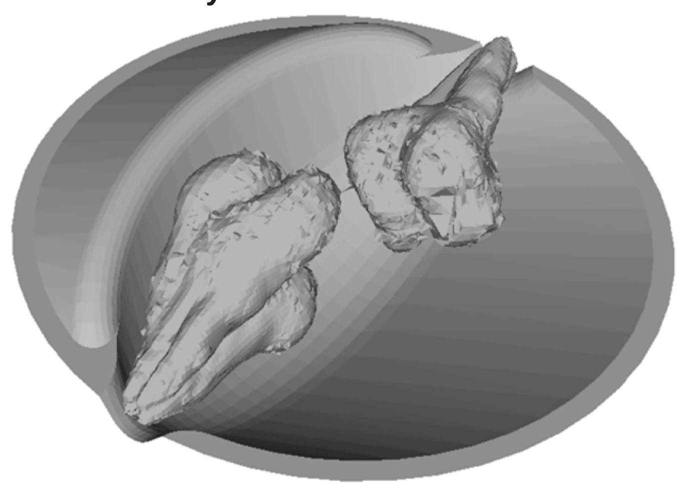

Figure 4. Schematic of the combustion system with plumes coming out of two side-mounted injectors.

Achates Power has developed a proprietary combustion system [9] composed of two identical pistons coming together to form an elongated ellipsoidal combustion volume where the injectors are located at the end of the long axis [10] (Figure 4).

This combustion system allows:

- High turbulence, mixing and air utilization with both swirl and tumble charge motion as is illustrated below with the high turbulent kinetic energy available at the time of auto ignition

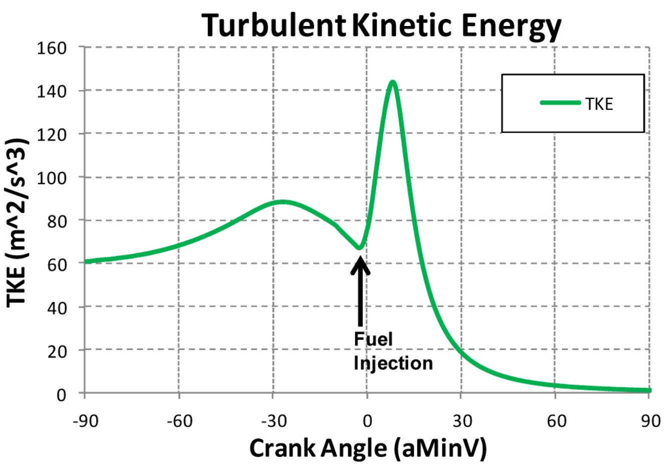

Figure 5. The OP2S allows for high turbulence, mixing and air utilization with both swirl and tumble charge motion.

- Ellipsoidal combustion chamber resulting in air entrainment into the spray plumes from two sides
- Inter-digitated, mid-cylinder penetration of fuel plumes enabling larger λ=1 iso-surfaces
- Excellent control at lower fuel flow rates because of two small injectors instead of a single higher flow rate
- Multiple injection events and optimization flexibility with strategies such as injector staggering and rate-shaping [10]

The result is no direct fuel spray impingement on the piston walls and minimal flame-wall interaction during combustion. This improves performance and emissions [1] with fewer hot spots on the piston surfaces to further reduce heat losses [10].

Air System

To provide a sufficient amount of air for combustion, two-stroke engines need to maintain an appropriate pressure difference between the intake and exhaust ports (i.e. to scavenge exhaust out of the cylinder after combustion and push in fresh air mass).

For applications that require the engine to change speed and load in a transient manner, such as automotive applications, external means of air pumping are required. Among the various possible configurations of the air system with turbocharger and supercharger combinations, the layout as described in Figure 6 is the preferred configuration [11].

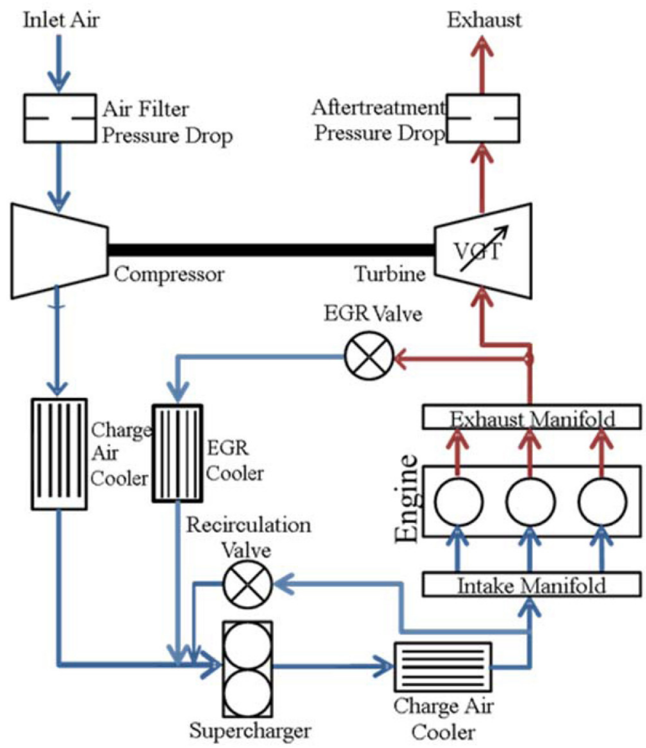

Figure 6. OP2S preferred air system layout.

Advantages of such an air system are summarized as follows:

- The compressor provides high pressure before the supercharger, which is multiplied by the supercharger. This means low supercharger pressure ratios are sufficient for high intake manifold density, reducing pumping work.
- The maximum required compressor pressure ratio is lower compared to regular turbo-only air systems of four-stroke engines.
- The use of a supercharger recirculation valve allows greater control of the flow through the engine, thus providing flexibility for precise control of emissions and for optimization across the engine map while reducing pumping.
- Lowering the flow through the engine by decreasing the pressure difference across the engine reduces the pumping penalty at low load points. This, together with having no dedicated intake and exhaust stroke for moving mass from and to the cylinder improves BSFC.
- Improved transient response due to the supercharger and recirculation valve [12].
- Very good cold start and catalyst light-off capability due to accurate control of the engine pressure difference [3].
- Increased low-speed torque by selecting the appropriate gear ratios on the supercharger [1].
- Increased capability to drive EGR with a lower pumping penalty compared to a conventional, turbocharged four-stroke engine as it is driven by a supercharger [1].
- Ability to cool both air and EGR together reduces fouling of the cooler [1][13].

Light-Duty Truck Engine Concept

An OP2S engine with 2.25L displacement producing 200 hp for light-duty applications has been studied in comparison to the DOE project that is being worked on by Cummins [2]. Table 2 shows the specification of the Achates Power 2.25L OP2S diesel engine for light-duty applications. The details of the package of this engine in a typical half ton pick-up truck are shown in Figure 7.

This OP2S engine design has two crankshafts, which are phased with respect to each other. The phasing between the exhaust and intake crankshafts can change the timing of the exhaust port opening compared to the minimum volume and the blowdown time on the exhaust ports for the same port heights. For applications, such as light-duty vehicles, it may be desirable to adjust this phasing at different speeds and loads. The variable crank phasing mechanism can also provide variable compression and expansion ratios, useful for applications requiring good fuel economy at a lower speed and load, while still requiring high power density. This has been implemented in the study.

Turbocharger and supercharger selections for OP2S engines are just as important as any other engine for achieving good fuel economy, transient response and altitude or high temperature ambient condition operation. For the higher speed range applications with emphasis on low speed, low load and low emissions points like the light duty, the variable geometry turbine provides considerable advantages and was applied in this study in conjunction with a two-speed supercharger.

Table 2. OP2S engine configuration for light-duty truck.

Cylinder	Inline 3
Number of Pistons	6
Number of Injectors	6
Swept Volume/Engine (L)	2.25
Bore (mm)	75.75
Stroke (mm)	166.65
Stroke/Bore Ratio(-)	2.2
Nominal Power (kW@RPM)	150@3600
Max. Torque (Nm@RPM)	500@1600-2100
Emission Standard	US 2010/Euro 6

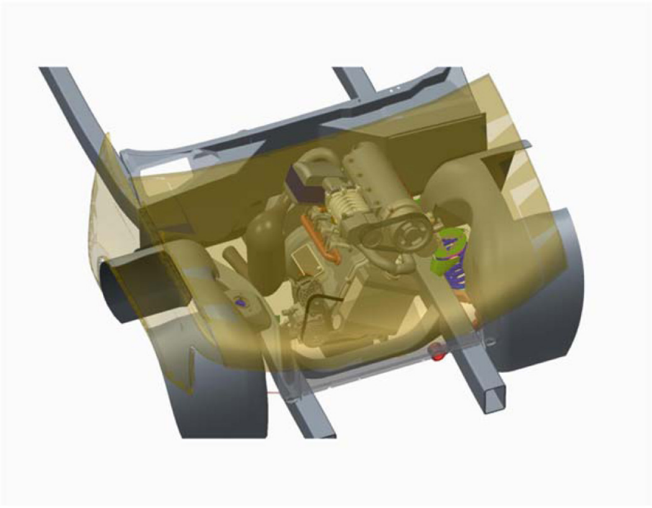

Figure 7. CAD showing OP2S packaged in light-duty truck.

Single-Cylinder Testing

The custom, single-cylinder research engine, shown in Figure 8, 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 1.64L. The liner geometry creates fixed port timing, and the piston geometry and injection spray pattern have been specified based on combustion simulation results. The common-rail fuel injection system is capable of injection pressures up to 2200 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 9. 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 8. 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. An FTIR as well as a California Analytical Instruments (CAI) emissions analyzer are used to measure the steady-state concentration of five exhaust species (CO_2, CO, O_2, HC, NOx) and intake CO_2. 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.

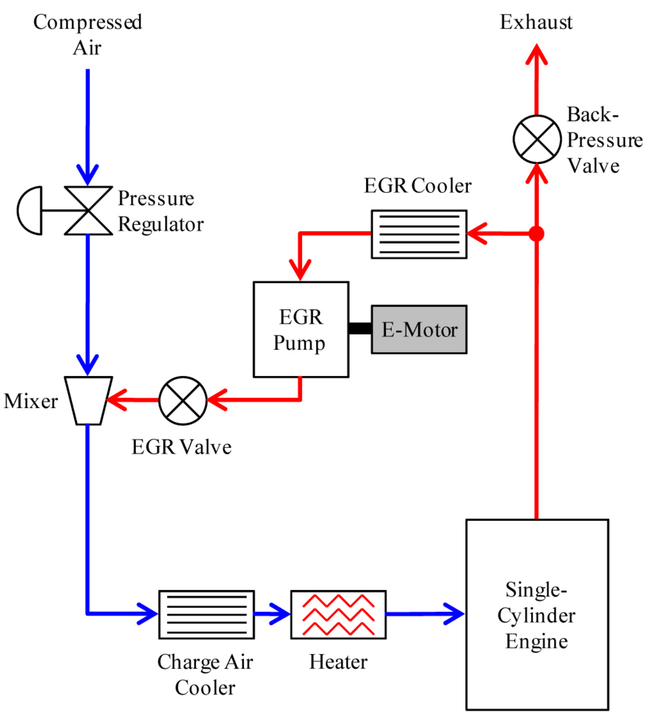

Figure 9. Schematic of the air and EGR conditioning system.

Single-Cylinder Results

More than 30 points were measured on the single-cylinder engine and were used as inputs to the multi-cylinder model. Balancing the trade-offs of emissions (NOx, Soot, HC and CO), combustion noise and maximum rate of pressure rise, temperatures, and efficiency were factors considered in the optimization process. The points are shown in Figure 10.

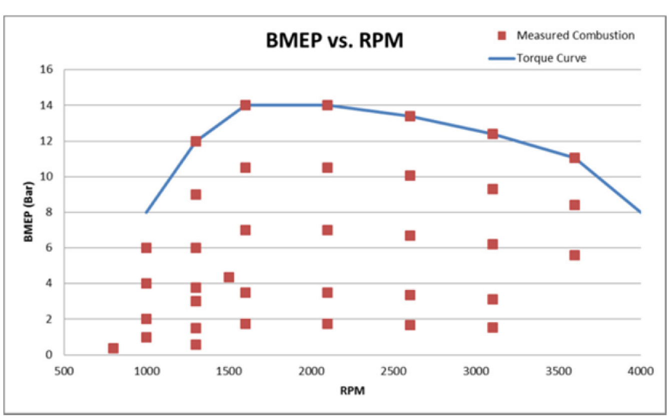

Figure 10. Measured combustion input for 1D model.

A 1300rpm 0.57Bar BMEP point produced a 49% ITE with a 1.02g/kWh ISNOx and a 0.003g/kWh ISSoot. The maximum pressure rise rate (MPRR) was 1.5Bar/degree. The AVL combustion noise was 79dB.

A 1300rpm 1.5Bar BMEP point produced a 50% ITE with a 0.92g/kWh ISNOx and a 0.013g/kWh ISSoot. The MPRR was 3.6Bar/degree. The AVL combustion noise was 84dB.

One of the key weighted points for the project is 1500rpm and 4.35Bar BMEP (on a multi-cylinder, IMEP target was used on the single cylinder). This point has an ITE of 50.3% with a 0.85g/kWh ISNOx. This point was one that a gravimetric PM measurement was taken and the results are 0.105 g/kWh ISPM with a breakdown of 0.070g/kWh ISEC (Elemental Carbon) and 0.035g/kWh ISOC (Organic Carbon). The pie chart in Figure 11 below shows the indicated specific fraction split between Elemental and Organic Carbon. An AVL415S smoke meter was also used and yielded an ISSoot value of 0.145g/kWh. This shows that using the AVL 415S approximates the soot on the high side by ~38% higher than the total PM by gravimetric measurement, making soot calculations using this method conservative. Fuel Specific Oil Consumption (FSOC) was also taken at this point and was 0.046% of fuel.

Another key data point taken was 1300rpm and 3.78Bar BMEP (on a multi-cylinder, IMEP target was used on the single cylinder). This point has an ITE of 49.5% with a 0.85g/kWh ISNOx. The gravimetric PM measurement was taken and the results are 0.074 g/kWh ISPM with a breakdown of 0.051g/kWh ISEC (Elemental Carbon) and 0.023g/kWh ISOC (Organic Carbon). An AVL415S smoke meter was also used and yielded an ISSoot value of 0.103g/kWh. The pie chart in Figure 11 below shows the indicated specific fraction split between Elemental and Organic Carbon. Once again, this shows that using the AVL 415S approximates the soot on the high side by ~39% higher than the total PM by gravimetric measurement, making soot calculations using this method conservative. Fuel Specific Oil Consumption (FSOC) was also taken at this point and was 0.027% of fuel. For more details on oil consumption, please refer to the published technical paper [14].

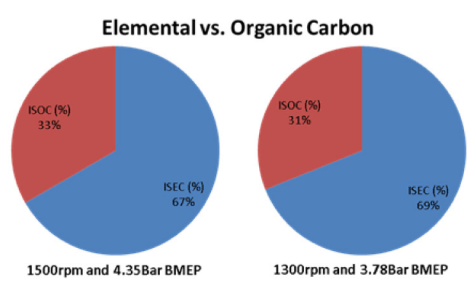

Figure 11. Breakdown of elemental vs. organic carbon

Figure 12 shows the heat release profile for a common cruising point of 1600 rpm and 25% load. This point has an ITE of 50.8% with a 0.83g/kWh ISNOx and a 0.07g/kWh ISSoot. One can see that the combustion is very fast, with a 10-90 Burn Duration of 15 degrees CA.

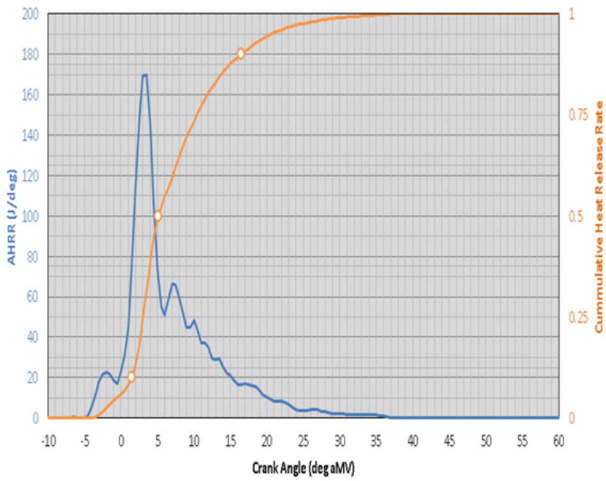

Figure 12. 1600 rpm 25% load heat release profile.

Figure 13 shows the heat release profile for a peak torque point of 1600 rpm and 100% load. This point has an ITE of 50.5% with a 4.9g/kWh ISNOx and a 0.03g/kWh ISSoot. Once again, combustion is fast with a 10-90 Burn Duration of 24 degrees CA.

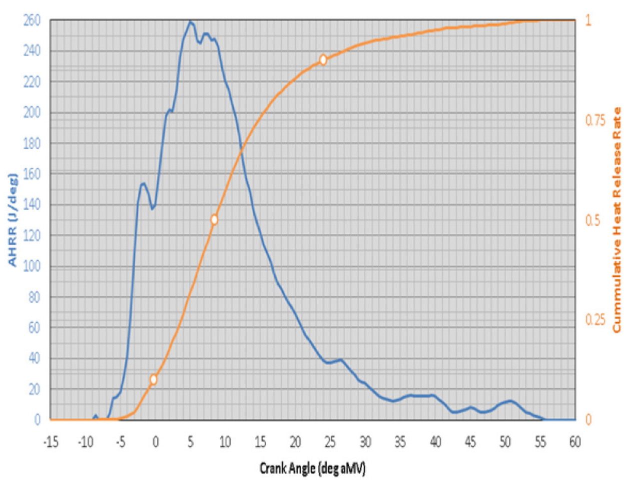

Figure 13. 1600 rpm 100% load heat release profile.

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, a two-step process was used with two independent 1D interface models, created in 1D engine system simulation software. The first step was using a single-cylinder model matching the test cell hardware. The combustion chamber geometry, the piston motion and the porting profiles are identical to what exists in the single-cylinder engine. Engine speed, fuel flow rate, air flow rate, EGR percentage, cylinder pressures at 30° before minimum volume, and intake and exhaust manifold 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. From this, the trapped conditions in the cylinder are determined as well as the in-cylinder heat transfer coefficient.

These values are then used as inputs to a 1D multi-cylinder engine system model. This model is used to provide multi-cylinder-based predictions of the friction and pumping work required conditions 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 14 shows the schematic of the input data and assumptions of the interface 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.

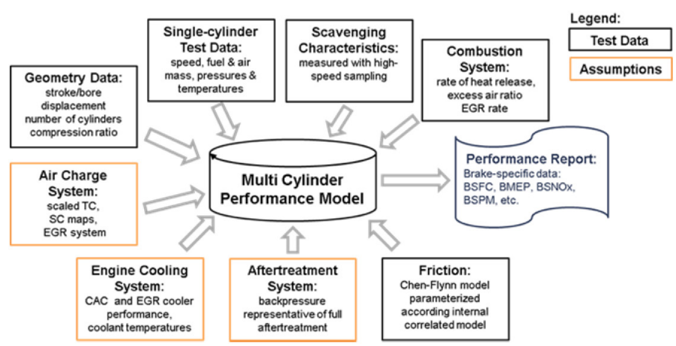

Figure 14. Multi-cylinder interface model input data flow.

The interface model air-handling system (Figure 6) consists of a supercharger, a turbocharger, a charge air cooler after each compression stage and an EGR cooler. 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 in conjunction with a variable geometry turbine, which also provides back pressure control. Further back pressure control is available from a back pressure valve, which is mainly used for aftertreatment catalyst warm-up.

EGR is introduced into the intake system after the compressor and 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, while the EGR cooler is on the main engine temperature coolant circuit. The charge air cooler's effectiveness values are set to 85%, which assumes a typical degree of cooler fouling. 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, and the scavenging efficiency versus scavenge ratio relationship was used in the interface model correlation process.

Fuel Economy and Emissions

Results

With the single-cylinder combustion data as inputs to the multi-cylinder GT-Power 1D model, the following maps represent the results:

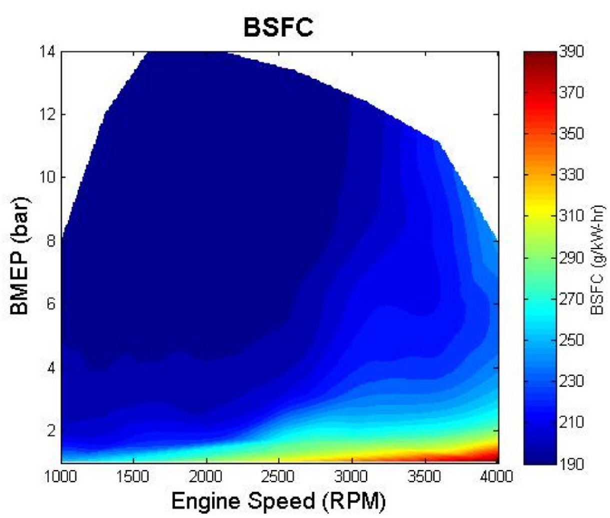

Figure 15. Multi-cylinder BSFC map.

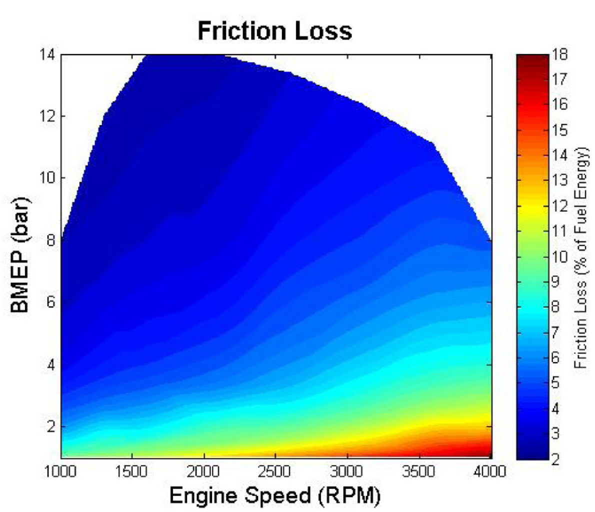

Figure 16. Multi-cylinder friction loss map.

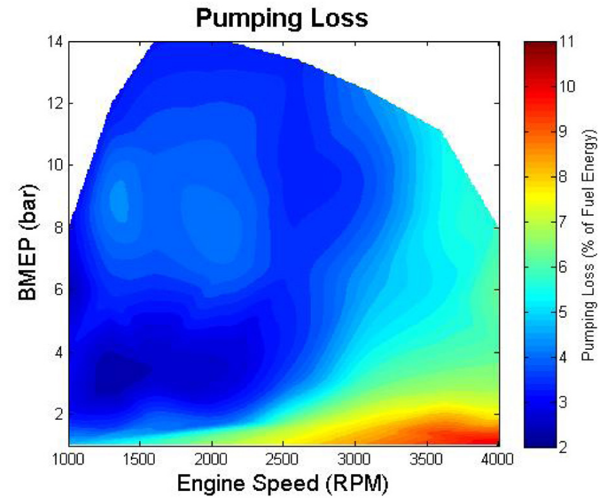

Figure 17. Multi-cylinder pumping loss map.

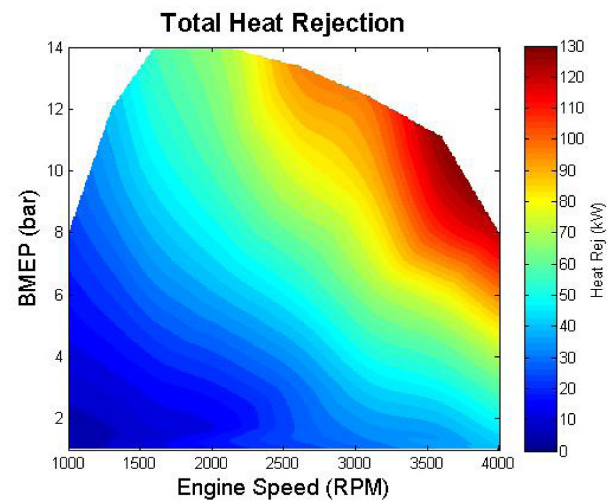

Figure 18. Multi-cylinder heat rejection map.

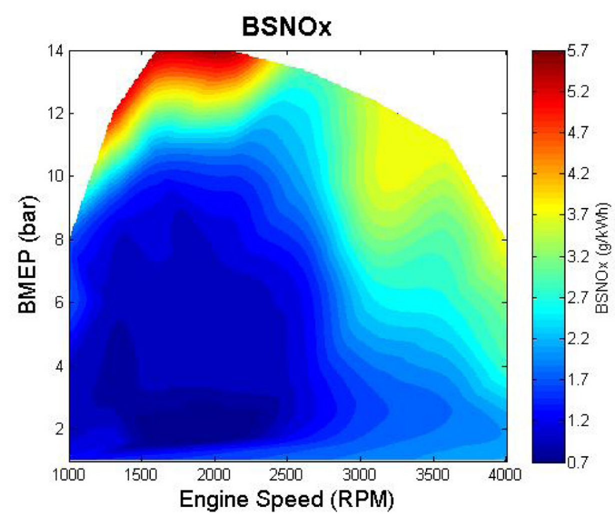

Figure 19. Multi-cylinder BSNOx map.

Figure 15 shows the full engine map for BSFC. The map is extremely flat compared to a conventional, four-stroke engine, reducing the need for increasing the number of speeds in the transmission and reducing the number of required shifts in a particular drive cycle. The best point efficiency is greater than 44% BTE, which occurs at full load from 1600-2100 rpm.

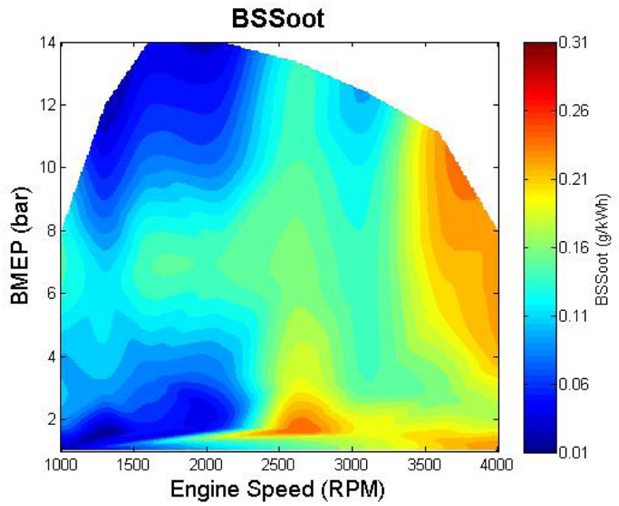

Figure 20. Multi-cylinder BSSoot map (from AVL415S).

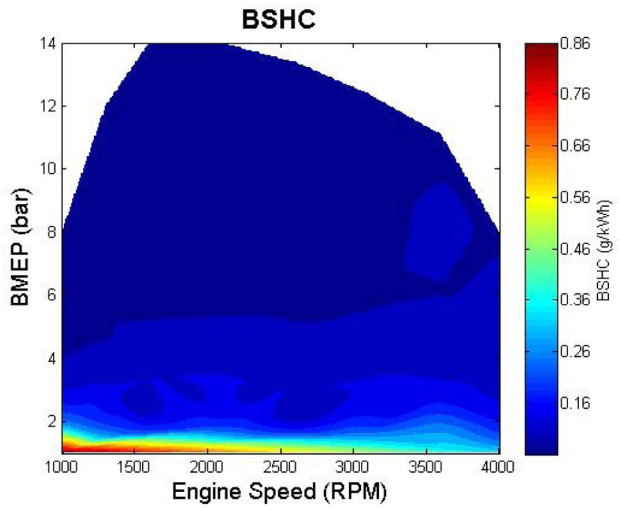

Figure 21. Multi-cylinder BSHC map.

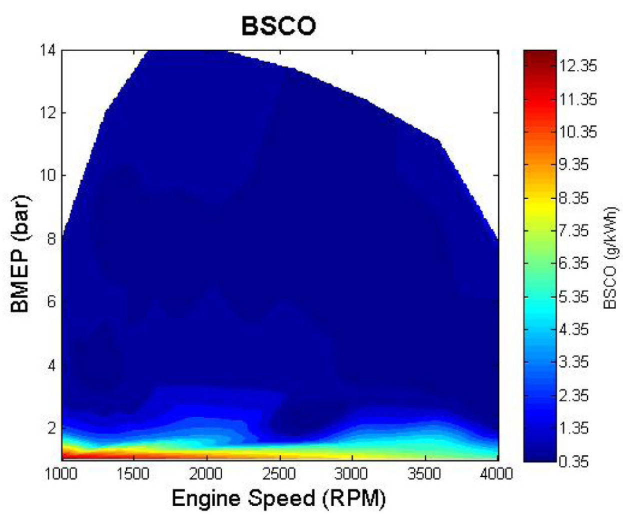

Figure 22. Multi-cylinder BSCO map.

Being a light-duty application, the EGR rate is quite high at the lower speeds and loads in order to have very low engine-out NOx emissions of ~1.0g/kWh during most of the FTP and highway cycles. This is shown in Figure 19.

Figures 23 and 24, respectively, show 40 operating points on the turbocharger compressor map and the supercharger compressor map. Unlike typical conventional, four-stroke engines, the OP2S engine does not need as wide of a compressor map. Therefore, achieving good efficiency at rated power, while still maintaining a good surge margin at peak torque and other low-speed, high-load points, is not as difficult.

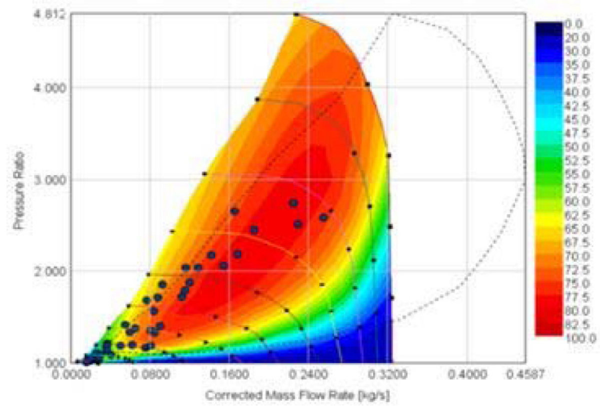

Figure 23. Turbocharger compressor map with operating points.

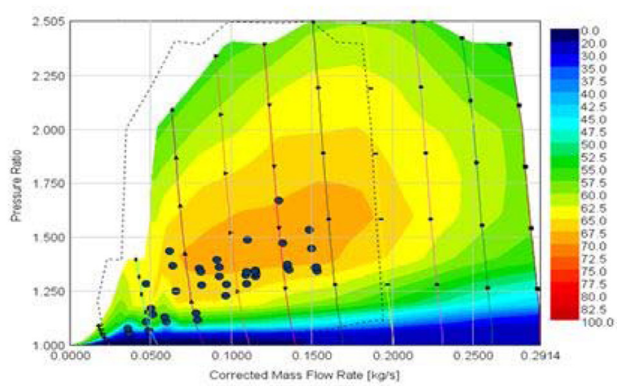

Figure 24. Supercharger compressor map with operating points.

Comparison to DOE Project

Cummins was generous to share the 10-mode points that are being used to determine the predicted cycle fuel economy and emissions on their light-duty ATLAS project. These are 10 steady-state points with weighting factors used to predict the transient cycle results. The vehicle that was used to generate these load points is the 2010 Nissan Titan full-size pick-up truck equipped with an eight-speed transmission tested at 5,500 lbs.

Four of the more heavily weighted of the 10 points were run on the single-cylinder engine to generate combustion data as input for the multi-cylinder model. The other six points used combustion profiles from the 30 points mentioned in the previous section, which were very similar in speed and load but not exact for that point. The results from the predicted 1D model were then input into the calculator with the weighting factors. The results shown in Table 3 are for the LA-4 cycle and the highway fuel economy cycle results are shown in Table 4. A significant improvement in fuel economy, NOx and Soot was achieved. The NOx, PM (using an AVL415S) and HC levels achieved with the OP2S engine meet the engine-out target that

Cummins estimated would allow this vehicle to achieve the fully phased-in Tier 3 or LEV III emissions with the proper aftertreatment selection. The LA4 and HFET cycle results are uncorrected fuel economy cycle values.

Table 3. LA-4 cycle (Engine-out emissions). *API OP6 only measuring soot with AVL415S and not the total PM.

Cycle	LA4			
Parameter	Fuel Econ	NOx	PM	HC
Unit	MPG	g/mile	g/mile	g/mile
Cummins Atlas	26.7	0.82	0.13	-
API OP6	34.1	0.47	0.03*	0.12 (THC)
% Improv.	28%	42%	74%	-

Table 4. Highway fuel economy cycle (Engine-out emissions). *API OP6 is only measuring soot with AVL415S not total PM.

Cycle	HFET			
Parameter	Fuel Econ	NOx	PM	HC
Unit	MPG	g/mile	g/mile	g/mile
Cummins Atlas	34.4	0.94	0.09	0.10 (NMHC)
API OP6	45.7	0.34	0.04*	0.12 (THC)
% Improv.	33%	63%	55%	-16%

The significant fuel economy advantage of over 30% that is being demonstrated in this exercise is more significant than what we have shown in other comparisons [1] for several reasons.

The first one is due to the fact that the engine operating range for these fuel economy cycles is focused on low speed and low load. The Achates Power engine's ability to manage pumping losses and maintain high indicated efficiencies at low loads leads to a greater advantage over a four-stroke engine in this operating range.

The second one is due to the fact that the engine-out NOx requirements for light-duty applications like this one are much lower than for heavy-duty applications. Once again, the charging system flexibility of the Achates Power engine enables high levels of EGR with a minimal pumping loss. At low-load operation, internal EGR can be used more extensively and actually lowers the pumping requirements. The lower the engine-out requirements, the greater the advantage of the Achates Power engine over conventional engines.

The FTP-75 was estimated by using the FE relationship that was presented in the Cummins paper [2] between the LA-4 and the FTP-75. The estimated city label is based on the estimated FTP-75 fuel consumption, corrected for the 5-cycle: 5-cycle city FE = 1 / (0.003259 + 1.18053 / FTP FE) and the highway calculated as such: 5-cycle highway FE = 1 / (0.001376 + 1.3466 / HFET FE). The combined FE is 55% weighted city and 45% weighted highway labels. We are thus able to estimate what the fuel economy labels would be for this vehicle equipped with an API OP6 engine using the proper coefficients. The CAFE FE is a straight 55% and 45% from the estimated raw FTP and HWY cycle FE estimates.

Table 5. Estimated fuel economy labels and CAFE.

Cycle	Estimated City Label	Estimated HWY Label	Estimated Combined Label	CAFE Combined
Parameter	Fuel Econ	Fuel Econ	Fuel Econ	Fuel Econ
Unit	MPG	MPG	MPG	MPG
API OP6	25	32	28	37

One can see from the results in Table 5 that the estimated combined label fuel economy of this full-size pick-up truck with the API OP6 engine of 28 MPG would be similar to what a much smaller vehicle achieves today with a conventional powertrain.

With respect to the CAFE fuel economy, we have estimated that a full-size pick-up truck like the Nissan Titan would need to achieve about 33 MPG to comply with the final phase-in in 2025. With a 37 MPG combined fuel economy, this vehicle configured with the Achates Power engine could be used to offset other vehicles that cannot comply.

This is accomplished with a conventional drivetrain layout and current production vehicle technologies that do not leverage any hybridization or vehicle improvements.

Catalyst Light Off and Temperature Management

It is well known that to pass the FTP-75 cycle it is extremely important to have very low emissions at the start of the cycle and heat the exhaust aftertreatment as fast as possible.

The strategy of a warm-up mode has proved to be the most effective solution to date (using different engine settings after a cold start to quickly increase exhaust temperatures and achieve the fastest catalyst light-off time). Once the catalyst is lit, the engine-out emissions level can be relaxed and the engine settings adjusted to optimize for fuel consumption. It should be noted that catalyst technologies are being developed that improve emissions conversion efficiency at lower temperatures. These new technologies promise to further reduce the minimum exhaust temperature requirement and enable less compromised and more fuel-efficient engine designs and calibrations.

Gasoline engines have historically relied on cold-start strategies that generate significant heat immediately after starting, such that the catalyst achieves very high emissions conversion efficiency within 10 to 20 seconds [3]. With conventional diesel engines, the situation is not as favorable; the trapped conditions required for stable diesel combustion limit the potential for achieving high exhaust temperatures. In contrast, the Achates Power opposed-piston, two-stroke diesel engine can generate trapped conditions to achieve stable combustion and high exhaust temperatures simultaneously without specific additional hardware. As a result, both the combustion temperature and, therefore, the exhaust gas temperature can be tailored to satisfy the requirements of the aftertreatment system for faster light-off. The proposed approach allows the opposed-piston diesel engine not only to overcome the traditional disadvantage of diesel versus gasoline engines with respect to catalyst light-off time, but also to provide superior fuel efficiency over a four-stroke diesel engine [1][3].

There are other driving conditions under which this type of strategy is expected to provide benefits. During more extreme cold starts (ambient temperatures below 0° C), good combustion stability and hydrocarbon (white smoke) control can be achieved due to elevated combustion temperatures. Good combustion stability and aftertreatment performance during low ambient temperature operation as well as light-load operation is of growing concern, particularly in low emissions regions. The ability to maintain sufficient exhaust gas temperatures under all speed and load conditions for active Diesel Particulate Filter (DPF) regeneration would be another benefit of this approach. Additional concerns are now surfacing with aftertreatment diagnostic requirements regarding the in-use rate or performance ratio, which has been increased to 30% for 2013 On Board Diagnostics (OBD).

In order to quantify the effect of the proposed operating strategy on exhaust temperatures, a series of tests was performed on a single-cylinder engine (already warmed up and at steady-state conditions) to validate the concept. Results were subsequently extrapolated to post-turbine steady-state conditions via the same multi-cylinder model used previously. Thus, this is the same hardware that produced the flat BSFC map in Figure 15 with a best-point BTE of 44.2%. This hardware-when used to generate high exhaust enthalpy for warm-up-is predicted to be able to achieve exhaust temperatures greater than 340°C at the turbine outlet (inlet to exhaust aftertreatment) at 1100 rpm and 0.5 Bar BMEP. At the same time, the NOx flow-rate was 0.37mg/s. Table 6 has a summary of this warm-up point on a warm engine at steady state. This same method can be applied to a cold engine. While balancing the trade-offs, it is an effective solution to a cold-start strategy. For more details on the method of this strategy, please reference the previously published paper [3].

Table 6. Warm-up point summary.

	Units	Warm-up Point
Speed	rpm	1100
Brake Power	kW	2.1
NOx	ppm	24
NOx Flow Rate	mg/s	0.37
AVL FSN	(FSN)	1.1
CO Flow Rate	mg/s	11
HC Flow Rate	mg/s	0.33
Exhaust Temp Post Turbine	degC	341
Total Exhaust Massflow	kg/h	36.1
Specific Exhaust Enthalpy	kJ/kg	346
AVL Noise	(dB)	80
COV (IMEP)	()	0.03

The flexibility of the OP2S engine to operate in a way to create high exhaust temperatures in one mode and then offer incredible BSFC in a normal operating mode is an excellent feature that far surpasses current conventional, four-stroke engines.

NVH

With any new engine design intended for the passenger car and light truck market, careful consideration of the inherent NVH characteristics is required. The OP2S three-cylinder architecture described here can be thought of as two separate three-cylinder crank-slider mechanisms.

As with any conventional three-cylinder, the angular spacing of each crankpin is 120°. As a result, the piston forces within each crank-slider are almost entirely cancelled. When one piston is generating a peak tensile load at the limit of its stroke, the other two pistons are each generating a compressive load that is almost exactly one-half of the magnitude of the first piston.

F1 is proportional to $\cos(\alpha)$
F2 is proportional to $\cos(\alpha+120°)$
F3 is proportional to $\cos(\alpha+240°)$

$$\cos(\alpha) + \cos(\alpha+120°) + \cos(\alpha+240°) = 0$$

The reason that the forces are not entirely cancelled is that the piston motion is not exactly sinusoidal. However, the variation between sinusoidal motion and the actual piston motion is so small that the residual (uncancelled) forces may be neglected, as shown by Figures 27 and 28.

Although the forces cancel each other out, they do not act within the same plane. For example, if the inertia of the piston in the front cylinder is generating a tensile force in the connecting rod, the compressive forces from the other two pistons are acting at a different point along the crank axis. This axial offset results in a torque, or moment, that is not internally cancelled within the crank-slider system.

A unique benefit of the OP2S three-cylinder architecture is that the majority of the moments inherent to a conventional three-cylinder are cancelled internally by the second crank-slider mechanism. This is because the second mechanism generates similar moments to the first, but with an opposite vector. Since the exhaust crankshaft is phased slightly relative to the intake, a small portion of the moments will remain.

To determine how substantial the remaining moments are, a simplified kinematic model was created in CREO/Mechanism (Figure 25). This model was set up to calculate all of the forces acting on the cylinder block that result from component mass and motion. The mechanism was analyzed at idle speed (800 rpm) and rated power speed (3600 rpm), with an exhaust crankshaft lead angle of 0° and 8°, respectively.

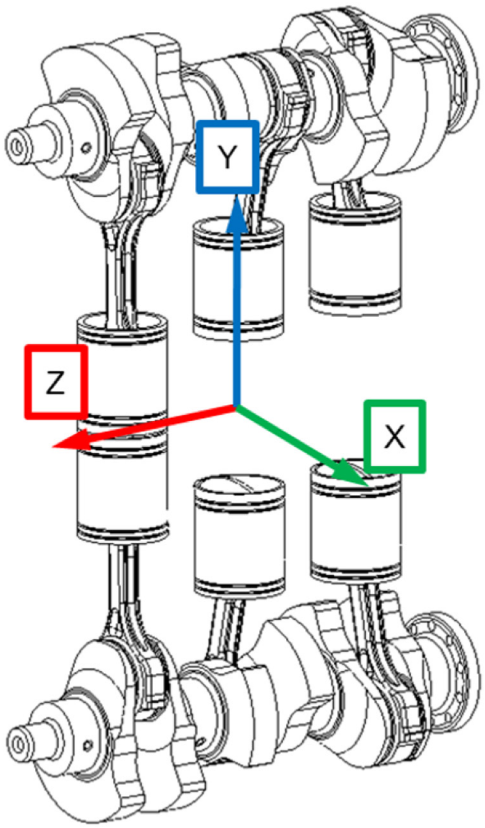

Figure 25. 2.25L OP2S three-cylinder mechanism geometry.

With the force and moment results for this configuration available, a benchmark data point was necessary. The Cummins ATLAS 2.8L four-cylinder would not provide a useful comparison because the balance shafts effectively cancel all forces and moments internally. The closest appropriate benchmark for this study was deemed to be a modern passenger car 60° V6. This was chosen because its engine vibration signature has similar characteristics (moments instead of forces) when compared to the OP2S three-cylinder, and because it is a common layout in modern passenger car and light-duty truck design.

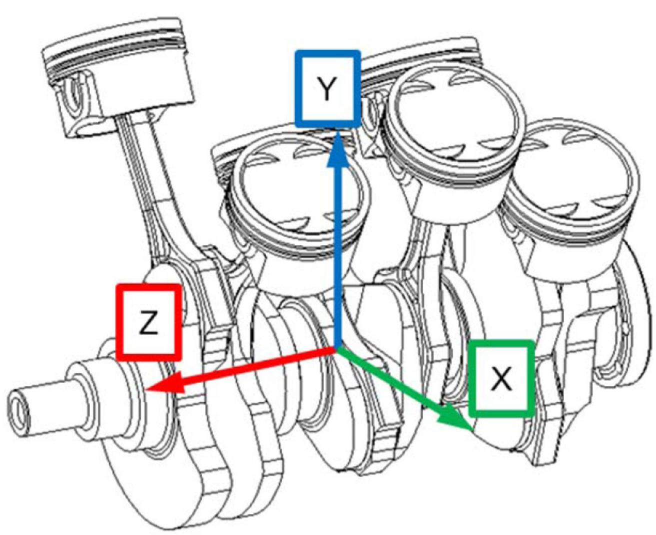

Figure 26. 3.5L 60° V6 mechanism geometry.

The specific 60° V6 chosen for analysis was the award-winning Honda 3.5L SOHC, which is used in a number of applications within the automaker's lineup. The engine rotating and reciprocating geometry was approximately modeled in CREO, based on publicly available data (Figure 26).

Again, the forces and moments were calculated within CREO/ Mechanism for idle and peak power engine speed.

A comparison of the results (Figure 27) shows that the magnitude of the OP2S engine vibration forces are significantly lower than those of a modern, four-stroke gasoline 60° V6. Forces in the X direction for both architectures are completely cancelled, so they are not plotted here. In order to clearly show the behavior of the OP2S engine, Figure 28 re-scales the data from Figure 27.

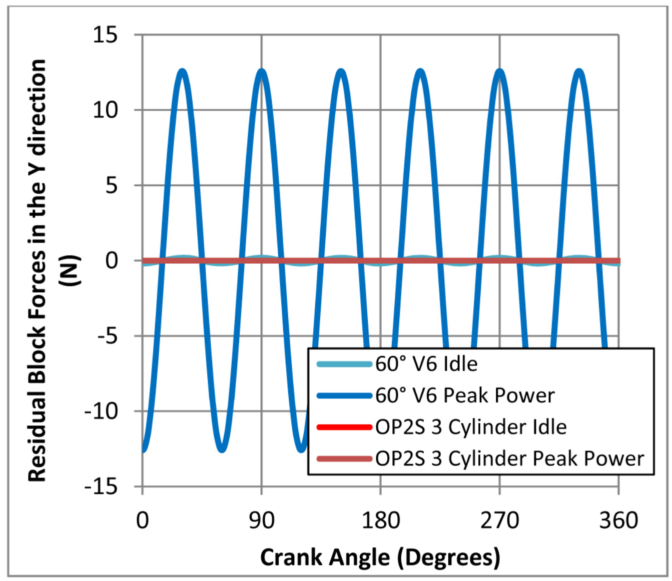

Figure 27. Comparison of the vibration force.

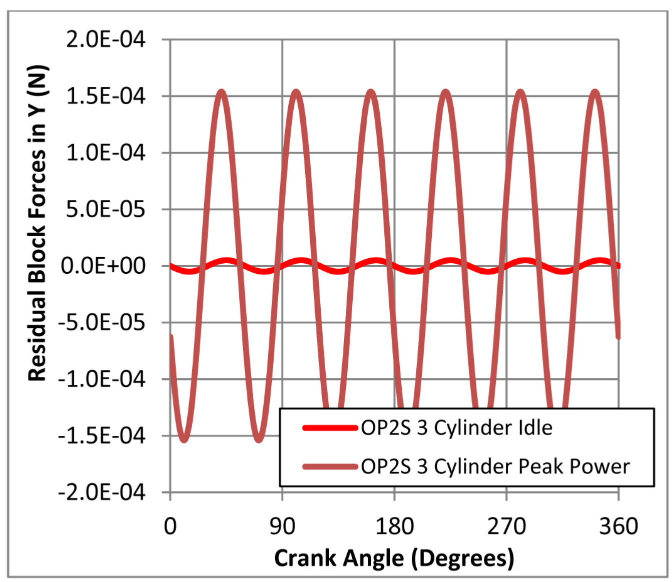

Figure 28. OP2S three-cylinder vibration force detail.

Although the force analysis provides an interesting comparison, the moments are the loads that dominate the vibration characteristics of each of these engine architectures. The results of the moment calculations are shown in Figures 29 and 30. As with the forces, the magnitudes of the 60° V6 moments at idle exceed those of the OP2S at rated power. This study shows that even though the vibration moments of the OP2S three-cylinder are not perfectly balanced, they are several orders of magnitude below the vibration moments of a world class 60° V6. The behavior of the OP2S engine vibration is re-scaled in Figure 31 to demonstrate the characteristic curve. For a given geometry, the peak moment magnitude will change linearly as a function of crank lead angle, and exponentially as a function of engine speed.

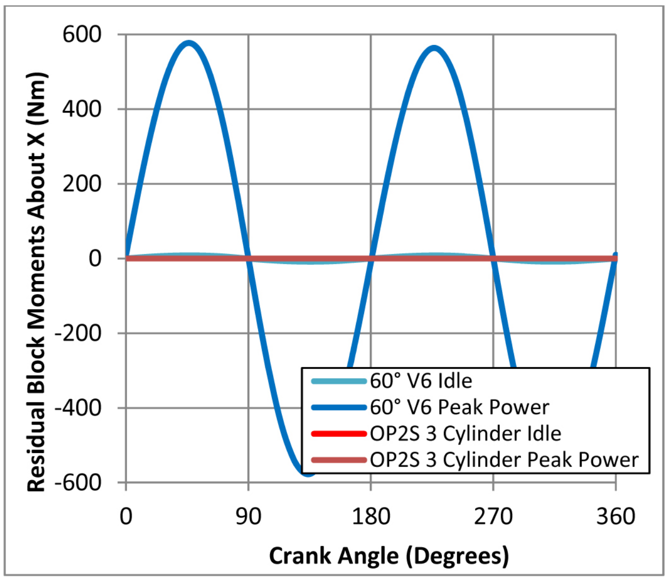

Figure 29. Comparison of the vibration moment about the X-axis.

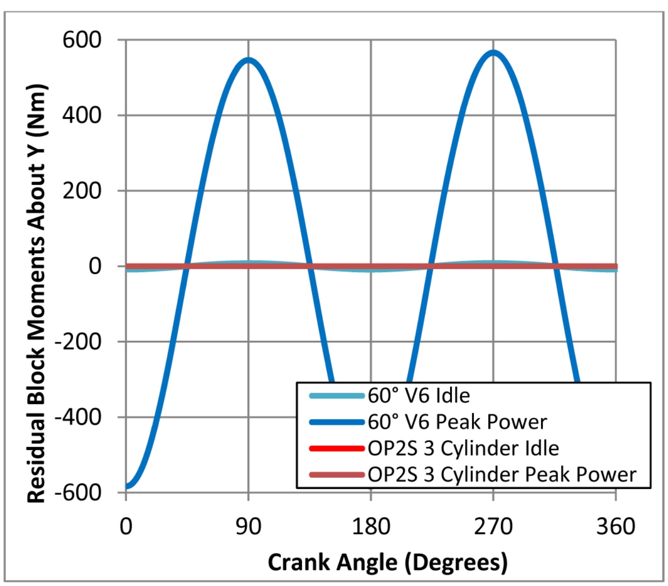

Figure 30. Comparison of the vibration moment about the Y-axis.

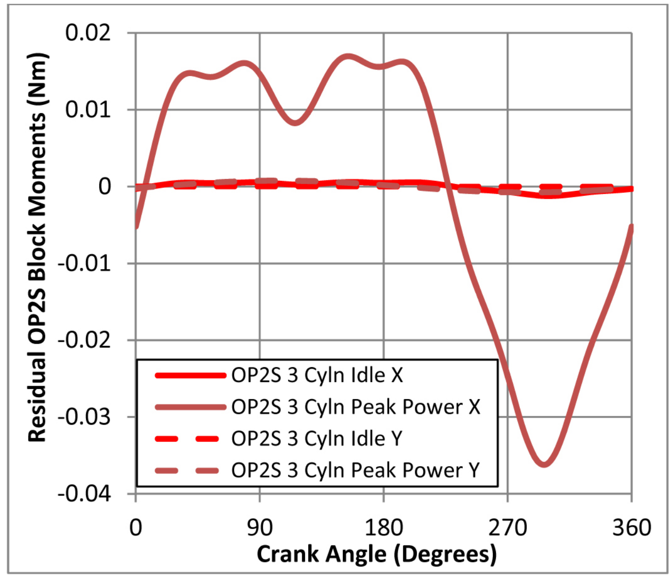

Figure 31. OP2S three-cylinder vibration moment detail.

Note that the V6 model has been counterweighted to minimize the magnitude of the moments. More or less mass eccentricity on the outermost counterweights would increase either the X- or the Y-axis moments. Further reduction in overall peak magnitude without balance shafts is not possible because of the 2nd order form that these minimum vibration curves exhibit. Because of the detailed engine mount design, the actual 60° V6 may exhibit higher vibration moment magnitudes than shown here.

The torsional vibration characteristics and reaction torques were not studied in detail here. For a rough comparison, the OP2S will have a firing event twice as often as a four stroke at the same engine speed with the same number of cylinders. That means that a four-stroke, six-cylinder operating at the same engine speed would have a torsional excitation frequency equivalent to an OP2S three cylinder. When compared to a four-cylinder, four-stroke operating at the same engine speed and brake torque, an OP2S three cylinder will likely have lower peak excitation torque amplitudes. A full system study-including flywheel(s) and harmonic balancer(s)-is necessary to characterize the details of this behavior, as is typical for any conventional powertrain.

Packaging

The OP2S engine was evaluated for packaging into a light-duty pickup truck platform. There were no apparent issues. Figures below show the results. While these are not customer specific, they provide great confidence in the ability of this engine to package well in this type of platform.

Figure 32. OP2S three-cylinder engine, left-side view.

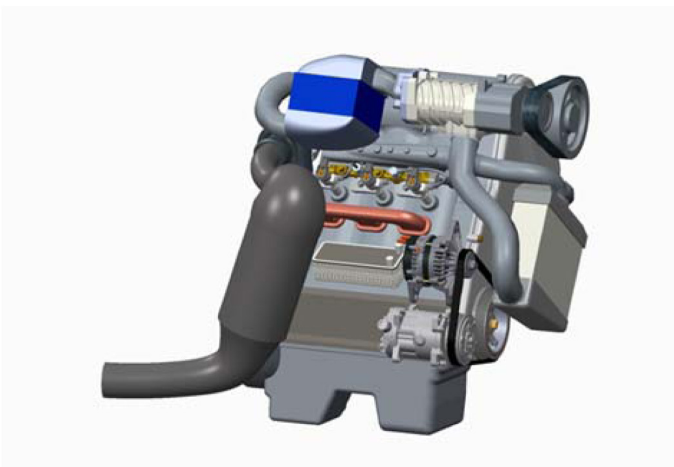

Figure 33. OP2S three-cylinder engine, right-side view.

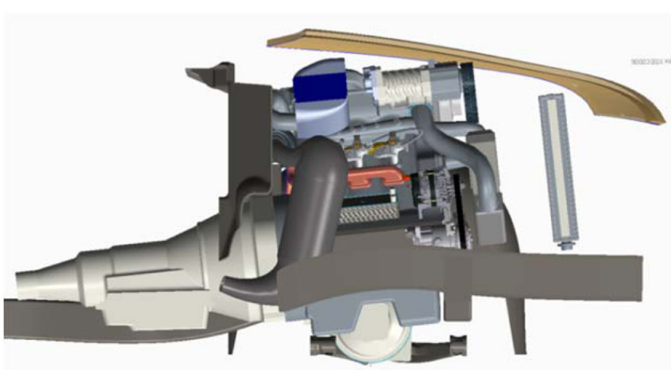

Figure 34. Side view engine in vehicle.

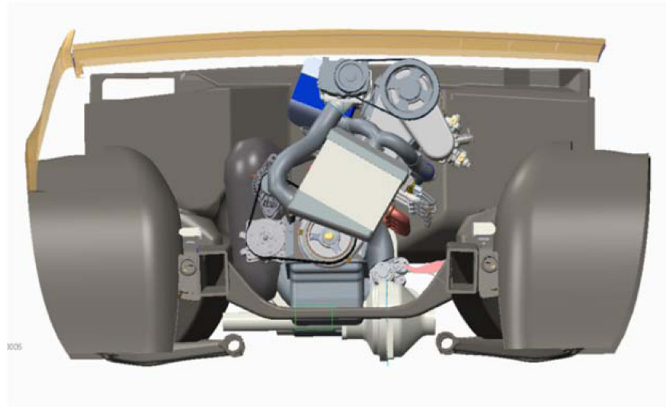

Figure 35. Front view engine in vehicle

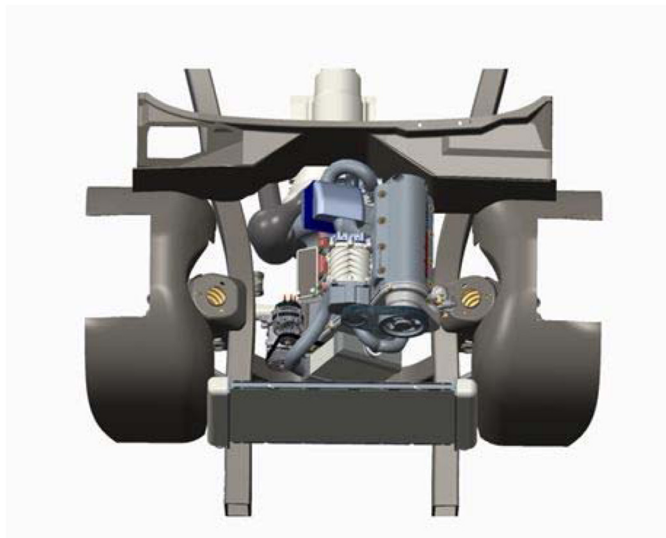

Figure 36. Top view engine in vehicle

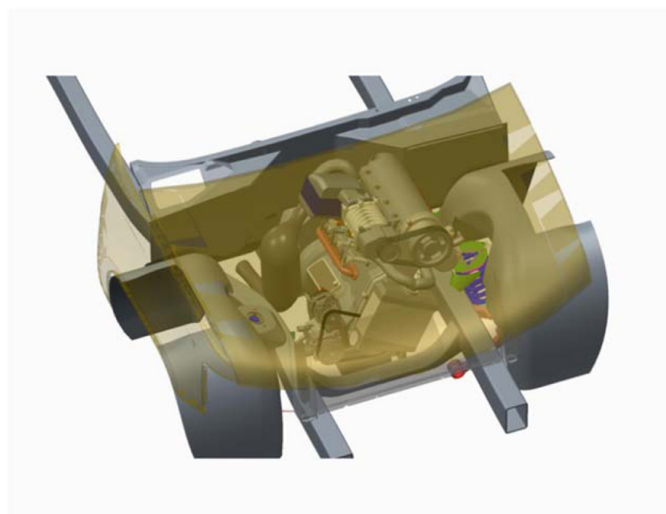

Figure 37. OP2S engine in vehicle.

Summary/Conclusions

The Achates Power opposed-piston, two-stroke engine-when configured to meet the requirements of a light-truck application-once again is able to meet all of the critical attributes and provide a very significant improvement in fuel economy.

This technology delivers an even greater fuel economy advantage in a light-duty application than in a heavy-duty application [1]. The lower the duty cycle load and the lower the engine-out NOx level, the greater the fuel economy advantage the Achates Power engine delivers.

The Achates Power engine configured for this work not only shows the potential for a 30% fuel economy improvement over the equivalent performance Cummins ATLAS engine, but also the potential to achieve the engine-out emissions targets necessary to meet the fully phased-in Tier 3 or LEV III emissions with the appropriate aftertreatment. We have also shown that this technology has excellent exhaust temperature management capabilities to manage catalyst warm-up requirements.

This engine technology enables some of the largest and heaviest vehicles that are subject to the CAFE regulation to not only comply with the final 2025 regulation levels without hybridization or vehicle improvements, but to even provide credits for other, less efficient vehicles.

Finally, this light-duty engine concept exhibits excellent vibration characteristics and packages in a typical full-size, light-duty truck for simple and transparent integration in standard vehicles.

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