Multi-Cylinder Opposed-Piston Engine Results on Transient Test Cycle

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Abstract

After having tested basic transient maneuvers such as load-step changes on the 4.9L three-cylinder opposed-piston diesel engine [1], a similar test-engine was subjected to a more aggressive test-routine - a hot-start heavy-duty FTP (Federal Test Procedure) transient cycle for the on-road engines. The three main objectives of this exercise were:

1. To assess the ability of the engine to meet the transient cycle requirements while maintaining close to the cycle-average BSFC for the FTP cycle derived from steady-state torque-to-fuel map.
2. To attain engine-out brake-specific emission levels that are compatible with US2010 EPA requirements with a conventional after-treatment system consisting of a diesel oxidation catalyst (DOC), a diesel particulate filter (DPF) and a selective catalyst reduction (SCR) system.
3. To compare hot-start FTP transient cycle fuel economy with a publicly available benchmark.

The initial results from the test are encouraging - the BSFC value is within 1.2% of the value derived from running FTP cycle on a steady-state torque-to-fuel map. The engine-out emissions ($BSNO_x$ and BSSoot) levels generated during the test are compatible with US2010 EPA tail-pipe emissions requirements and can be controlled with contemporary after-treatment systems. Furthermore, compared to the MY2011 Cummins ISB 6.7L engine, Achates Power's OP engine presents a cycle-average BSFC advantage of 18% during hot-start FTP cycle. These results highlight the capabilities of the Achates Power OP engine to successfully run aggressive transient maneuvers without compromising the required performance attributes.

Introduction

Achates Power, Inc. (API) has been dedicated to modernizing the opposed-piston engine since its inception in 2004 and has solved various mechanical challenges faced by this engine architecture, including oil consumption, piston cooling, cylinder cooling, and wrist pin lubrication. Achates Power has also developed a unique set of performance and emissions strategies and combustion system recipes which enables the OP engine to meet current and future emissions while delivering excellent fuel consumption.

Initial work was conducted on a single cylinder engine to minimize the cost and complexity and to speed-up the turnaround time. In 2014, development and testing was directed to a 4.9L, multi-cylinder engine. Using the multicylinder engine, steady state performance and emission results were generated [2]. With the initial steady state calibration established, the next step was to prove out the transient capability of the engine [1]. It is critical to advance the understanding of the transient behavior of the OP engine by testing it on transient cycles in order to assess the ability of the engine to match the cycle while keeping the emissions under control. This paper highlights the results from testing OP engine on FTP (Federal Test Procedure), a heavy-duty transient cycle for on-road engines.

The paper starts with a detailed description of the test-bed -engine architecture, test-cell instrumentation and major control components and control software that enable efficient transient operation. A short summary highlighting the performance of research engine during steady-state is then provided. This is followed by the results from HD FTP transient cycle. Finally, these results are compared to the HD FTP test results from the US2010 EPA-compliant 2011 Cummins ISB 6.7L engine [3].

Test Bed Description

Engine Architecture

The multi-cylinder OP engine platform that is used to generate the results presented in this paper is heavily based on the single cylinder engine and shares most of its power cylinder components.

Table 1 shows the specifications and the performance attributes for the multi-cylinder OP engine.

Table 1. Multi-cylinder Achates Power OP engine specification

Displacement	4.9 L
Arrangement, number of cylinders.	Inline 3
Bore	98.4 mm
Total Stroke	215.9 mm
Stroke-to-Bore Ratio	2.2
Compression Ratio	15.4:1
Nominal Power (kW @ rpm)	205 @ 2200
Max. Torque (Nm @ rpm)	1100 Nm @ 1200-1600

Even though this engine was conceived as a research and test-platform, it powered all the accessories that were required to operate it. These accessories included lubrication oil pumps, a high-pressure fuel pump, a supercharger and a supercharger drive, and water-pumps. Being a test-platform, some of the components were intentionally designed to be oversized to provide flexibility required for exploring the capabilities of the engine. As a result, the size of the engine and the amount of friction that it has to overcome is greater than that of an optimized production engine. Figure 1 shows the engine in the test-cell. In order to provide realistic pumping operation, exhaust pressure was modified in real-time to simulate an aftertreatment system by using a back-pressure valve.

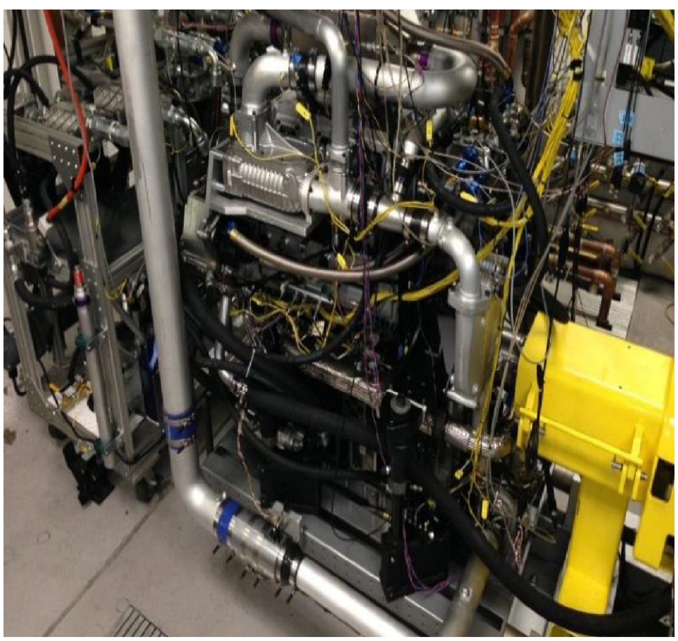

Figure 1. Engine first installed in test cell

Figure 2 provides an overview of the air-path for the three-cylinder API OP diesel engine. Upstream of the engine, a compressor driven by a fixed-geometry turbine is used to draw in fresh air. To aid the airflow across the engine, there is a supercharger driven by a two-speed drive that allows it to run at two different engine-speed ratios. For this engine, the two drive ratios, 3.2 and 4.6, are used. A supercharger bypass valve is used to control the airflow across the engine. The supercharger also acts as a pump to pull in the exhaust gases along the EGR loop. A venturi in the EGR loop, along with a delta-pressure sensor mounted across it, is used to measure the EGR mass-flow. An EGR valve is used to control the EGR flow to the engine. Downstream of the engine, a back-pressure valve is used to maintain the exhaust pressure levels (shown in Figure 3) due to a conventional aftertreatment system.

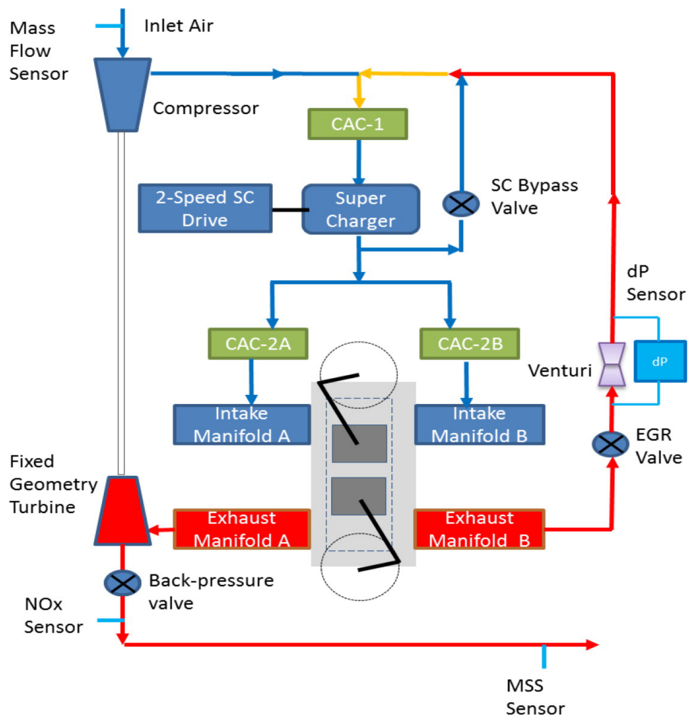

Figure 2. Air path schematic

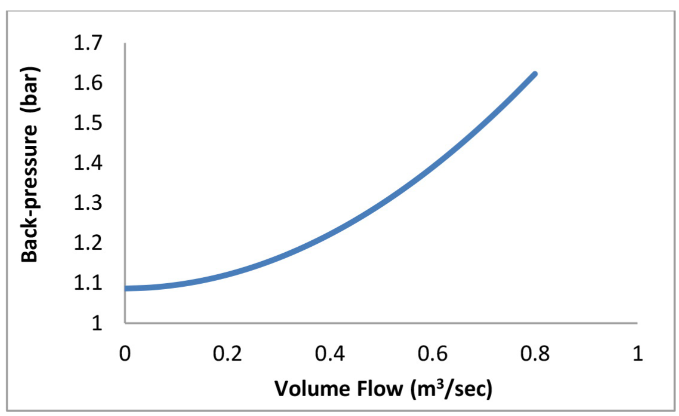

Figure 3. Target exhaust pressure

Dynamometer System

The dynamometer system consists of a SAJ SE-400 eddy current absorber unit with a capacity of 400kW and 2000Nm. The inertia offered by the dyno is 0.82kg-m^2. The engine is coupled to the absorber with a Cardan shaft of 10,000Nm capacity. The speed-control loop of the dynamometer is executed at a frequency of 200Hz. Since this is an absorbing dynamometer, it is not possible to execute the motoring portion of the FTP test-cycle. During such instances of the cycle, 10% of the brake-torque relative to the engine speed is commanded in order to avoid the conflict between ECU-based idle controller and the dynamometer speed controller. Unlike a transient dynamometer, this dynamometer does not support closed-loop torque control to meet torque set-points. The torque profile is forwarded to the ECU over CAN and the ECU software has a torque-to-fuel map that generates a fuel command to meet the desired torque.

Test Cell Instrumentation

Torque Measurement

Torque is measured by a Honeywell torque sensor (TMS9250) which is capable of measuring from 0 Nm to 4000Nm with an accuracy of +/- 4Nm. The torque flange is mounted in the driveline between the engine and the dyno-absorber. Zero-offset correction is performed before the test to account for any drift in the sensor measurement.

Engine-Out Soot Measurement

Particulate Matter (PM) consists of three main fractions - solid fraction (soot), Soluble Organic Fraction (SOF) and sulfate particulates. The mass of PM generated during engine test is normally determined by weighing the PM collected on the sampling filter. During the heavy-duty FTP test presented in this paper, such gravimetric measurement technique was not employed for two reasons:

a. Gravimetric measurements do not provide real-time characteristics of PM measurements during transients;
b. API's PM measurement system was limited by its ability to maintain a constant dilution ratio during transient maneuvers.

For this test, soluble organic fraction and sulfate particulate were not measured. The AVL483 Micro Soot Sensor (MSS), which measures solid fraction (soot), is employed for real-time measurement during the transient cycle. This sensor measures soot using the photoacoustic method, by which sample gas with soot particles is exposed to modulated light resulting in their contraction and expansion. This phenomenon results in the generation of sound waves in the carrier gas that are then detected by the microphone. The AVL483 MSS measures soot as a concentration of soot in exhaust (mg/m^3). In the test cell, the micro soot sensor is mounted in the exhaust line as shown in Figure 4. The AVL483 MSS is accompanied by an exhaust conditioning module that maintains a constant dilution ratio. For the purpose of this test, a dilution ratio of 20 was used.

Cycle average brake-specific soot is calculated using the following formula:

$$BS\ Soot = \frac{1}{1000\,.\ \rho_0 .\ \sum_{i=1}^{n} BPwr_i} \cdot \sum_{i=1}^{n} C_i \cdot q_{mew,i}$$

where,

BS Soot = Brake Specific Soot (g/kW-hr)

BPwr = Instantaneous Brake power (kW). Calculated by multiplying measured brake torque with engine speed.

ρ_0 = Density of exhaust gas under standard condition (0°C, 1013 mbar). It can be equated to the density of air, ρ_0(air) = 1.293 kg/m^3 within approximately 1%.

q_{mew} = Exhaust gas mass-flow (kg/hr)

C = Soot concentration (mg/m^3)

Soot measurement suffers from a transport delay due to the long travel path as well as the response time of the sensor. Whereas the sensor response time is a property inherent to the sensor, transport delay is dependent on the exhaust flow-rate. These properties result in temporal misalignment of the soot data relative to the event responsible for generating it. In order to compensate for this temporal misalignment, a variable time-shifting post-processing methodology was employed. It was assumed that all the major soot spikes originated at the time of sudden change in fuel-command. The variable time-delay algorithm aligned these major soot spikes (values greater than 50mg/m^3) to the respective spikes in fuel commands.

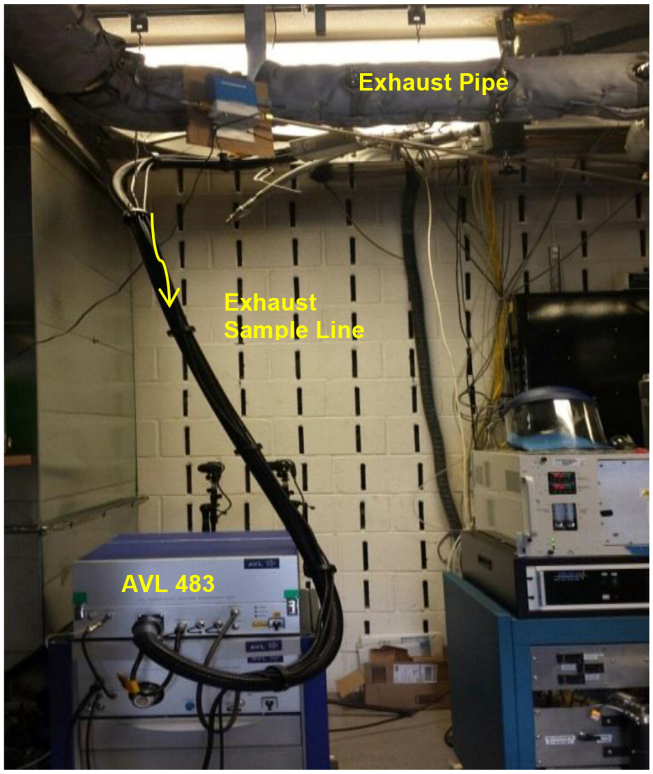

Figure 4. AVL MSS483 in test-cell

Engine-Out NO_x Measurement

In order to measure engine out NO_x in real-time, Continental's NO_x sensor was used in lieu of an FTIR-based gas-analyzer because of its capability to record NO_x spikes during transient events. Figure 5 shows the difference in the capability of these two measurement systems. During steady-state operations, both FTIR and Continental's NO_x sensor generate similar measurement results. But during transient operations, the FTIR based analyzer system installed in the API test-cell exhibit temporal delay as well as damping of the measurement amplitude. The difference in performance can be attributed to both response time of the sensors, as well as transport delay. Whereas Continental's NO_x sensor is mounted immediately downstream of the back-pressure valve, the FTIR sample lines are located further downstream along the exhaust pipe.

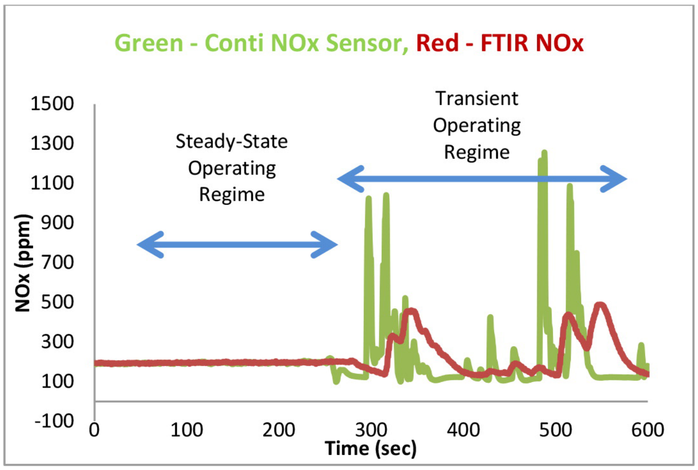

Figure 5. Comparison between FTIR NOx and Continental NOx sensor during steady-state and transient operations

Cycle average brake-specific NO_x is calculated using the following formula:

$$BSNO_x = \frac{1}{1000} \cdot \frac{MW\ NO_x}{MW\ Exhaust} \cdot \frac{1}{\sum_{i=1}^{n} BPwr_i} \cdot \sum_{i=1}^{n} NO_{x,i} \cdot q_{mew,i}$$

where,

$BSNO_x$ = Brake-specific NO_x (g/kW-hr)

BPwr = Instantaneous Brake power (kW). Calculated by multiplying Brake Torque with engine speed.

q_{mew} = Exhaust gas mass-flow (kg/hr)

MW NO_x = Molecular weight of NO_x = 46.006 kg/kmole

MW Exhaust = Molecular weight of exhaust. The cycle-average MW of exhaust is within 0.5% of MW of air, therefore, MW of air (28.97 kg/kmole) is used.

NO_x = Instantaneous concentration of NO_x in exhaust as measured by the sensor (ppm)

Since Continental's NO_x sensor is located closer to engine exhaust compared to AVL483 MSS, the time-delay for the sensor to register a NO_x variation is significantly lower. For a step change in injection timing at different speed-load points, the time constant is between 0.70 sec to 0.90sec. Therefore, the NO_x sensor data was advanced in time by an average value of 800 msec during post-processing.

Fuel Flow Measurement

Both soot and NO_x measurements are based on an instantaneous exhaust mass flow rate. Exhaust mass flow is calculated as following:

$$q_{mew} = Measured\ Airflow + Commanded\ fuel\ flow$$

The air mass flow-rate is measured using a MAF sensor located at the inlet to the compressor. The fuel mass flow-rate measurements from the make-up tank based fuel-flow meter in the test-cell suffer from slow-response time, hence, it is not well suited for measuring real-time fuel-flow during a transient cycle. On the other hand, the commanded fuel-flow rate is a value generated inside the controls software, hence, it is easy to obtain. It is assumed that the delay between commanded fuel quantity and delivered fuel quantity is negligible. A comparison was also made between the total fuel consumed over the whole cycle, as measured by the fuel flow meter in the test-cell, and the total fuel commanded over the whole cycle. The difference in these values was observed to be less than 0.8%. Therefore, the commanded fuel flow rate is used to calculate exhaust mass-flow rate.

Controls System Actuators

Control system components related to the air-handling system are briefly introduced. Response time of these components is of significant importance and can help in improving transient response.

Two-Speed SC Drive

In order to efficiently operate the supercharger at different engine speeds and to provide faster airflow response during transient engine operation, API designed and developed a proprietary two-speed supercharger (SC) drive. The design consists of a planetary gear-set and hydraulically actuated clutch. By varying the input through a ring gear, fixed input (sun gear), and the output (planet carrier), the two-speed drive can be in either the high-speed mode or low-speed mode. Clutch actuation of the two-speed drive is controlled by the ECU. The steady-state operating regime of two-speed drive is shown in Figure 6.

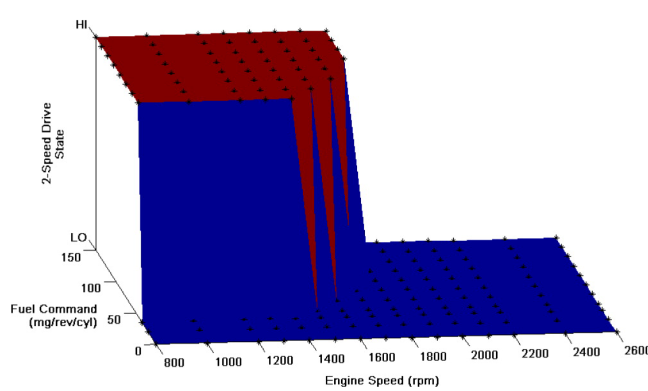

Figure 6. Operating regime for two-speed SC Drive during Steady State

Since the two-speed drive is hydraulically actuated, its response time is dependent on the oil-pressure and oil-temperature. At 3.5 bar oil-pressure, the step-response time of the two-speed drive, when switching from low-ratio to high-ratio, is 0.6 sec, whereas the response time in the opposite direction is 0.3 sec. On the engine, the two-speed SC drive is actuated by the oil from lubrication circuit. Since the two-speed SC drive mounted on the engine is not equipped with the speed-sensors at the input and output shafts, the change in the pressure ratio across the super-charger is used to confirm if the drive-ratio switching has occurred.

SC Bypass Valve

The supercharger (SC) bypass, or recirculation valve is a 48mm diameter butterfly valve, driven by an H-bridge in the Engine Control Unit (ECU). The SC bypass valve will fully seal when pushed closed. This valve is the base controller of airflow through the engine. API has developed control software to minimize the response time of this critical component while maintaining durability of the plate and bore. The valve can be driven fully closed to fully open in 0.16 seconds and fully open to fully closed in 0.19 seconds. The valve can be moved faster in one direction due to an internal fail-safe spring which forces it open in the case of a system fault.

EGR Valve

The EGR valve is a 63mm diameter butterfly valve smart actuator, with the set point sent over a Controller Area Network (CAN) from the ECU. All motor drive software is contained within the valve with the ECU sending a command message every 20msec. The valve can be driven fully closed to fully open in 0.18 seconds and fully open to fully closed in 0.16 seconds. The valve can be moved faster in one direction due to an internal fail-safe spring, which forces it closed in the case of a system fault.

Controls Software

Achates Power has developed proprietary controls software that addresses unique challenges faced by the OP engine configuration. For air-handling control, API has developed strategies to control airflow using the supercharger recirculation valve and a two-speed drive, whereas EGR is controlled by the EGR valve. For the airflow controller, air mass-flow feedback is provided by a MAF sensor mounted before the compressor. For the EGR control, EGR mass flow feedback is provided by a delta pressure sensor, which is mounted across the venture along the EGR loop (see Figure 2). API has also developed and implemented controls strategies for controlling rail-pressure for the common rail system, which allows it to utilize two injectors per cylinder to inject the fuel.

For the FTP transient cycle testing, a smoke-limiter, transient modifier and feed-forward controllers for air-handling and EGR were incorporated into the controls software.

The smoke-limiter is active during driver acceleration demand - when the airflow requirements in the engine are not immediately met due to transport delay and turbo lag. Since the fuel-system reacts faster to driver torque demand than the air-handling system, AFR decreases. If global or trapped AFR goes below a threshold, it can result in incomplete combustion thereby generating smoke. The smoke-limiter regulates fuel-flow rate into the engine in order to prevent the global or trapped AFR value from going below a certain threshold. A trade-off associated with calibrating the smoke-limiter aggressively is that although it reduces soot emissions it also causes slower torque response to the driver's torque request.

During a transient event, it is essential to quickly meet the airflow set-point and this is attained by two transient modifiers - the SC Drive Ratio modifier and the EGR valve modifier. Figure 6 shows the operating regime of two-speed drive during steady-state operation. During transient operation, depending on the torque demand, the two-speed drive is switched to a higher drive ratio (if it is not at high-drive ratio already). Shifting to a higher drive ratio during a transient event allows for faster build-up of the airflow. Apart from switching to a higher drive-ratio during driver acceleration demand, the EGR valve is commanded to almost close position, which causes the turbine to be exposed to higher exhaust flow. This results in increased turbine work, thereby driving the compressor to generate more boost and rapidly building the airflow.

In order to reduce soot, apart from smoke-limiter, additional fuel-related transient modifiers - rail-pressure and injection timing - can also be used. Increasing the rail-pressure during a transient event results in better combustion. Advancing injection timing also results in reducing soot by allowing more time for soot oxidation. The rail-pressure modifier and injection timing modifier are constrained by soot-NO_x tradeoff and require a significant calibration effort. Therefore, in this round of testing, a rail-pressure modifier was deployed but injection timing modifier was not used.

All the controls strategies reside in a M470 Open ECU© rapid-prototyping platform provided by Pi Innovo. For data-acquisition and calibration ATI Vision software is used. ATI Vision acquires data from the ECU over a CAN (Controller Area Network) at a frequency of 50 Hz. Inside the ECU, sensor data is mostly acquired at 100Hz.

Steady State Results Summary

Despite higher friction in the research engine, the measured steady-state fuel consumption and emissions showcase the potential benefits of the OP engine. The results from a full engine mapping exercise are shown in Figure 7, Figure 8 and Figure 9. The 12-mode steady state test, which was replicative of heavy-duty SET (Supplemental Emission Test) without idle mode, was performed on the engine. The 12-mode cycle average fuel consumption of 199 g/kWh is only 8 g/kWh higher than the best point, which illustrates the flat nature of the engine fuel consumption characteristics. Figure 8 and Figure 9 highlight the levels of engine-out brake-specific emissions across the whole operating regime. Engine-out measurements of both $BSNO_x$ (cycle average 3.44 g/kW-hr) and BSSoot (cycle average 0.021 g/kW-hr) are both well within the range required to meet tailpipe emissions standards with conventional aftertreatment equipment.

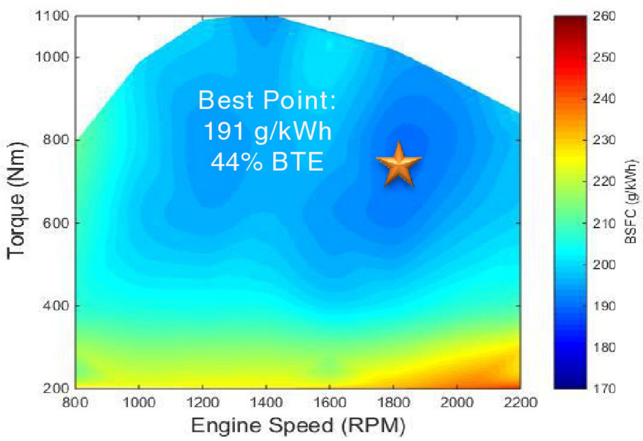

Figure 7. Multi-cylinder OP research-engine measured BSFC map

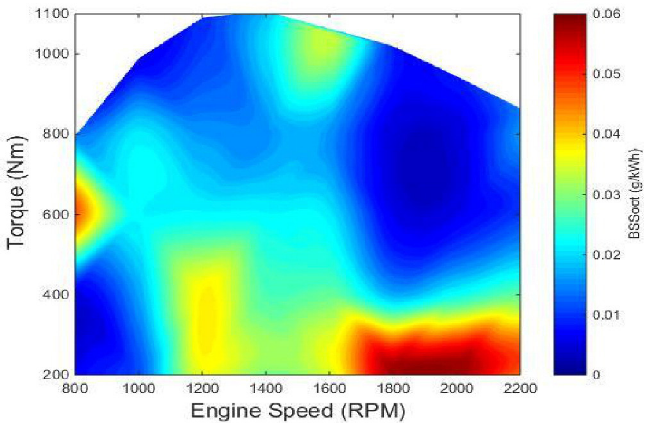

Figure 8. Multi-cylinder OP research engine measured (engine-out) BSSoot map

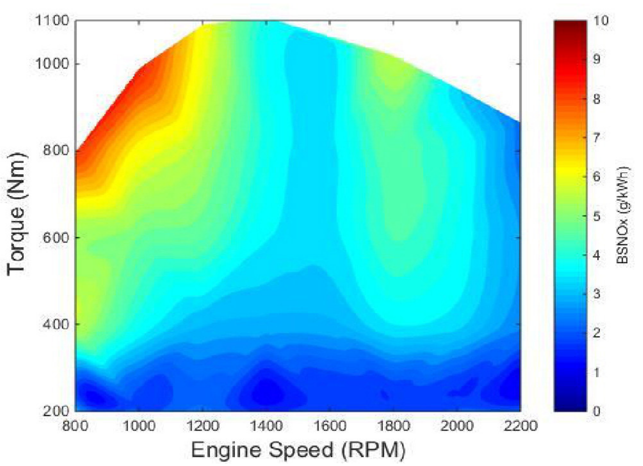

Figure 9. Multi-cylinder OP research engine measured (engine-out) $BSNO_x$ map

This exercise provided the opportunity to validate and further correlate the OP-specific 1-D and CFD models that API has developed through the years to predict the performance of multi-cylinder engines. The validated models help in predicting the performance attributes of a production-level version of this engine. A roadmap for an optimized engine was discussed in [2] with the following improvements aimed at multiple aspects of engine design:

- Combustion improvements using next generation piston bowls, injector nozzle hole size, spray angles and clocking.
- Reduction in pumping work by optimizing port design to improve the scavenging process and using efficient supercharger and turbocharger designed for opposed-piston engines.
- Reduction in friction by eliminating superfluous and redundant bearings and seals (used in the research engine for development flexibility and convenience) and re-sizing and/or optimizing the cranktrain components, the power-cylinder, the oil/coolant flows and the gear-train.

With these improvements, the expected production level BSFC map is shown in Figure 10.

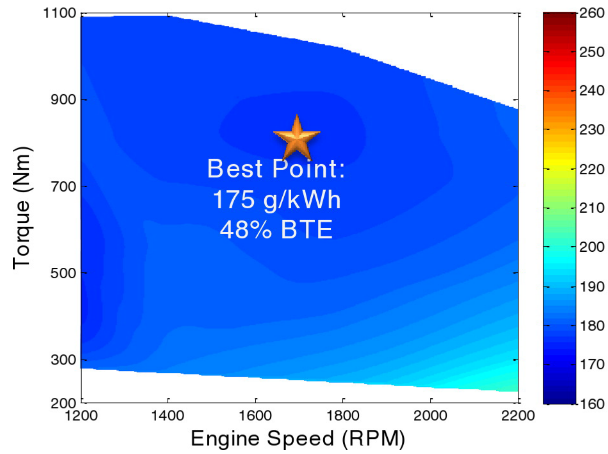

Figure 10. 4.9L production level BSFC map prediction

The production level engine would be capable of achieving a 12-mode cycle average fuel consumption of 180 g/kWh.

HD FTP Cycle Testing

The heavy-duty FTP transient cycle is used for regulatory emission testing of heavy-duty on-road engines in the US. The cycle includes the "motoring" segment, and, therefore requires a DC or AC electric dynamometer capable of both absorbing and supplying power [4]. As mentioned earlier, since the API test cell is equipped with an eddy-current absorbing unit, motoring is not possible. During the motoring portion of the cycle, 10% of maximum brake-torque relative to the engine speed is commanded. Such an arrangement allows for generation of power during the motoring segment but it also results in a fuel consumption penalty during those segments.

Furthermore, the FTP cycle test consists of a cold-start test followed by minimum of three hot-start tests separated by 20 minute intervals. Overall FTP results are obtained by using a weighting factor of 1/7 and 6/7 for the cold and hot-start results, respectively [4]. The test results presented and discussed in this paper are confined to the hot-start portion of the FTP cycle.

Test Cycle Requirements

As mentioned earlier, the intent of this test is not to receive certification, but to present the capability of engine to efficiently handle transient operations while producing engine-out emissions compatible with US2010 EPA emissions regulations. Nonetheless, it is of the interest to observe how closely the cycle target speed and torque can be followed. For the FTP cycle, the statistical criteria for validating the duty cycles is shown in the Table 2[4].

Table 2. Statistical criteria for validating FTP cycle [4]

Parameter	Speed	Torque	Power
Slope of regression line, a_1	$0.95 \leq a_1 \leq 1.03$	$0.83 \leq a_1 \leq 1.03$	$0.83 \leq a_1 \leq 1.03$
Absolute value of intercept, $\|a_0\|$	≤ 10% of warm idle	≤ 2% of maximum mapped torque	≤ 2% of maximum mapped power
Standard Error of Estimate, SEE	≤ 5% of maximum test-speed	≤ 10% of maximum mapped torque	≤ 10% of maximum mapped power
Coefficient of determination, R^2	≥ 0.970	≥ 0.850	≥ 0.910

Test Cycle Assumptions and Modifications

For engine mapping the following values are used

Minimum speed (Idle speed) = 800 rpm

Maximum engine speed = 2200 rpm.

Figure 11 shows maximum torque curve with respect to engine speed and the respective points on FTP cycle.

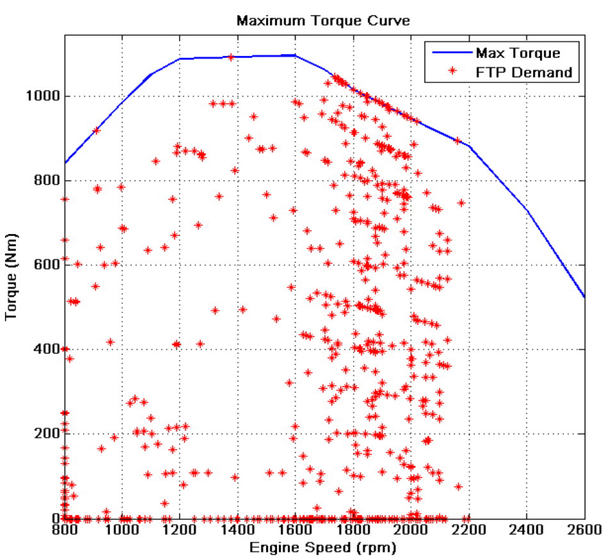

Figure 11. Engine torque-speed curve and FTP cycle requirements

Test Cycle Performance and Results

Control Features and Performance

A slew of control features were successfully deployed for the purpose of smooth transient operation. Figure 12 shows the statistics for the performance of the air-handling controller and the rail-pressure controller during the FTP cycle. Air-handling controller statistics show that even though most of the air-handling set-points are being met, there are a few instances where actual airflow is lower than the desired airflow. This can be attributed to a higher volume of air-system in this research engine. With a more optimized air-system design, for an engine targeted for production, these set-points can be easily met. With two injections per revolution (from two separate injectors) in a two-stroke engine, rail-pressure control presents its own challenges.

The bottom graph in Figure 12 shows that even though the rail-pressure set-points are being met for most of the cycle, the controller tends to be over-aggressive at some points. With an optimized rail-pressure controller, these outliers can be eliminated.

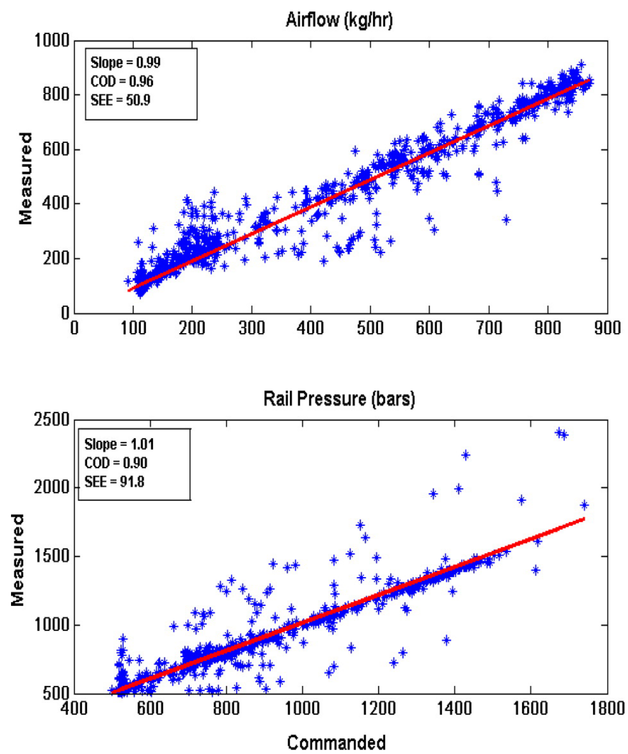

Figure 12. Performance of air-handling and rail-pressure controller

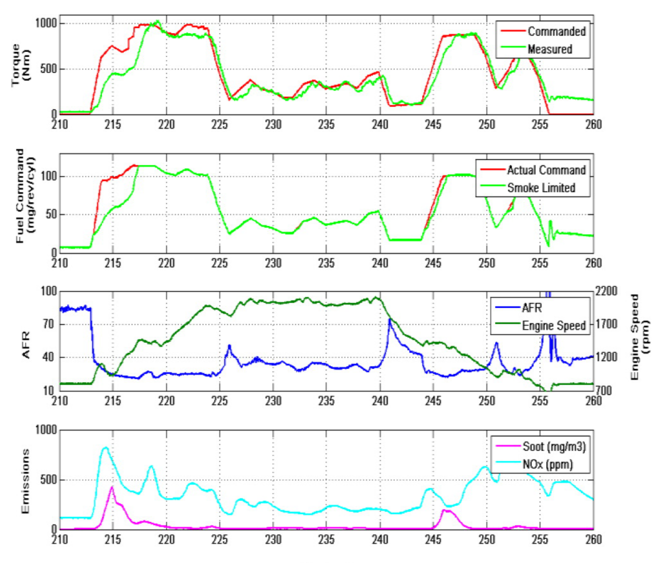

Figure 13. Smoke-limiting operation during NYNF section

Figure 13 shows the performance of the smoke-limiter across a snippet of the first New York Non-Freeway (NYNF) cycle. In response to the load demand, instantaneous fueling results in a drop in AFR, which triggers the smoke-limiter to intervene and regulate fuel to the engine. This helps in restricting the soot spikes within the desired threshold for this test.

Figure 14 shows the performance of air-handling controllers for airflow and EGR along with individual actuators during the same snippet of NYNF cycle.

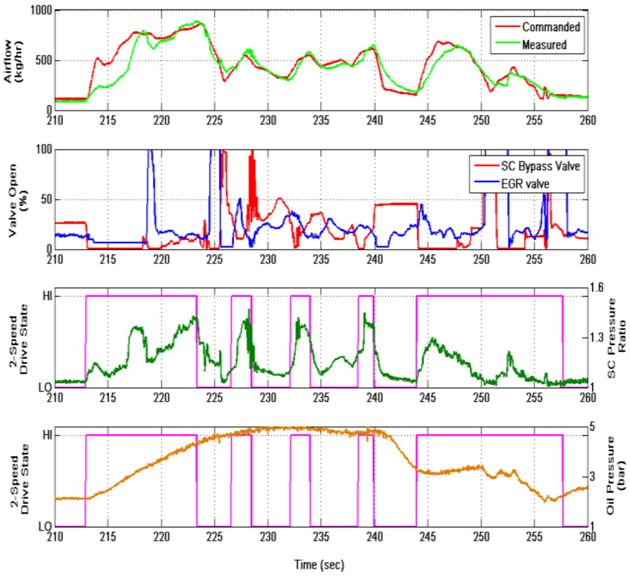

Figure 14. Air-handling control during NYNF section

The load increase as shown in Figure 14, at 213 seconds, starts from idle. During this event there is a delay in meeting the air-flow demand even when the SC bypass valve and the EGR valve have been closed. Even though the two-speed SC drive is commanded to switch to a higher drive-ratio, it does not actually switch till approximately 4 seconds after the command. This is evident from the pressure ratio across the super-charger (dark green) which does not changes until the oil-pressure (orange) to the super-charger drive increases to 3 bar. The low engine speed, slow-response oil-valves and an oversized oil-circuit in this research engine results in the delay in building the oil-pressure that is required to switch the ratio of the two-speed SC drive. With an optimized oil-circuit it will be possible to actuate the SC drive across the whole operating range of the engine without compromising the engine efficiency. This will increase the response time of the two-speed SC drive and significantly eliminate the delays in meeting airflow demand especially when starting from an idle condition.

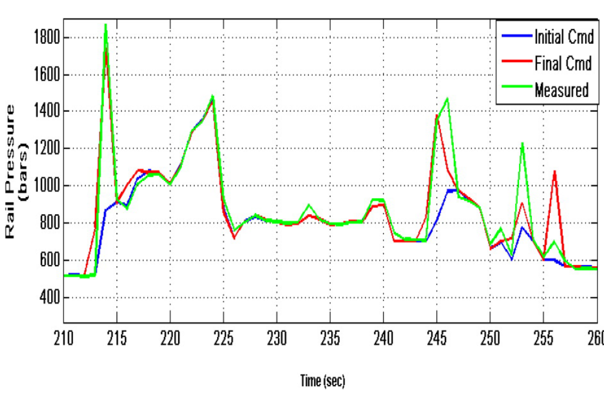

Figure 15. Transient Modifier for Rail Pressure

During this FTP test, three transient modifiers were utilized for the SC Drive ratio, for the EGR Valve position and for the rail-pressure. Transient modifiers modify the set-points for the desired variables during transient events on the basis of the intensity of transients. The first two modifiers are shown in Figure 14, and they directly impact the air-handling system response. The rail-pressure modifier is shown in Figure 15.

Test Cycle Results

Figure 16 highlights the statistical results for the FTP cycle, which confirms that the API engine is capable of closely matching driver torque demand. Coefficient of Determination (COD) or R^2 value for brake torque, engine speed and engine brake power meets the cycle statistics requirements stated in Table 2.

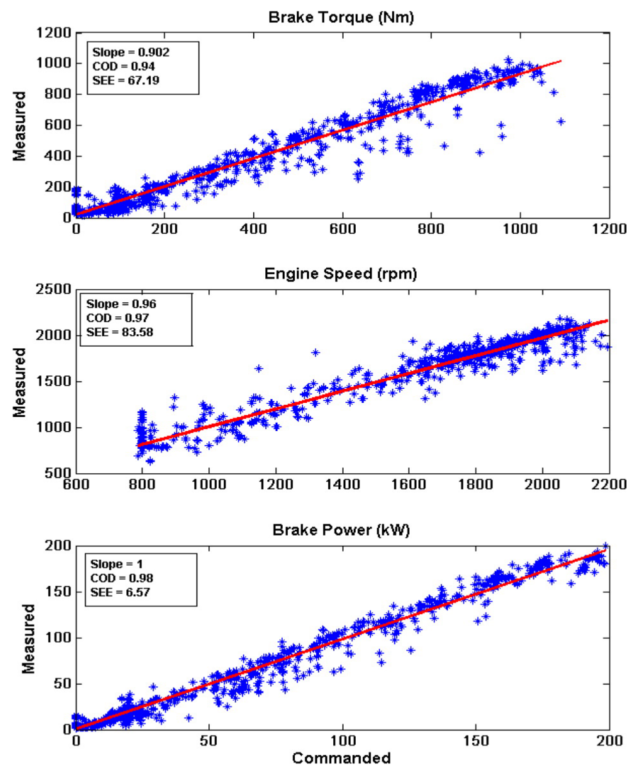

Figure 16. Statistical results for FTP cycle

The engine-out brake-specific cycle average values over the FTP cycle are shown in the Table 3. NO_x measurement shown in the table has been corrected for humidity.

Table 3. Cycle average results from hot-start FTP cycle

BSFC (g/kW-hr)	Engine-out BS Soot (g/kW-hr)	Engine-out BS NO_x (g/kW-hr)
217.3	0.056	4.3

Achates Power has collaborated with a leading catalyst supplier to perform simulation and analysis of after-treatment system for OP diesel engines. The results from this study --which will be presented at the GAMC Emissions 2016 conference in Troy, Michigan -- show that the properties of exhaust gases generated during the FTP test in conjunction with an SCR system provide the potential to meet the US2010 EPA tailpipe NO_x levels requirements.

With 95-99.9% filtration efficiency for elemental carbon [5] , a conventional DPF will filter the resulting engine-out soot from the FTP cycle to meet US2010 EPA PM requirement with enough margin (a factor of five) to offset for SOF and sulfate fraction of PM which were not measured during this test.

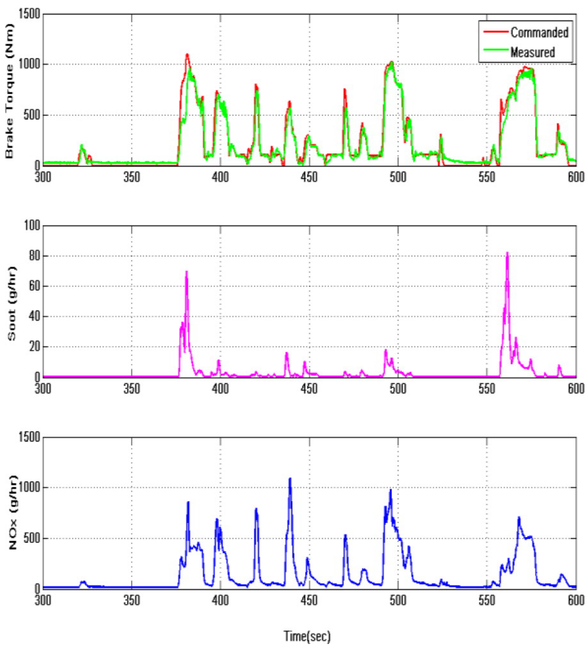

Figure 17. LA Non-freeway section performance

Figure 17, and Figure 18 show the performance of the engine during the LA Non-Freeway and LA Freeway portion of the cycle.

During the non-freeway sections of the FTP, there are multiple instances of instantaneous torque demand which, as shown in Figure 17, are easily met. Torque demand is accompanied by a spike in NO_x, which results from closing the EGR valve to 10% opening angle in order to build more boost. There are fewer soot spikes during the non-freeway section because the two-speed SC drive is able to provide the air on demand during the transient event.

Figure 18 shows that the torque demand during the freeway portion of the FTP cycle is easily met without generating major soot spikes. NO_x values during the freeway section are relatively higher because the engine is running near the rated speed points.

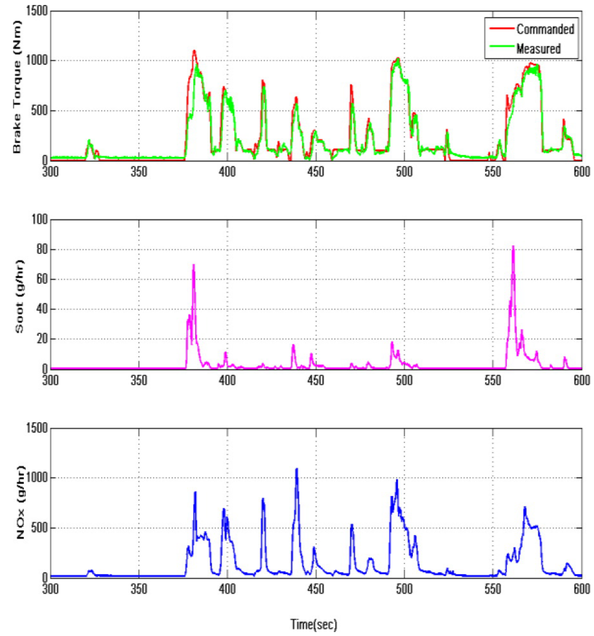

Figure 18. LA Freeway section performance

Steady-State Map-based FTP Cycle Results

One important objective of conducting this exercise was to identify the deviation in the measured results for FTP cycle from the ideal performance during the cycle as derived from the steady-state map. When simulating the measured cycle torque and cycle speed from the test on a steady-state map - for BSFC, BSNOx and BS Soot - the results between simulations and actual measurements from the FTP test are close to each other. The simulated value represents the brake-specific values obtained if the cycle speed and torque is converted to fuel without factoring penalties due to engine transient operations such as smoke-limiter or airflow lag.

Table 4 shows that the results for the BSFC from the simulation and actual measurement are within 1.2%, thereby demonstrating the capability of the API OP engine to match the steady state performance even during transient operations because of the novel combustion system, flexible air system and respective control strategies.

Table 4. Comparison between FTP results on Steady-state map and actual measurements

	Units	**From Steady-state map**	**From Measurement**
Total Fuel Consumed	(g)	**3163**	**3201**
Cycle-average BSFC	(g/kW-hr)	**215.2**	**217.3**
Engine-out cycle-average Soot	(g/kW-hr)	**0.01**	**0.056**
Engine-out cycle-average NO_x	(g/kW-hr)	**4.1**	**4.3**

Comparison with Cummins ISB 6.7L Engine

Engine efficiency measurements and an energy audit were performed on a medium duty MY2011 Cummins ISB 6.7L engine in [3]. As shown in Table 5, the performance ratings for the Cummins engine is close to API's test engine. Cummins engine was equipped with a DPF and SCR system, whereas API's OP test-engine maintains exhaust-pressure to simulate a conventional after-treatment system (DPF/DOC and SCR).

Table 5. Performance specs comparison between API's and Cummins ISB MD engines

	API OP Engine	MY2011 Cummins MD Engine [3]
Displacement (L)	4.9	6.7
Rated Power (kW)	205	242.5
Rated Speed (rpm)	2200	2400
Peak Torque (Nm) @ Speed	1100 Nm @1200-1600 rpm	1016@1600rpm
Compression Ratio	15.4 : 1	17.3 : 1
EGR	HP cooled	HP cooled
After-treatment System	Exhaust pressure simulating DPF/DOC/SCR for MD engine	DPF-SCR

The results from running a hot-start FTP cycle on Cummins engine are shown in [3]. Table 6 shows that with a similar level of work done during the cycle, the brake-specific fuel consumption for API's OP engine is lower than the brake-specific fuel consumption for a Cummins ISB 6.7L engine by 18%.

Table 6. Hot-start FTP Cycle performance comparison

	API OP Engine	MY 2011 Cummins MD Engine [3]
Total energy generated over the cycle (kW-hr)	**14.73**	**15.30**
Total fuel-consumed (kg)	**3.201**	**4.00**
BSFC (g/kW-hr)	**217.3**	**261.4**

Summary and Conclusions

Opposed Piston diesel engines possess inherent efficiency advantages when compared to a traditional internal combustion engine. The results in this paper show that this advantage also extends to the aggressive transient cycles. The results of a successful hot-start HD FTP cycle on the Achates Power Opposed Piston diesel engine were presented in this paper. These results demonstrate the capability of the OP engine to not only provide significant BSFC advantage over a conventional four-stroke diesel engine, but also highlight its ability to generate engine-out emission levels that are compatible with US2010 EPA requirements with a conventional after-treatment system.

In conclusion, the objectives of this study were satisfied and the transient testing of an OP engine during a hot-start cycle was characterized. This study hopes to contribute the following:

1. The measured brake-specific fuel consumption (BSFC) of OP engine during hot-start FTP cycle is within 1.2% of the value derived from a steady-state torque-to-fuel map.
2. The brake-specific engine-out exhaust emissions generated during hot-start FTP cycle are compatible with US2010 EPA requirements using conventional after-treatment system.
3. During a hot-start FTP cycle, Achates Power's OP research engine presents a cycle-average BSFC advantage of 18% when compared to the MY2011 Cummins ISB 6.7L engine.

Even though this research OP engine has a higher charge system volume and a higher friction than an optimized engine, it still provides a significant performance improvement over a four-stroke production engine. An optimized production version of the OP engine will most likely provide an even superior performance and will preserve its advantage when compared to the proposed advances in future four-stroke engines.

References

1. Redon, F., Sharma, A., and Headley, J., "Multi-Cylinder Opposed Piston Transient and Exhaust Temperature Management Test Results," SAE Technical Paper 2015-01-1251, 2015, doi:10.4271/2015-01-1251.
2. Naik, S., Redon, F., Regner, G., and Koszewnik, J., "Opposed-Piston 2-Stroke Multi-Cylinder Engine Dynamometer Demonstration," SAE Technical Paper 2015-26-0038, 2015, doi:10.4271/2015-26-0038.
3. Thiruvengadam,A., Pradhan,S., Thiruvengadam,P., Besch,M., Carder,D., Delgado, O., "Heavy-Duty Vehicle Diesel Engine Efficiency and Energy Audit", http://www.theicct.org/heavy-duty-vehicle-diesel-engine-efficiency-evaluation-and-energy-audit ,10/12/2014.
4. EPA, 40 CFR 1065, "Engine Testing Procedures", https://www.gpo.gov/fdsys/granule/CFR-2011-title40-vol33/CFR-2011-title40-vol33-part1065.
5. Majewski W.Addy, "Diesel Particulate Filters", Dieselnet, https://www.dieselnet.com/tech/dpf.php

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Abbreviations

API - Achates Power Inc

OP2S - Opposed Piston 2-Stroke Engine

SC - Supercharger

DOC - Diesel Oxidation Catalyst

DPF - Diesel Particulate Filter

SCR - Selective Catalytic Reduction

BSFC - Brake Specific Fuel Consumption

FTP - Federal Test Procedure

SET - Supplemental Emission Test

EPA - Environmental Protection Agency

PM - Particulate Matter

COD - Coefficient of Determination

EGR - Exhaust Gas Recirculation

AFR - Air-Fuel Ratio

HD - Heavy Duty

MD - Medium Duty

ECU - Engine Control Unit

CAN - Controller Area Network

FTIR - Fourier Transform Infrared Spectrometer

MSS - Micro Soot Sensor

SOF - Soluble Organic Fraction

BS - Brake specific

MAF - Mass Air Flow

CFD - Computational Fluid Dynamics

GAMC - Global Automotive Management Council

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