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# Multi-Cylinder Opposed Piston Transient and Exhaust Temperature Management Test Results

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## Abstract

In a recent paper, Opposed-Piston 2-Stroke Multi-Cylinder Engine Dynamometer Demonstration [1] published at the SAE SIAT in India in January 2015, Achates Power presented work related to the first ever opposed piston multi-cylinder engine fuel economy demonstration while meeting US 2010 emissions.

The results showed that the research 4.9L three cylinder engine was able to achieve 43% brake thermal efficiency at the best point and almost 42% on average over the 12 modes of the SET cycle. The results from this test confirmed the modelling predictions and carved a very robust path to a 48% best BTE and 46.6% average over the cycle for a production design of this engine.

With the steady state performance and emissions results achieved, it was time to explore other attributes.

In this second paper, the OP engine transient capabilities were assessed by testing the engine performance and emissions response under a typical transient maneuver: 25% to 100% load step at low and constant engine speed. It can be seen that during such transition, it is possible to control both NOx and Soot with a minimal torque lag.

Additionally, this paper will present the ability of the OP engine to manage exhaust temperatures. It is shown that the air system control flexibility and robust combustion system that Achates Power developed for the OP engine can be used to achieve high exhaust gas temperatures for a diesel engine at idle-like speeds and loads. In this initial exercise, it is shown that close to 500°C engine out temperature can be achieved at 1100RPM with no torque output.

## Introduction

Achates Power (API) has been dedicated to modernizing the opposed piston engine since its inception in 2004. Since then API has diligently solved every single mechanical challenge for the

technology including oil consumption, piston cooling, cylinder cooling, and wrist pin lubrication to name the main ones.

Achates Power also developed a unique set of performance and emissions strategies and combustion system recipes that unleashed the potential for the OP engine to meet current and future emissions while delivering excellent fuel consumption.

Most of this early work was better served running a single cylinder engine to minimize the cost, complexity and to speed-up the turnaround time.

The next step was to integrate all these solutions in a self-sustaining multi-cylinder engine. The first running of 4.9L engine occurred during the first quarter of 2014.

Subsequently, steady state performance and emission results was generated and published in the SAE paper Opposed-Piston 2-Stroke Multi-Cylinder Engine Dynamometer Demonstration [1]

With the initial steady state calibration established, the next step was to prove out the transient capability of the engine.

Another aspect critical to having a viable product and meeting tailpipe emissions is the ability to manage exhaust after-treatment temperatures for catalyst light-off or DPF regeneration.

## Multi-Cylinder OP Research Engine Steady State Results Summary

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

This multi-cylinder research engine was designed to meet the performance levels shown in the Table 1.

Table 1. Multi-cylinder OP2S engine specification.

| Displacement | 4.9 L |
|---|---|
| Arrangement, number of | Inline 3 |
| Bore | 98.4 mm |
| Total Stroke | 215.9 mm |
| Stroke-to-Bore Ratio | 2.2 |
| Compression Ratio | 15:01 |
| Nominal Power (kW @ rpm) | 205 @ 2200 |
| Max. Torque (Nm @ rpm) | 1100 Nm @ 1200-1600 |

This engine was conceived as a research test platform with flexible components and systems beyond what would be done in a production intent version. With such a large number of trade-offs made, the engine's size and friction were affected negatively. Figure 1 depicts the engine on the test bed during the testing.
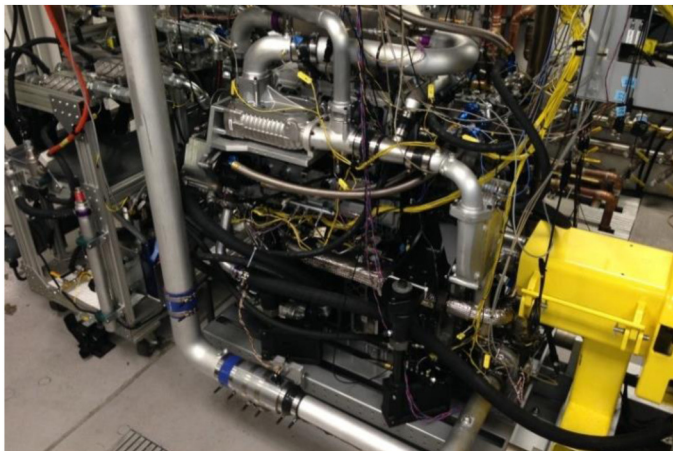


Fig 1. Engine first installed in test cell.

As presented in the paper [1] the fuel consumption measurements while meeting engine out emissions levels compatible with US EPA 2010 levels are nevertheless compelling as can be shown in the Figure 2.
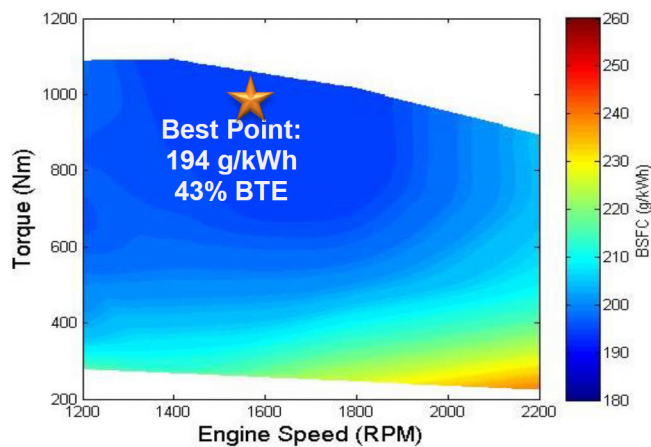


Fig 2. Multi-cylinder OP research measured BSFC map.

The measured BSFC map also confirmed the flat response of the engine. The 12 mode cycle average fuel consumption of 201 g/kWh is only 7 g/kWh higher than the best point illustrating the flat nature of the engine fuel consumption characteristics.

This exercise provided the opportunity to validate and further correlate the OP specific models that Achates Power developed through the years to predict the performance of multi-cylinder engines.

With the models validated it was now possible to confidently and accurately predict what a production level version of this engine would be able to deliver. The details of this roadmap are provided in the paper [1]. The expected production level BSFC map is shown in figure 3.
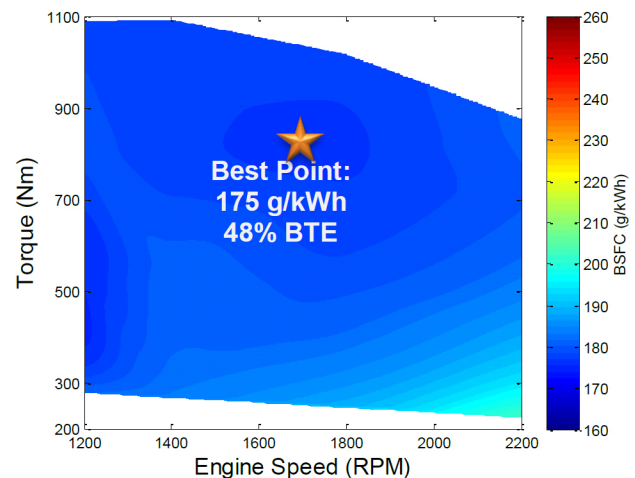


Fig 3. 4.9L production level BSFC map prediction

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

## Transient Testing

Transient testing was performed on the 4.9L research engine with the objective of characterizing the following performance metrics of the system, in general, and the controls software, in particular:

1.  Response time of air-handling system which includes airflow control

2.  Emission levels (soot and NOx)

For tip-in operations when there is a step change in driver torque request, in order to have a faster engine response, it is essential to build up the required amount of air in the system quickly while maintaining the desired emission levels. The initial transient test-profile used for this study was not a step-change in torque request but a driver torque request ramp from 25% load to 100% load in one second at a constant speed. With existing OP2S engine configuration shown in Fig.6, there are two design constraints that limit the faster build-up of air in the system - super-charger drive ratio and opening angle of super-charger recirculation valve (Also called bypass valve). In order to characterize the response of the air-handling system control, two super-charger drive ratios were selected - 3.8 and 5.0. With a higher drive ratio, it is possible to improve the response of air-handling system by quickly meeting the airflow requirements. For engine out NOx measurement, a smart NOx sensor was mounted at turbine outlet. Micro-soot sensor was used to measure engine-out soot-density.

A transient test-profile was generated and executed on the engine test bench to transition the torque request between A25 and A100 in one second. Two different fueling strategies were used - one with the same ramp-up time as the driver torque request (Fig.4), and the second one with a slower fuel ramp-up time of 2 seconds to replicate a rudimentary smoke-limiter (Fig. 5). Whereas a more evolved smoke-limiter limits the fuel quantity in order to prevent AFR to go below a threshold, the rudimentary smoke-limiter used in this test was generated through quick calibration changes in order to characterize the impact of changing the fuel ramp-up time on soot generation during the transients.

Apart from EGR rate (commanded and measured), all other values are presented relative to their maximum values collected in the data. Airflow (commanded and measured), torque (commanded and measured), fuel (commanded), supercharger recirculation valve opening position (measured) and EGR valve opening position (measured) are presented relative to their maximum values. In order to compare the emission levels with two super-charger drive ratios, emission spike amplitudes - for NOx and soot - are presented relative to the maximum of emission values between the two super-charger drive ratios. As will be presented later in this paper, it was observed that the transient maneuvers with the lower super-charger drive-ratio - 3.8 - generated higher emission values. Therefore, the emission spike amplitudes for both super-charger drive ratios are presented relative to the maximum emission values obtained with drive-ratio 3.8.
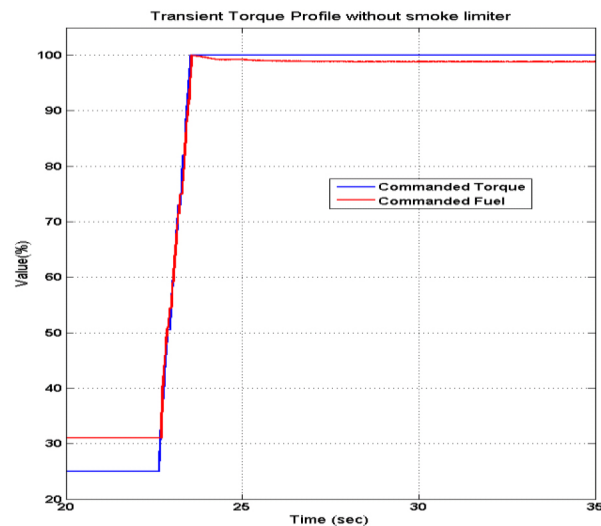


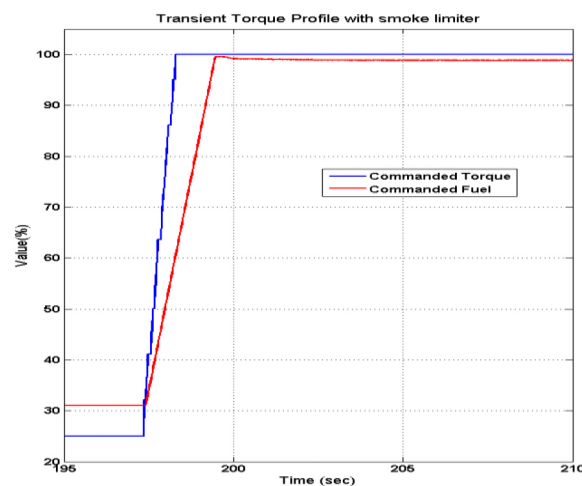Fig 4. Torque command profile without smoke-limiter



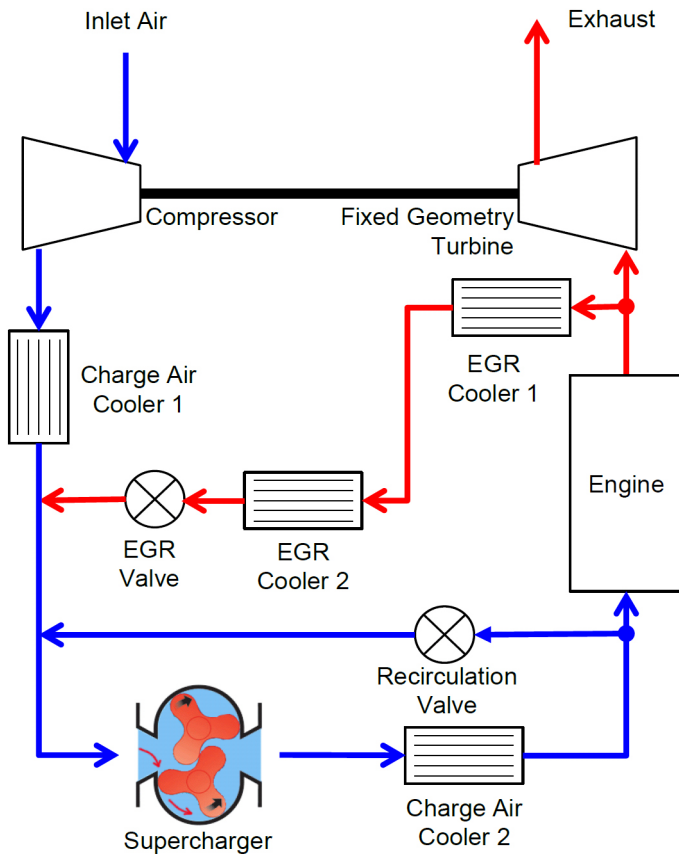Fig 5. Torque command profile with rudimentary smoke-limiter

Figure 6. Opposed-piston, two-stroke air system layout.

## Case1: SC DR = 3.8

Fig.7, Fig.8 and Fig.9 show the response of the engine to the commanded torque profile without the smoke-limiter. Fig.10, Fig.11 and Fig 12 shows the response to the torque request with smoke-limiter enabled.
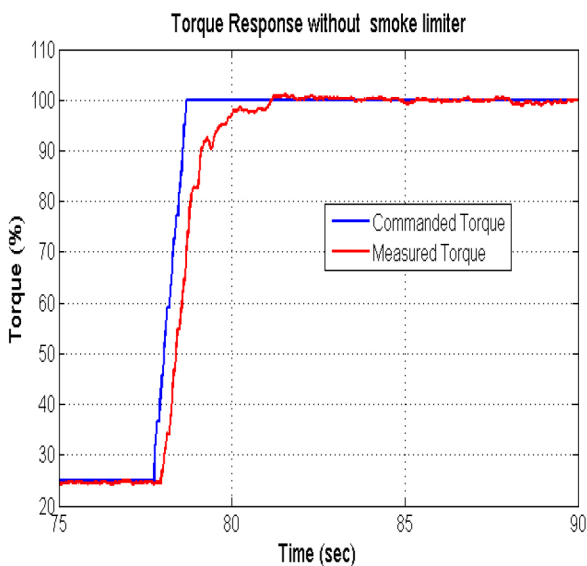


Fig 7. Torque response without smoke-limiter with SC DR = 3.8

Comparing the two responses, it is observed that without a smoke-limiter, torque response is faster but it also results in significant soot-spike due to the lag in airflow build-up as shown in Fig.8. In order to build up the air faster, the controller commands the super-charger recirculation valve to close immediately to maximize the flow through the engine. As shown in Fig.8, even with supercharger recirculation valve fully closed slower build-up of causes significant soot generation. In smoke-limiter case, as shown in Fig.10, both the fuel and airflow command are ramped up slowly. Such airflow command is easily met by closing the supercharger recirculation valve, thereby resulting in a lower soot spike. Fig.9 and Fig.12 show the control of EGR rate through EGR valve during the transient in both -smoke-limitng and non-smoke-limiting mode.
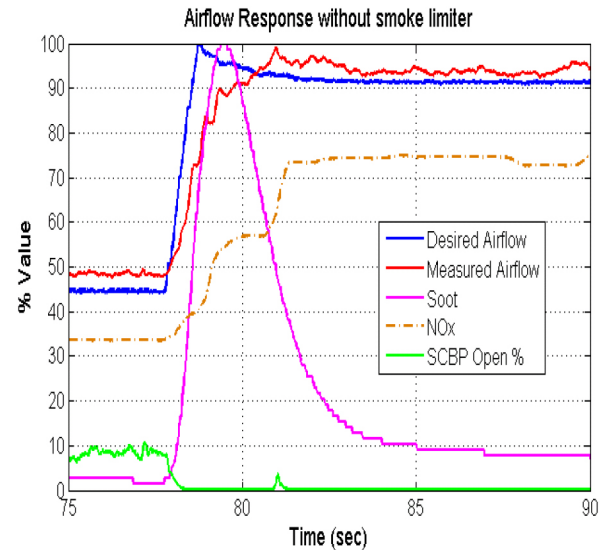


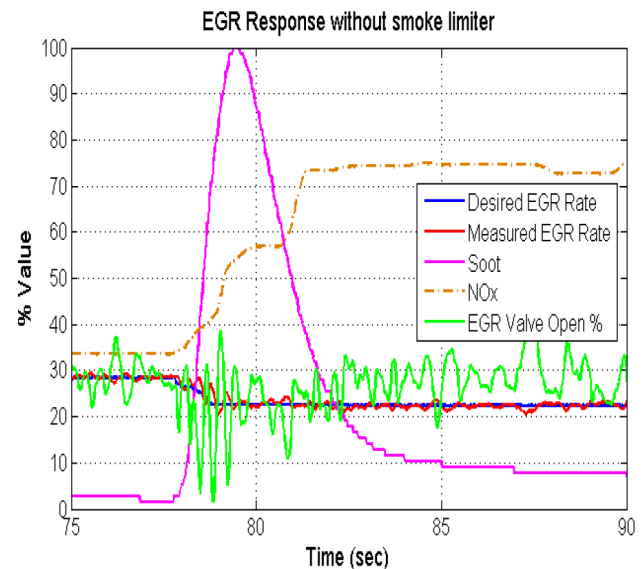Fig 8. Airflow response without smoke-limiter with SC DR = 3.8



Fig 9. EGR response without smoke-limiter with SC DR = 3.8

**Torque Response with smoke limiter**



Fig 10. Torque response with smoke-limiter with SC DR = 3.8

**Airflow Response with smoke limiter**



Fig 11. Airflow response with smoke-limiter with SC DR = 3.8

**EGR Response with smoke limiter**



Fig 12. EGR response with smoke limiter with SC DR = 3.8

## Case 2: SC DR = 5.0

Fig.13, Fig.14 and Fig.15 shows the response of the engine and the controllers to the commanded torque profile without smoke-limiter. In terms of soot-generation, a behavior similar to case 1 is observed. But with a higher drive-ratio, airflow builds up is faster when compared to case 1. As a result, the soot-spike is lower than case 1. With a higher drive ratio, initially, the supercharger recirculation valve is opened to a higher percentage (30%) (Fig. 14) when compared to case 1 where it was 10% (Fig. 8). In order to meet airflow command at higher drive ratio, the valve has to travel slightly more to be completely closed when compared to case 1. Also, as shown in Fig.15, there is some undershoot in EGR rate due to faster build-up of air when compared to case 1 which causes some observable oscillations in EGR controller response.
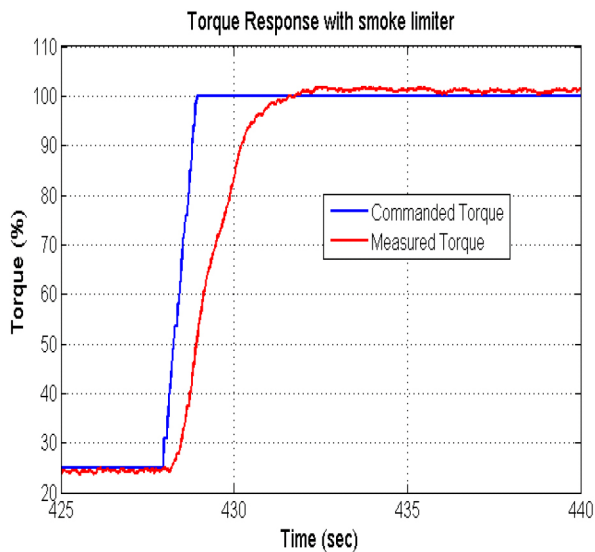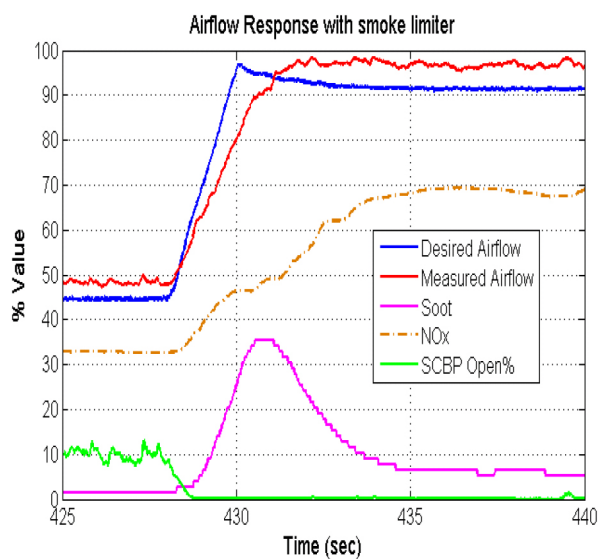
**Torque Response without smoke limiter**



Fig 13. Torque response without smoke-limiter for SC DR = 5.0
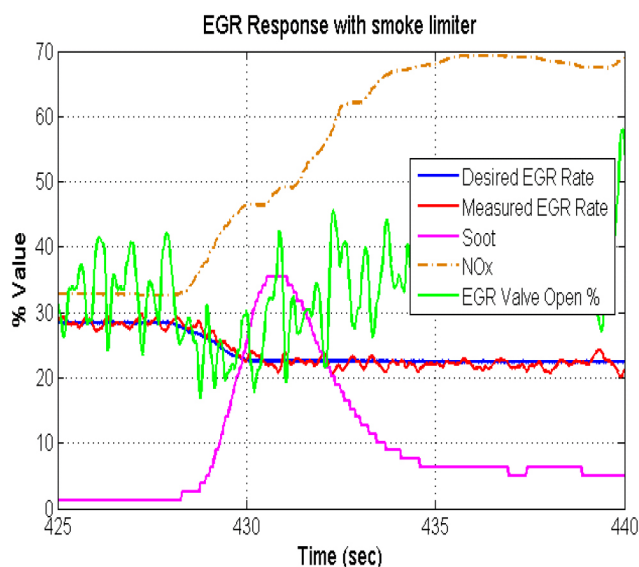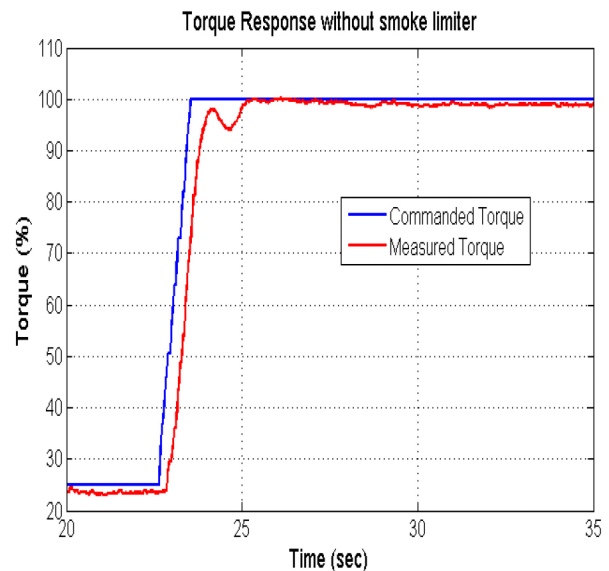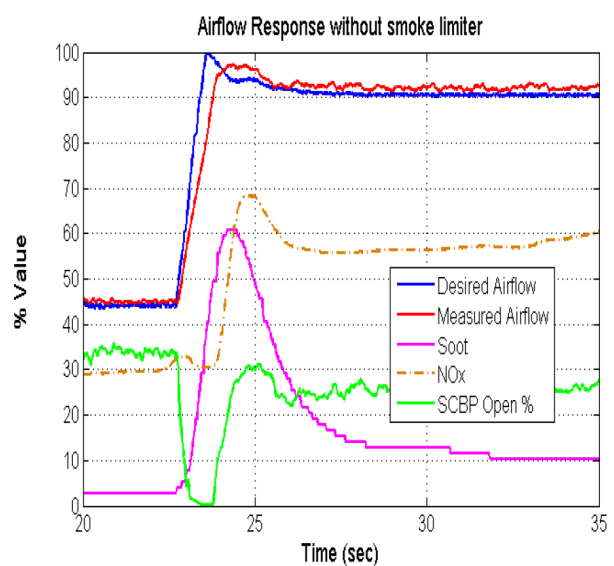
**Airflow Response without smoke limiter**



Fig 14. Airflow response without smoke-limiter for SC DR = 5.0

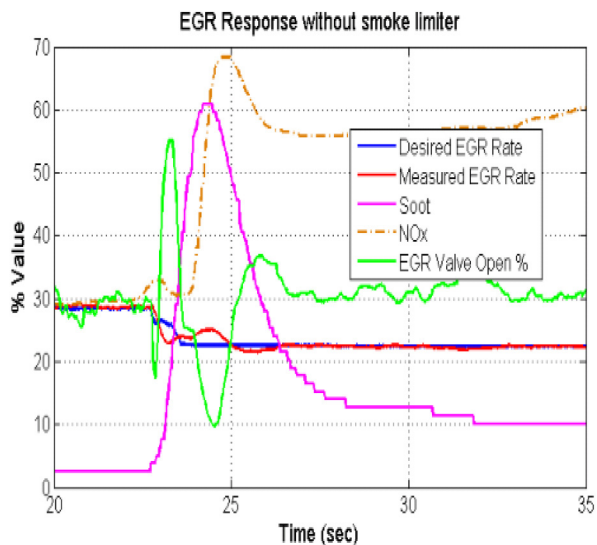Fig 15. EGR response without smoke-limiter for SC DR = 5.0



Fig 17. Airflow response with smoke-limiter for SC DR = 5.0

Fig.16, Fig.17 and Fig.18 show the response of the engine and the controllers to the commanded torque profile with smoke-limiter. When compared to case 1, due to the capability of higher drive ratio to push more air, airflow requirements are met easily and airflow closely follows the set-point, thereby resulting in lower soot generation. Furthermore, in smoke-limiting mode, airflow and EGR set-point ramps are gradual than in non-smoke limiting mode, thereby making them easier to control. An interesting observation during smoke-limiting maneuver is that the SC-recirculation valve does not close completely thanks to the more gradual airflow demand ramp-up. The airflow set point is easily met with the higher drive ratio and a slightly open SC-recirculation valve. In Fig.18, the oscillations in EGR controller response are reduced when compared to Fig.15 due to the more gradual ramp in airflow as well as EGR set-points.
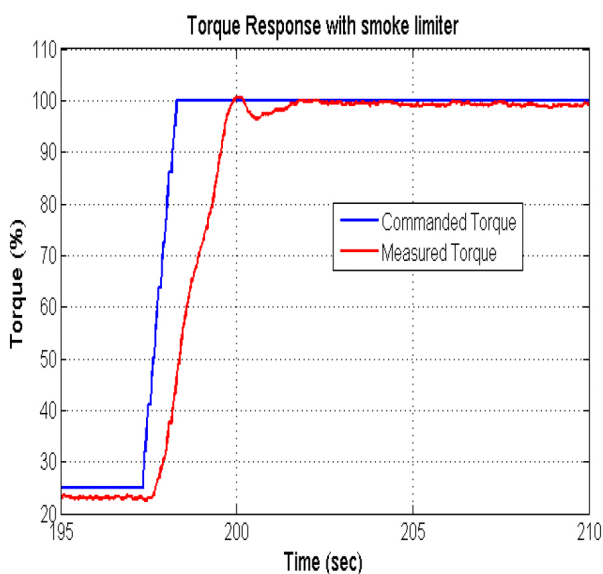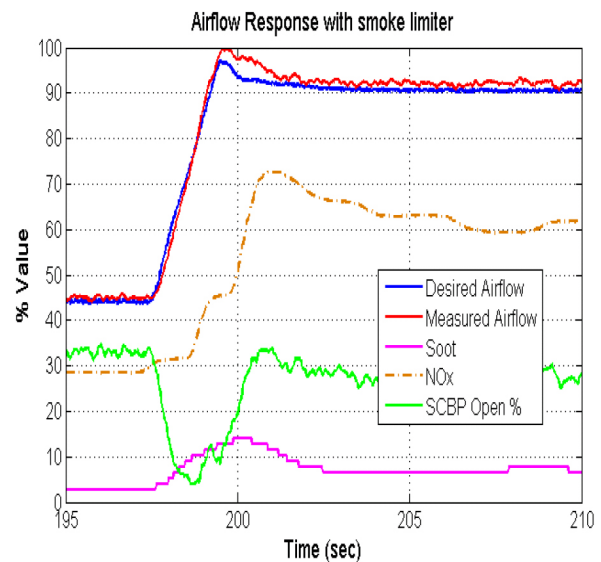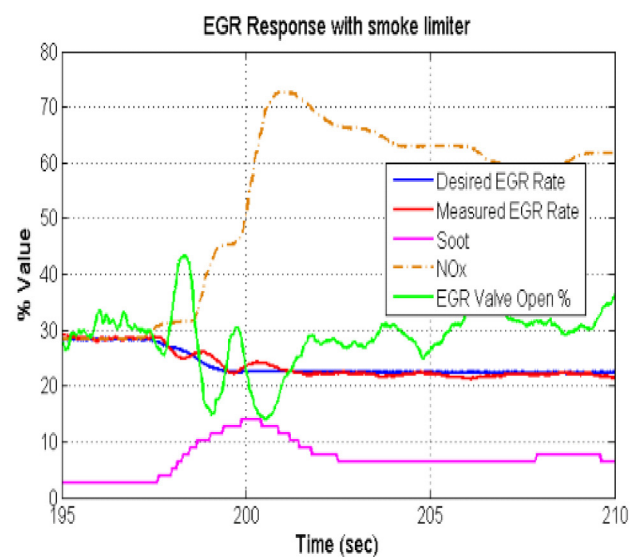


Fig 18. EGR response with smoke-limiter for SC DR = 5.0

## Summary

Table 2. Transient maneuver performance and emission results with different SC Drive Ratios

| SC Drive Ratio | Smoke Limiter Enabled | Torque Response Time (sec) | Airflow Response Time (Sec) | Soot Spike Amplitude (peak - settling value %) | NOx spike Amplitude (peak - settling value %) |
|---|---|---|---|---|---|
| 3.8 | No | 1.3 | 1.8 | 91 | None |
| 3.8 | Yes | 2.5 | 2 | 34 | None |
| 5.0 | No | 0.7 | 0.55 | 50 | 7 |
| 5.0 | Yes | 1.5 | 0.1 | 7 | 9 |



Fig 16. Torque response with smoke-limiter for SC DR = 5.0

Table 2 summarizes the performance and emission results from transient maneuvers tested on the 4.9L research multicylinder engine. The initial results show very good potential for managing transient

operation. There are numerous opportunities to improve the performance -response and emission - during transients by using transient modifiers and AFR based smoke limiter. Further improvements in the engine design supporting reduction of air-flow path volume will help to meet air-flow requirements faster.

## Catalyst Light-Off Mode

In order to satisfy modern emissions regulations, light duty and chassis-certified diesel vehicles rely on cold start techniques for rapid warmup and light-off of the emissions aftertreatment system. With conventional diesel engines, the limited ability to modify trapped conditions, along with the need for stable combustion limit the potential for achieving high exhaust temperatures. This is evidenced by strategies in use for the latest level heavy duty emissions certifications, where manufacturers have implemented fuel injection directly into the exhaust stream to warm diesel oxidation catalysts. In contrast, the opposed-piston two-stroke diesel can generate trapped conditions to achieve stable combustion (COV of IMEP <5%). and high exhaust temperatures simultaneously.

In a previous publication at the Emissions 2012 conference [2] API presented the results from a single cylinder study about the ability to achieve the very high exhaust gas temperatures by leveraging the air and combustion system flexibility of the OP engine.

In 2012, during this previous study the boundary conditions and the air system capability were simulated using GT power. These conditions were then applied to the single cylinder engine and after multiple convergence loops between the test cell and the simulation it was predicted that a multi-cylinder was going to be able to deliver the necessary flexibility to manage exhaust gas temperatures.

At that time it was predicted that a multi-cylinder was able to achieve 410°C after the turbine while at idle and producing very low emissions.

The Achates Power catalyst light-off control scheme seeks to maximize exhaust enthalpy, while retaining good combustion stability. The emissions control during this operating mode are critical as the aftertreatment system will not yet be functioning. The NOx production in particular needs to be minimzed after a cold start while the SCR aftertreatment is getting up to temperarture.

The strategy seeks to absorb crankshaft energy with the supercharger, which also provides high boost pressure, enabling beneficial scavenging characteristics. In a vehicle application, additional energy also could be absorbed by fully engaging the vehicle's charging system, storing some of this energy for later re-use. By creating the heat within the combustion cycle, any number of crankshaft driven accessories can be used to absorb additional energy, increasing the amount of enthalpy available for warming the emissions aftertreatment equipment.

By using the backpressure valve and the supercharger, the scavenging and trapping characteristics of the engine at idle can be altered significantly. The so-called modified delivery ratio $\Lambda^* = m_{del}/\rho_{del}V_{tr}$ is

the ratio of the mass of the delivered air $m_{del}$ (fresh air and external EGR) over the mass of the trapped charge $\rho_{del} \cdot V_{tr}$ (density of the delivered charge multiplied by the trapped volume at port closing). Figure 19 represents the amount of cooled fresh charge (in blue) in the combustion chamber compared to the amount of hot residual charge (in red). As the modified delivery ratio is increased, the percentage of fresh charge trapped in the cylinder is increased and the quantity of residual charge from the previous cycle decreases[2].
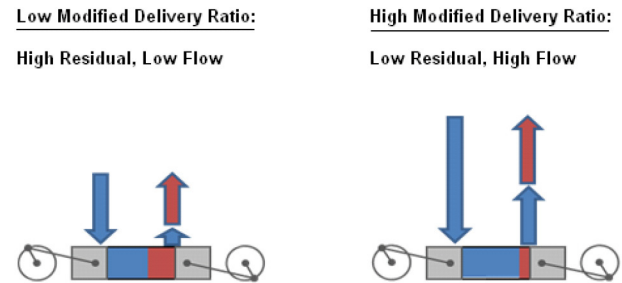


Fig 19. Graphical representation of varying the modified delivery ratio

Running the engine at low modified delivery ratios yields significant internal residuals (shown in Figure 19), which in turn increases the trapped temperature of the charge. In this operating mode, a sufficiently high intake manifold pressure is required to achieve an adequate air-fuel ratio and good combustion stability during light load operation, low ambient temperature operation and extreme cold starts. The mixture is kept lean, though significantly less so than the standard idle. (The reference condition called standard idle does not represent the actual idle speed and load that will be used in a vehicle but merely a reference condition at similar speed and load to the the catalyst warm-up mode showcased in this paper.) When coupled with a series of split, late injections, this creates a significant temperature rise in the exhaust for catalyst light-off and management, yielding good conversion efficiency for aftertreatment performance [2]. The strategy yields low NOx emissions - more than six times lower than the reference condition - and noise as well. The boundary conditions are given in Table 3, while the performance metrics are shown in Table 4.

Table 3. Comparison of Boundary Conditions

| OPERATING CONDITIONS | LIGHT-OFF | STD IDLE | |
|---|---|---|---|
| Speed | 1098 | 1000 | (rpm) |
| Delivered Air Flow | 140.3 | 171.2 | (kg/hr) |
| Delivered Fuel Flow | 8.2 | 1.9 | (kg/hr) |
| Exhaust Mass Flow | 148.5 | 173.1 | (kg/hr) |
| Injected Fuel Mass | 41.6 | 10.4 | (mg/rev/cyl) |
| SOI | 16.2 | -12.9 | (deg aMV) |
| Injection Duration | 30.7 | 9.0 | (deg) |
| Injection Pressure | 2149 | 646 | (bar) |

| AVERAGE GAS TEMPERATURES | LIGHT-OFF | STD IDLE | |
|---|---|---|---|
| Intake Manifold Inlet | 41.1 | 40.3 | (C) |
| Intake Manifold | 40.4 | 39.4 | (C) |
| Exhaust Manifold | 532.3 | 126.9 | (C) |
| Turbine Outlet | 493.3 | 116.5 | (C) |

| AVERAGE GAS PRESSURES | LIGHT-OFF | STD IDLE | |
|---|---|---|---|
| Intake Manifold | 2.10 | 1.19 | (bar) |
| Exhaust Manifold | 2.09 | 1.14 | (bar) |

Table 4. Comparison of Performance Output

| OUTPUTS | LIGHT-OFF IND | LIGHT-OFF BRAKE | STD IDLE IND | STD IDLE BRAKE | |
|---|---|---|---|---|---|
| MEP | 2.34 | 0.09 | 1.50 | 0.15 | (bar) |
| PMEP | 0.77 | | 0.17 | | (bar) |
| FMEP | 1.46 | | 0.76 | | (bar) |
| COV of IMEP | 3.90 | | 2.47 | | (%) |
| Thermal Efficiency | 21.5 | 0.8 | 39.9 | 5.6 | (%) |
| Power | 21.1 | 0.8 | 8.9 | 1.3 | (kW) |
| Torque | 183.9 | 6.7 | 85.1 | 6.5 | (N-m) |
| Peak Pressure | 70.1 | | 57.0 | | (bar) |
| Loc. of Peak Pressure | -0.3 | | 2.5 | | (deg aMV) |
| MPRR | 2.2 | | 4.4 | | (bar/deg) |
| Loc. of MPRR | -12.8 | | -1.0 | | (deg aMV) |

| EMISSION LEVELS | LIGHT-OFF RAW | LIGHT-OFF EI | STD IDLE RAW | STD IDLE EI | |
|---|---|---|---|---|---|
| $CO_2$ | 9.7 | 3103.8 | 2.1 | 3137.6 | (%)/(g/kg fuel) |
| CO | 2005 | 41.0 | 199 | 18.9 | (PPM)/(g/kg fuel) |
| NOx | 28 | 1.0 | 181 | 27.4 | (PPM)/(g/kg fuel) |
| THC | 109 | 1.1 | 30 | 1.4 | (PPM)/(g/kg fuel) |
| AVL Soot Density | 62.7 | n/a | 0.9 | n/a | (mg/m³) |

| EMISSIONS-BASED CALCULATIONS | LIGHT-OFF | STD IDLE | |
|---|---|---|---|
| Exhaust Enthalpy | 21.1 | 4.7 | (kW) |
| Delivered AFR (carbon-based) | 20.3 | 96.9 | (-) |
| Trapped AFR | 27.2 | 155.4 | (-) |
| Combustion Efficiency | 98.9 | 99.4 | (%) |
| EGR Rate | 18.1 | 32.3 | (%) |
| Scavenging Efficiency ($\eta_{sc}$) | 54.4 | 78.4 | (%) |
| Trapped Residual Fraction | 45.6 | 21.6 | (%) |

Table 3 and 4 shows that the exhaust temperatures are raised to almost 500 degrees Celsius, which is extremely favorable for exhaust aftertreatment light-off. Table 4 shows the emission and performance levels of the two modes. Also on table 4 the exhaust enthalpy is calculated which shows a 300% increase with the catalyst warm-up mode. The combustion stability is excellent with the measured COV of IMEP remaining below 4%. The light off mode also has much lower NOx levels than the standard idle condition. With a pressure rise rate at 2.2 bar/degree, we can expect low combustion noise as well. Figures 20 and 21 illustrate the divergence of the combustion strategies by overlaying the injection strategy, the cylinder pressure trace, and the rate of heat release in one plot for each strategy. It is interesting to note that the injector layout helps prevent any cylinder liner fuel impingement. Because the injectors are facing each other each respective fuel spray momentum is cancelled out by the other. This is also confirmed by the negligible fuel in oil dilution measurements.

While the standard idle calibration has room for further optimization, it is clear that the light-off mode offers significant advantages that can be activated almost immediately after startup or during any extended idle where it is necessary to maintain the temperature of catalysts. In the case of the catalyst warm-up strategy, the injection events are delayed far down the expansion stroke, while still providing enough time for the combustion to complete before the exhaust ports open. It is possible to attain further temperature increases if higher levels of NOx or soot are acceptable for short periods in exchange for activating the after treatment equipment.
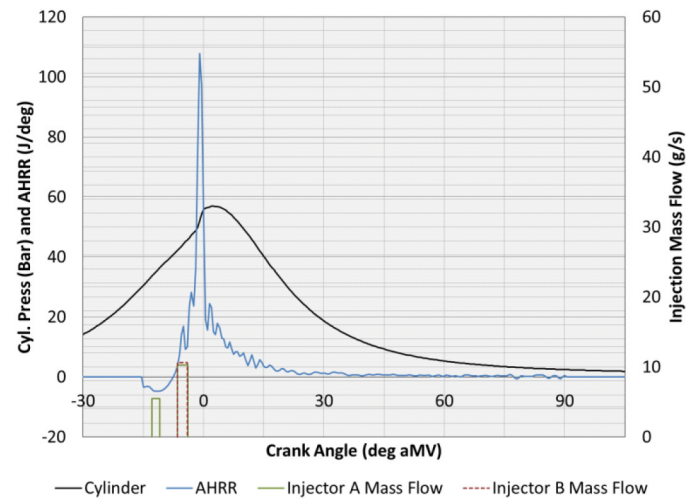


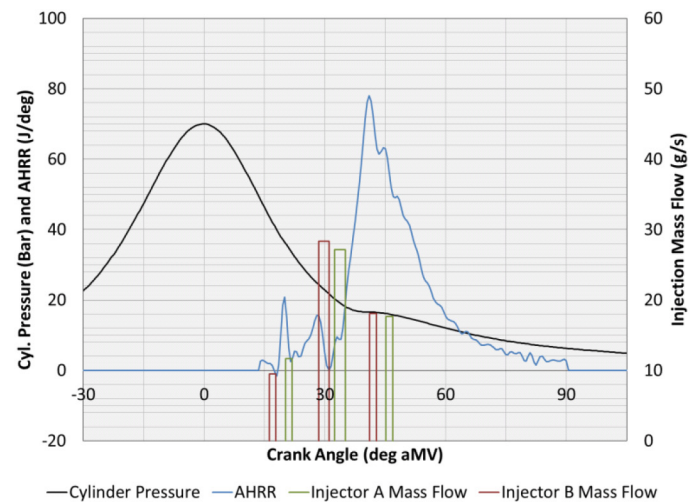Fig 20. Standard Idle Mode Combustion Strategy



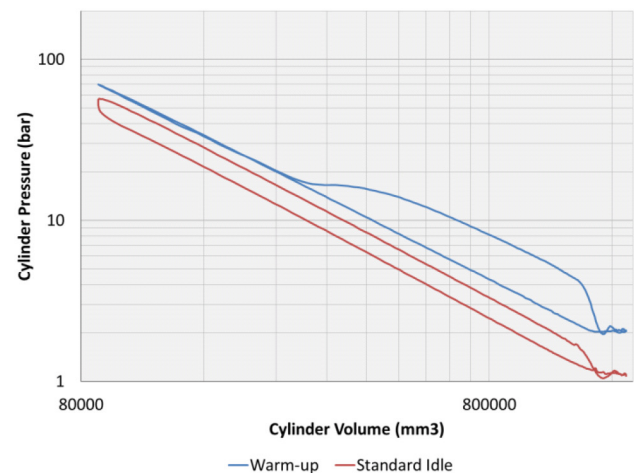Fig 21. Catalyst Warm-up Mode Combustion Strategy



Fig 22. Comparison of Two Combustion Mode P-V Diagrams

Figures 20, 21 and 22 graphically show the difference in heat release and cylinder pressure to volume relationship that come from this control strategy compared to the base calibration mode. The heat release with the catalyst warmup mode is late and longer, maximizing exhaust enthalpy. This performance should yield fast light-off times for exhaust aftertreatment while minimizing the accrued emissions during the light-off process.

There are additional operating conditions where the strategy could be utilized to enhance ability to comply with emissions. During extended motoring operation, catalysts often cool below their activation temperature. Cyclic use of the feature could be used to maintain temperatures so that when the driver torque demand is reintroduced that there is no subsequent spike in emissions.

The catalyst warm-up strategy illustrates the extreme flexibility of the opposed piston engine. The ability to modify trapped conditions more than a similar four-stroke engine yields opportunity for cold starts, DPF regeneration, cold ambient operation, or any other situation where high exhaust temperature is advantageous.

## Summary/Conclusions

With the basic steady state engine performance, fuel economy and emissions demonstrated with a multicylinder engine Achates Power started exploring other critical attributes necessary to deliver a successful engine to the industry.

In this paper we have presented the intial results of the transient control and exhaust temperature management capability of the opposed piston engine

These initial results showed once again that the opposed piston engine architecture has the flexibility to meet the attributes necessary to satisfy the regulatory and customer expectations.

## References

1.  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.

2.  Kalebjian, C., Redon, F., and Wahl, M. "Low Emissions and Rapid Catalyst Light-Off Capability for Upcoming Emissions Regulations with an Opposed-Piston, Two-Stroke Diesel Engine", Emissions 2012Conference.

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