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Low Emissions and Rapid Catalyst Light-Off Capability for Upcoming Emissions Regulations with an Opposed-Piston, Two-Stroke Diesel Engine

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ABSTRACT

In order to satisfy modern emissions regulations, light-duty and chassis-certified diesel vehicles rely on cold start techniques for rapid warm-up of the aftertreatment system. However, with traditional turbocharged four-stroke diesel engines, the limited flexibility of the air management system restricts the control of exhaust temperatures at idle and low load. In contrast, the aircharge system and the gas exchange mechanism of opposed-piston two-stroke engines can be adjusted to allow for a substantially larger range of charge conditions to aid in the control of emissions and exhaust temperatures. Measurements on a single-cylinder, opposed-piston diesel engine confirmed the potential for very high exhaust temperatures at idle and low load points. Based on single-cylinder experimental results, post turbine exhaust temperatures in excess of 360°C at idle have been predicted for a multi-cylinder engine. This in combination with very low NOx emissions and very stable combustion confirms that this technology is able to meet the most stringent emissions regulations. Although the chassis-certified vehicles have to focus heavily on the cold start emissions, there is a benefit to applying the same technique to a dynamometer-certified application.

INTRODUCTION

With the arrival of stricter emissions and fuel efficiency regulations such as LEV III and CAFE requirements, substantial changes in engine technology are required to reach the targets. As a brief summary, LEV III standards will be introduced in phases between 2014 and 2022. By 2022, the fleet average emissions level will be SULEV, 0.030 g/mi NMOG+NOx for all light-duty vehicles (below 8,500 lbs GVW). For chassis-certified, medium-duty vehicles, the standards are a fleet average of 0.170 g/mi NMOG+NOx for GVW 8,500 – 10,000 lbs and 0.230 g/mi NMOG+NOx for GVW 10,000 – 14,000 lbs. 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 [6]. 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 [6].

Along with these emissions reductions, Corporate Average Fuel Economy (CAFE) standards are increasing at about 5% per year, from today's requirements of 27.5 mpg for cars and 23 mpg for trucks to targets of 39 mpg for cars and 30 mpg for trucks in 2016. Eventually, the proposal takes CAFE to 54.5 mpg in 2025 (see Figure 1). These standards on emissions and CAFE create significant challenges.

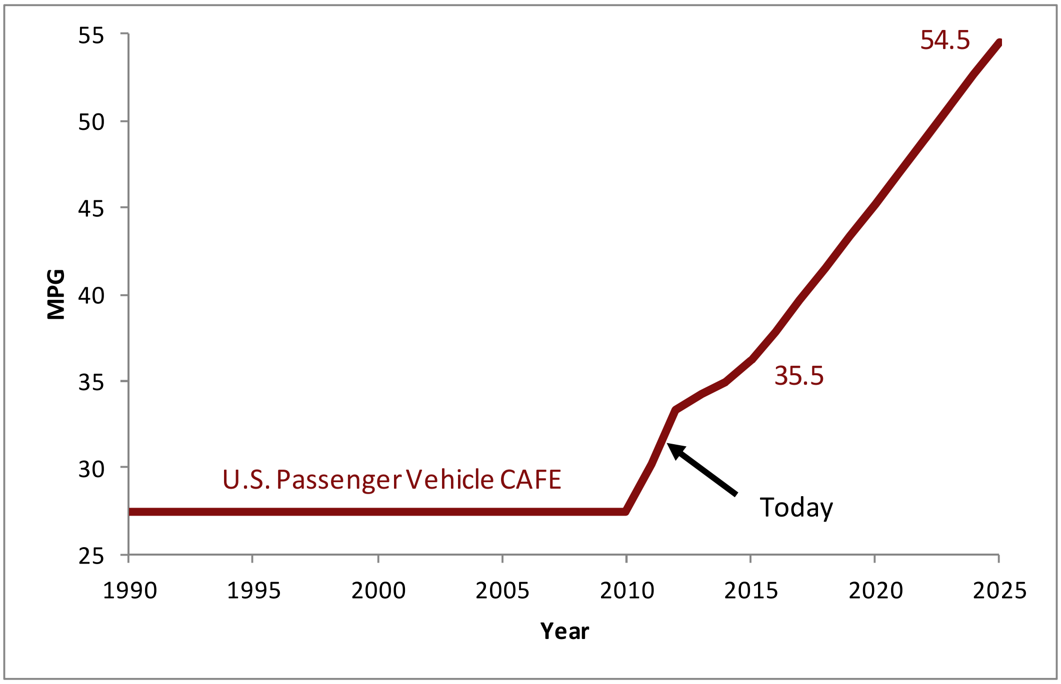

Figure 1 – CAFE targets

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 opposed-piston, two-stroke diesel engine is capable of producing high exhaust temperatures even at idle condition to facilitate a fast catalyst light-off. It also has the advantage of being significantly more fuel efficient once the catalysts are at operating temperature ([1], [2] and [3]).

This paper introduces an exhaust temperature control strategy that leverages some unique characteristics of the opposed-piston, two-stroke engine to achieve higher exhaust temperatures during the catalyst light-off phase than are possible with conventional four-stroke diesel engines. Dynamometer testing on a 1.06 L single-cylinder, opposed-piston research engine under idle conditions has confirmed the capability to achieve exhaust temperatures well above catalyst light-off temperatures while producing very low NOx emissions and having good combustion stability. The higher temperatures will yield shorter catalyst light-off times, reduce cold start emissions and improve overall fuel economy and CO_2 emissions.

Fuel Economy and Emissions Trade-off

The constant drive for lower fuel consumption by vehicle owners and manufacturers to meet increasingly challenging CAFE requirements naturally results in lower exhaust temperatures as more of the fuel energy is being converted into mechanical work instead of exhaust enthalpy. Nevertheless, the tightening tailpipe emissions regulations force engine developers to compromise engine efficiency in two ways. The first compromise is to alter engine combustion to manage engine-out emissions. The second is to provide exhaust temperatures that enable the catalysts to perform the final emissions conversion to meet the tailpipe target. The strategy that has proved to be the least detrimental to fuel consumption is to use 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 [9]. 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 opposed piston, two-stroke diesel engine can generate trapped conditions to achieve stable combustion and high exhaust temperatures simultaneously without specific additional hardware. This key advantage will be explained in detail in subsequent sections. 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 4-stroke diesel engine [2], [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^{o}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 in-use rate or performance ratio, which have been increased from 10% in 2011 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 research engine and the results were subsequently extrapolated to post-turbine conditions via a multi-cylinder model.

SINGLE CYLINDER RESEARCH ENGINE AND METHODOLOGY

A custom single-cylinder research engine, shown in Figure 2, was manufactured in-house and tested on a 300 hp AC dynamometer. The engine had a bore of 80 mm and a stroke of 211.8 mm, resulting in a displaced volume of 1.06 L. The engine was tested at multiple trapped compression ratios, ranging from 16.7:1 to 17.7:1. The port geometry was set to a constant swirl ratio and port timing, and the piston geometry and injection spray pattern had been specified based on analytical combustion simulation results. A common-rail fuel injection system was used, capable of creating injection pressures up to 2000 bar and producing multiple injection events per engine cycle.

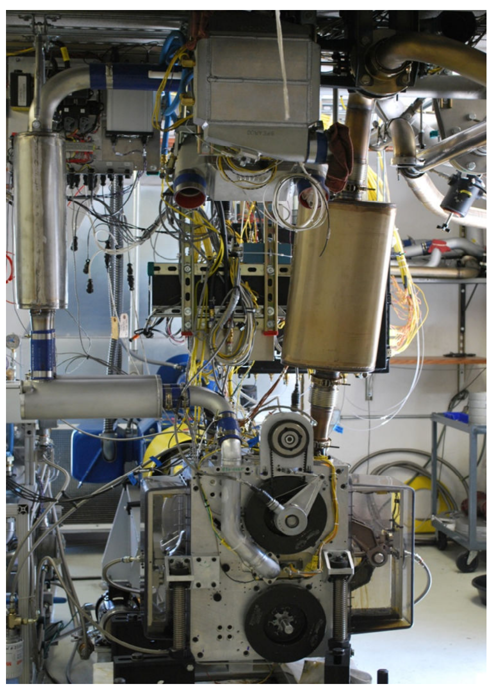

Figure 2 – Single cylinder research engine installed in test cell

Conditioned combustion air and EGR were delivered to the intake manifold of the single-cylinder engine via the system shown in Figure 3. An external air compressor supplied conditioned compressed air, regulated to a controlled pressure, mixing it with exhaust gas taken from the exhaust side of the engine via an EGR pump. This pump pulled the exhaust through a gas-to-water heat exchanger before delivering exhaust gas to the intake stream. The EGR rate delivered to the engine was controlled by the EGR pump speed and a ball valve located downstream of the pump. After the air and exhaust gas had been mixed, the intake gas flowed through a heat exchanger followed by a heater to precisely control the intake manifold temperature. The exhaust manifold pressure was set with a back pressure valve in the exhaust system to simulate the effect of turbo and aftertreatment system.

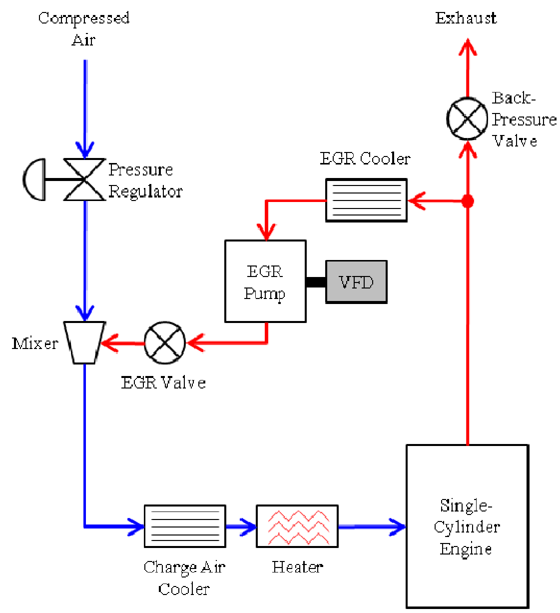

Figure 3 – Schematic of the air and EGR conditioning system

In-cylinder pressure was measured at 0.5° crank-angle resolution with a Kistler 6052C piezoelectric pressure transducer coupled to a Kistler 5064 charge amplifier. The cylinder pressure signal was pegged to an average of the intake and exhaust manifold pressures and measured with Kistler 4005B and 4049A high-speed pressure transducers, respectively. Custom in-house software was used to acquire and process the data. A California Analytical Instruments (CAI) emissions analyzer was used to measure the steady-state concentration of five exhaust species (CO_2, CO, O_2, HC and NOx) and intake CO_2. An AVL 415s Smoke Meter provided a measure of exhaust soot content.

The opposed piston, two-stroke diesel engine can be operated in a way that enables significant heat to be expelled into the exhaust system. 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 4 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.

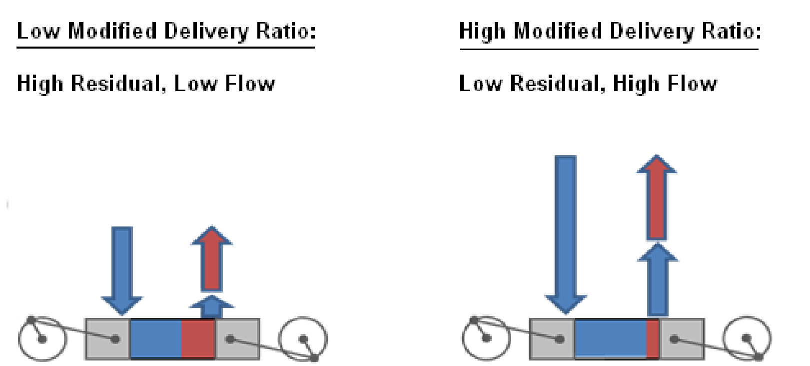

Figure 4 – Graphical representation of varying the modified delivery ratio

Running the engine at low modified delivery ratios yields significant internal residuals (shown in Figure 4), 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. In addition, temperatures can be further enhanced when low modified delivery ratios and high intake manifold pressure is coupled with a late-injection strategy. This creates a significant temperature rise in the exhaust for catalyst light-off and management, yielding good conversion efficiency for aftertreatment performance. This approach also supports in-use rate diagnostic requirements for the aftertreatment as well as the temperature required to regenerate the DPF under all driving conditions, including idle and light loads.

For this strategy to be effective, it is important to precisely control the modified delivery ratio and, therefore, the composition of the trapped charge. The modified delivery ratio is primarily a function of the scavenging characteristics of the engine, the intake manifold conditions and the exhaust backpressure. The intake manifold conditions, in turn, depend on the supercharger speed and recirculation settings and the rate of cooled external EGR (see Figure 6 and related discussion in the following section).

Data from several idle operating points were recorded from the single-cylinder research engine with coolant at about 70°C. Results from three of these operating points are summarized in Table 1. Figure 5 shows the associated trapped temperature profile for each of these three points. This temperature is determined from the measured pressure trace during combustion. The peak combustion temperature is reduced, thereby reducing NOx. Figure 6 shows the measured relationship between the trapped temperature and the exhaust temperature. For approximately the same Indicated Mean Effective Pressure (IMEP), there is a significant reduction in NOx and an increase in exhaust temperature, both of which are favorable for cold start emissions performance.

Variable	Units	Normal	Warm-Up Point-1	Warm-Up Point-2
AVL FSN	(FSN)	0.16	0.49	0.13
AVL Noise	(dB)	84.4	81.9	80.6
AVL Soot Concentration	(mg/m^3)	2.2	NA	1.7
Max Cyl Pressure Rise Rate	(bar/degCA)	3.38	2.76	3.15
Intake Port Closing Pressure	(bar)	1.49	2.27	2.91
IMEP	(bar)	3.45	3.58	3.65
Injection Volume for Pilot 1	(mm^3)	3.0	2.0	2.0
Injection Volume for Pilot 2	(mm^3)	3.0	2.0	2.0
Carbon Based AFR	(none)	28.5	20.1	20.2
CO	(ppm)	111	1725	1820
Combustion Air Flow	(kg/h)	33.0	32.7	42.5
Crankshaft Speed	(RPM)	1017	1020	1020
EGR %	(%)	37.4	40.3	38.9
Exhaust Manifold Gas Temp	(C)	234	352	398
Injection Timing	(°BTDC)	-2.0	-12.0	-19.0
Injection Volume	(mm^3/rev)	22.1	30.1	36.6
Modified Delivery Ratio	()	0.67	0.48	0.48
NOx	(ppm)	115	27	22
Exhaust Manifold Pressure	(bar (abs))	1.39	2.13	2.66
Fuel Rail Pressure	(bar)	905	1200	1200
Scavenging Efficiency	(%)	57.8	44.3	44.3
THC	(ppm)	155	266	439
Trapped AFR	()	34.7	19.4	20.1
Trapped Mass	(mg/cycle)	1067	1334	1621
Trapped Residual Fraction	(%)	42.2	55.6	55.7
Trapped Temperature	(C)	138	235	262
	()	0.017	0.020	0.032

Table 1 –Key parameters and measured results for three operating points on research engine

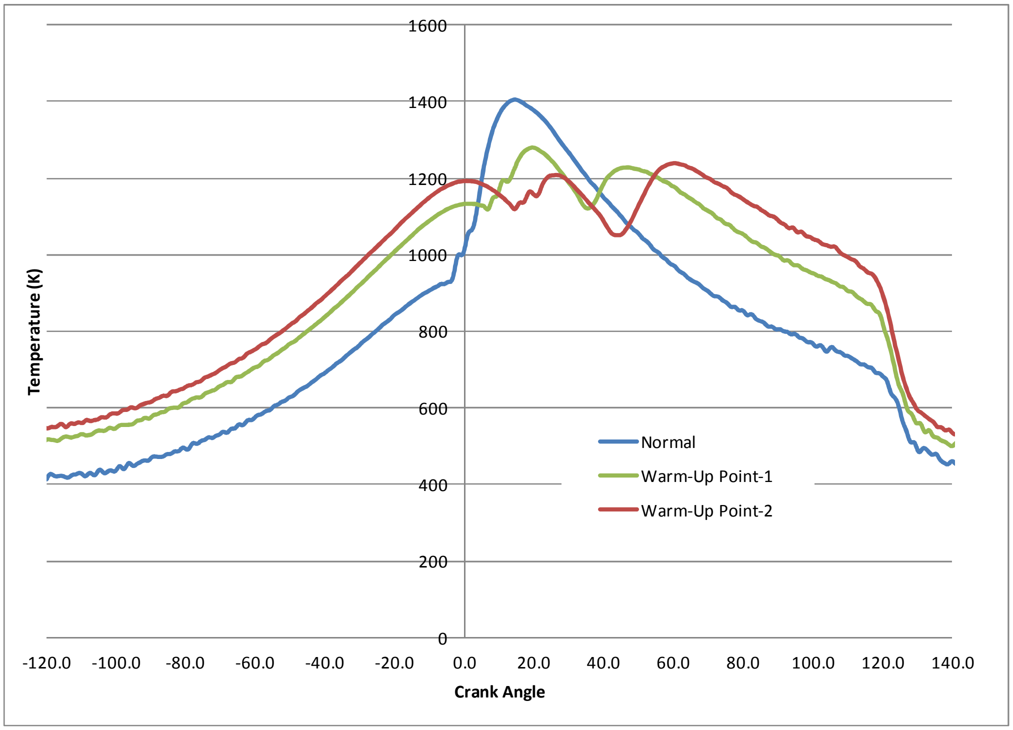

Figure 5– Combustion temperature profiles for three operating points – calculated from pressure trace

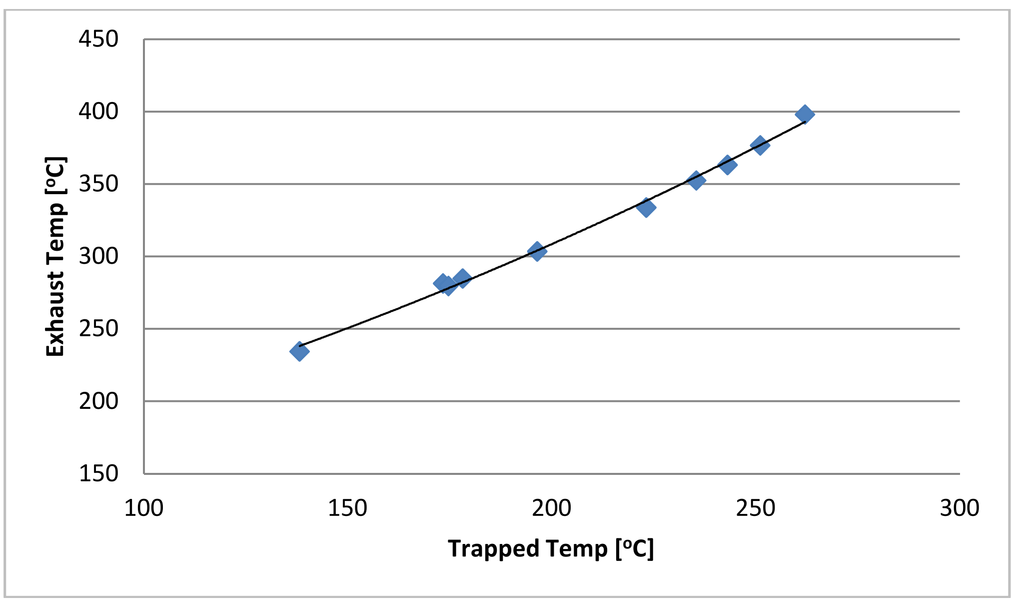

Figure 6 – Relationship between trapped temperature and exhaust temperature

INTERFACE MODEL

Friction and pumping energy losses, which represent the difference between indicated work and brake work, are specific for each engine configuration and do not translate from a single-cylinder to a multi-cylinder engine by simple multiplication. In order to predict the brake-specific performance of a multi-cylinder engine based on single-cylinder combustion results, an "interface model" was created in 1D engine system simulation software. Specifically, a three-cylinder, opposed-piston engine was modeled with 3.2 L displacement and with rated power and torque of 225KW@3800 rpm and 700Nm@2000 rpm, respectively. This model was correlated to the experimental boundary conditions and measured in-cylinder pressure trace to provide multi-cylinder-based predictions of the friction and pumping work required at the operating point measured on the dynamometer. The results from the interface model delivered predictions of multi-cylinder, brake-specific performance and emissions parameters based on the measured single-cylinder results.

Figure7 shows the schematic of the input data and assumptions of the interface model. The combustion chamber geometry, the piston motion and the porting profiles in the multi-cylinder model were identical to the single-cylinder research engine, while the number of cylinders and associated manifold configurations were application specific. Measured test cell values—such as engine speed, fuel flow rate, air flow rate, EGR percentage and intake pressures and temperatures—matched the measured modeled values. The rate of heat release was derived from the measured cylinder pressure and input directly into the combustion sub-model. Assumptions for the air-handling equipment, charge cooling components and aftertreatment system were used in the pumping loss prediction. The Chen-Flynn mechanical friction model was based on the mechanism design and analysis and was correlated to experimental friction results. The work needed to drive all accessories, including the supercharger, was also taken into account.

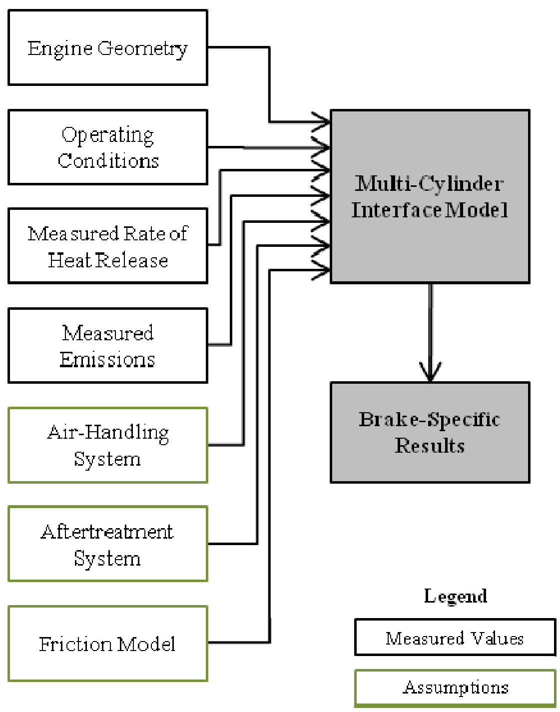

Figure 7 – Multi-cylinder interface model input data flow

The interface model air-handling system (Figure 8) consisted of a supercharger, a turbocharger and a charge air cooler after each compression stage. The size and characteristics of the air-handling system components were application specific. The compressor and turbine were modeled as 'mapless' components with user-specified efficiencies that are consistent with the operating point and available turbocharger supplier data, and the supercharger model used a full map obtained from a supplier. A two-speed drive mechanism was assumed for the engine-supercharger connection. Different drive ratios for the supercharger are useful for maintaining high thermal efficiency over the entire engine map (see also [2], [3]), for increasing low-speed torque and for enhancing the cold start capability of the engine. A supercharger recirculation loop and valve were included to control the inlet manifold pressure and a variable geometry turbine was modeled for over-boost, over-speed protection and back pressure control, as commonly used in four-stroke diesel applications today.

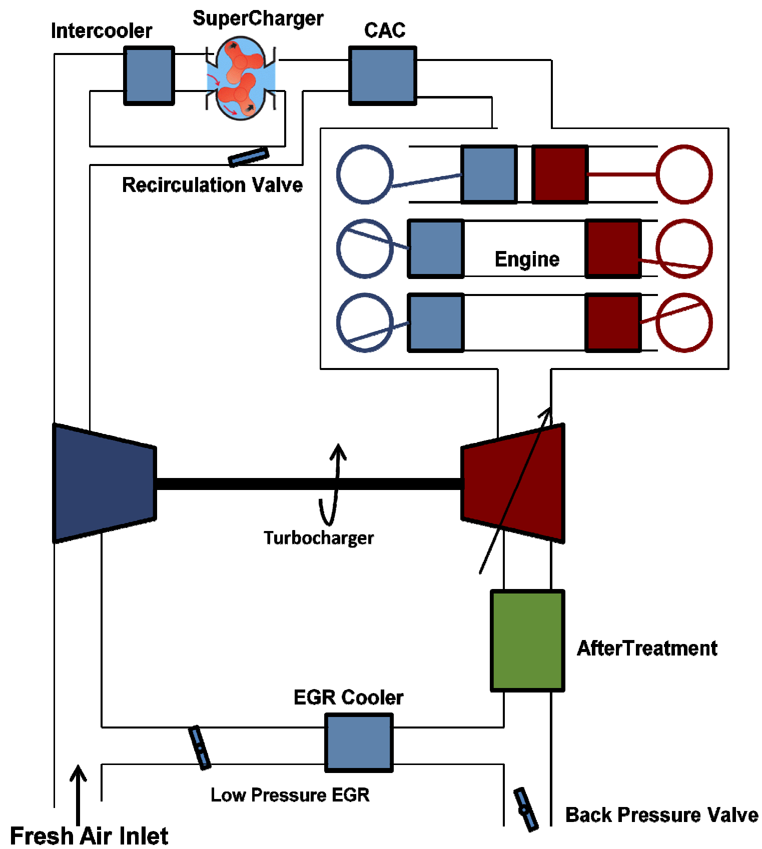

Figure 8 – Air-handling system configuration modeled

Low Pressure EGR was introduced into the intake system before the compressor. It was assumed that the charge air coolers were of the air-to-water type and were located on a secondary low temperature coolant circuit. The charge air coolers' effectiveness values are set to 90%, which is a valid assumption even with a certain degree of cooler fouling. The final charge air cooler was 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.

For each operating condition, the compressor and turbine efficiencies were specified based on supplier data and the two-stroke scavenging schedule was set to match measured results at the given speed and engine load. The cylinder pressure trace, intake air flow per cylinder and EGR percentage were matched to the experimental results when using the measured rate of heat release by adjusting the following parameters: supercharger drive ratio (to one of the prescribed values), supercharger recirculation valve position, turbine effective diameter and EGR valve position.

The measured data from the single-cylinder research engine was then applied to the multi-cylinder interface model. The results from the model are shown in Table 2. The goal of Warm-Up Point-1 and Point-2 was to generate significant heat with low NOx and low BMEP, to be used for an idle point. Low modified delivery ratio and high intake manifold pressure coupled with retarding injection timing proved extremely beneficial. Warm-Up Point-1 was able to meet the desired targets on the set of hardware shown in Figure 10 above. Although Warm-Up Point-2 showed potential for higher post turbine exhaust temperatures, the temperature limit of the selected supercharger was exceeded. The obvious remedy would be to use a different set of hardware with a higher temperature limit or to improve the performance of the charge air cooler upstream of the supercharger. Also, the brake power for Warm-Up Point-2 turned out to be slightly below zero due to a higher than expected power consumption of the supercharger required to produce the intake manifold conditions used in the test cell.

For Warm-Up Point-1, the exhaust enthalpy increased by 60% compared to the baseline and the post turbine exhaust temperature was predicted to be 367^{o}C, which represents an over 120^{o}C increase compared to the baseline and more than enough to light-off a catalyst.

Case Name	Units	Baseline	Warm-Up Point-1	Warm-Up Point-2
Speed	rpm	1016	1021	1016
IMEP	bar	3.45	3.57	3.66
BMEP	bar	2.62	0.53	-0.32
Indicated Power	kW	18.6	19.4	19.8
Brake Power	kW	14.2	2.9	-1.7
Indicated Thermal Eff	%	47.1	36.2	30.7
Brake Thermal Eff	%	35.8	5.4	NA
Friction Loss	%	6.1	4.7	4.1
Pumping Loss	%	5.2	26.1	29.2
Exhaust + Coolant Heat Losses	%	52.9	63.8	69.3
ISFC	g/kWh	177.7	231.2	273.2
Fuel Flow Rate	kg/h	1.10	1.49	1.80
NOx Flow Rate	mg/s	3.9	0.9	0.9
SOOT Flow Rate	mg/s	0.1	NA	0.1
CO Flow Rate	mg/s	2.7	40.9	53.2
HC Flow Rate	mg/s	2.0	3.4	7.0
Peak Cylinder Pressure	bar	68.1	78.4	92.6
Start Of Injection	(°BTDC)	-2	-12	-19
50% Mass Fraction Burnt	degCA	6.8	24.0	46.2
MRPR	bar/degCA	3.4	2.8	3.1
AFR	-	28.7	20.2	20.3
EGR Rate	%	37.2	40.5	38.8
Intake Temp	degC	37.9	40.0	41.5
Intake Manifold Pressure	bar	1.40	2.24	2.78
Exhaust Temp	degC	277	387	431
Exh Temp Post Turb	degC	246	367	410
Exhaust Press	bar	1.37	2.22	2.76
Exh Press Post Turb	bar	1.11	1.85	2.31
Total Exhaust Massflow	kg/h	98.4	95.1	115.3
Specific Exhaust Enthalpy	kJ/kg	236	378	427
Rate of Exhaust Enthalpy	KW	6.5	10.0	13.7

Table 2 – Modeled results for three operating points on multi-cylinder engine

CONCLUSION

Due to regulatory changes in both tailpipe emissions for LEV III and CAFE requirements, there is an acute need for a more fuel-efficient engine solution in conjunction with lower tailpipe emissions for chassis-certified applications. In that regard, a faster catalyst light-off during cold start is mandatory for a successful emissions strategy. Besides being very efficient when the catalysts are at temperature, the opposed-piston, two-stroke diesel engine can be operated in a way that generates extremely hot, high enthalpy exhaust to aid during catalyst light-off, light load or idle conditions. Running the engine with low modified delivery ratios in combination with adequate intake manifold boost pressures and a late injection strategy yielded a turbine outlet temperature of 367^{o}C (specific exhaust enthalpy of 378 KJ/kg) at 1000 rpm and only 0.53 bar BMEP. This was accomplished while only producing 27 ppm of NOx, which corresponds to a humidity corrected NOx flow-rate of 0.9 mg/s. The flexibility afforded by the large range of trapped conditions that can be realized with an opposed-piston, two-stroke diesel engine is a key enabler to control emissions, particularly during cold start conditions. The described methodology provides similar benefits for dynamometer-certified applications as well as significantly improved overall fuel efficiency. In summary, the opposed-piston, two-stroke diesel engine represents a viable alternative to conventional engine technology to more easily meet the challenges of stricter emissions and fuel efficiency regulations.

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