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California Energy Commission
Clean Transportation Program

FINAL PROJECT REPORT

Ultra-Low Carbon Powertrain Optimized for Medium-Duty E-85 Cargo Vans

Prepared for: California Energy Commission
Prepared by: Cummins, Inc.

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PREFACE

Assembly Bill 118 (Núñez, Chapter 750, Statutes of 2007) created the Clean Transportation Program, formerly known as the Alternative and Renewable Fuel and Vehicle Technology Program. The statute authorizes the California Energy Commission (CEC) to develop and deploy alternative and renewable fuels and advanced transportation technologies to help attain the state's climate change policies. Assembly Bill 8 (Perea, Chapter 401, Statutes of 2013) reauthorizes the Clean Transportation Program through January 1, 2024, and specifies that the CEC allocate up to $20 million per year (or up to 20 percent of each fiscal year's funds) in funding for hydrogen station development until at least 100 stations are operational.

The Clean Transportation Program has an annual budget of about $100 million and provides financial support for projects that:

- Reduce California's use and dependence on petroleum transportation fuels and increase the use of alternative and renewable fuels and advanced vehicle technologies.
- Produce sustainable alternative and renewable low-carbon fuels in California.
- Expand alternative fueling infrastructure and fueling stations.
- Improve the efficiency, performance and market viability of alternative light-, medium-, and heavy-duty vehicle technologies.
- Retrofit medium- and heavy-duty on-road and nonroad vehicle fleets to alternative technologies or fuel use.
- Expand the alternative fueling infrastructure available to existing fleets, public transit, and transportation corridors.
- Establish workforce-training programs and conduct public outreach on the benefits of alternative transportation fuels and vehicle technologies.

To be eligible for funding under the ARFVTP, a project must be consistent with the Energy Commission's ARFVTP Investment Plan, updated annually. The Energy Commission issued PON-09-004 to provide funding opportunities under the ARFVTP for Medium & Heavy-Duty Advanced Vehicle Technology. In response to PON-09-004, the recipient submitted application 41, which was proposed for funding in the Energy Commission's Notice of Proposed Awards June 10, 2010. The agreement was executed as ARV-10-044 on September 21, 2011 in the amount of $2,712,131.

ABSTRACT

This Ultra-Low Carbon Powertrain project report describes the design, development, and testing of a prototype powertrain concept fueled by E85 ethanol and targeted to decrease CO2 emissions on a full-fuel-cycle basis by over 50 percent. This project developed a downsized 2.8L engine for use in class 4-6 medium-duty vehicles with power and torque capabilities appropriate for this market. A stop/start system was integrated with an automatic transmission to reduce idle fuel consumption. This project demonstrated that an E85 engine with a close coupled 3-way catalyst is feasible for delivering 2010 criteria emissions levels allowing for a cost competitive platform. System analysis, test cell testing, chassis dynamometer testing, and full vehicle demonstration were used to predict and quantify the results. A peak engine brake thermal efficiency of over 42 percent was achieved which enabled equivalent fuel economy to a gasoline powertrain and equivalent fuel cost to a diesel powertrain. This project concluded that over 50 percent CO2 reduction was achievable on cellulosic biofuels relative to currently available gasoline and diesel vehicles. Recommendations for future work were made to increase the level of field demonstration or examine next steps for improved powertrain efficiency using this E85 powertrain concept as a starting platform.

Keywords: California Energy Commission, Cummins, E85, ethanol, cargo vans, engine technology development.

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EXECUTIVE SUMMARY

This Ultra-Low Carbon Powertrain project was chartered to decrease carbon dioxide emissions on a full-fuel-cycle basis by over 50 percent using cellulosic ethanol as part of an E85 fuel (85 percent ethanol and 15 percent gasoline). It was envisioned that the fuel properties of E85 could be used to their advantage in a purpose-built engine designed for this fuel. This engine would be downsized relative to current engines to increase the specific load on the engine which increases the efficiency engine over a typical vehicle drive cycle. The combination of direct injection of fuel into the cylinder and utilizing a high compression ratio leverage the fuel properties to drive high engine efficiency.

A top down approach was used to determine the engine displacement for this design by looking at the required torque and power for class 4-6 medium duty applications. Bounds were placed on engine displacements which could meet these requirements. It worked out that the displacement range suitable for this project also happened to be the same displacement of a clean sheet diesel engine design being worked by Cummins as part of a Department of Energy grant. The timing worked such that the requirements of the E85 engine could be considered during the design phase of the diesel version to leverage the high cylinder pressure requirements for both platforms. A suite of modeling tools were used to guide the E85 engine design to allow for placement of all the required components, to develop specifications for the engine breathing systems, and to assess the structural needs of the engine.

Test cell testing was used to develop the engine calibration and controls algorithms required to hit the project goals. Test cell testing showed that engine efficiency results for this platform were excellent and exceeded 42 percent brake thermal efficiency. Feasibility to achieve California Air Resources Board's (ARB) new and more stringent 2010 criteria emissions levels for trucks was also demonstrated using the engine dynamometer. Controls for the stop/start system were fine tuned in the test cell to quickly and properly restart the engine as well as control the alternator charging rate to take full advantage of engine deceleration events for energy recovery to the batteries.

With the engine and stop/start system controls defined and tuned during the test cell testing phase, a full vehicle was built up to begin testing the integrated powertrain in a real application. Chassis dynamometer testing was conducted first to get a consistent comparison against current gasoline and diesel powertrains using the same vehicle characteristics. Freightliner Custom Chassis Corporation donated a gasoline MT45 Step Van to use for the baseline testing. This vehicle then had the powertrain removed to allow for install of the E85 powertrain and perform fuel economy and carbon dioxide emissions testing on the same vehicle. Federal Express loaned a diesel-powered vehicle of very similar specifications to use for baseline comparison in the same test environment. The results of this chassis dynamometer testing showed that across 4 different vehicle weights reflective of class 4-6 vehicles and on 4 different drive cycles, the E85 powertrain had similar fuel economy to the gasoline vehicle and similar fuel cost to the diesel power vehicle. Achieving similar fuel economy to the gasoline vehicle was a remarkable accomplishment given the fact that E85 has 30 percent less energy than gasoline per gallon of fuel. The chassis dynamometer testing was also used to quantify the level of carbon dioxide emissions reduction for the E85 powertrain and showed that the project goal of at least 50 percent reduction could be achieved. In fact, a 75-80 percent carbon dioxide reduction was possible relative to the baseline powertrains when cellulosic ethanol was used for the E85 blend.

Finally, a real-world vehicle demonstration was conducted for 6 weeks in the Sacramento area. The vehicle was run on an urban route running through downtown and a rural route out through the countryside. These two routes help to bound the fuel economy potential of this powertrain, and to show that both the engine and stop/start system are fit for these types of applications. The vehicle was run with a gross vehicle weight of approximately 14,000 lbs. and confirmed the fuel economy results achieved on the chassis dynamometer are representative of real-world operation.

This project demonstrated that it is possible to design a cost effective E85 platform than can achieve the performance required for class 4-6 medium duty vehicles and provide significant carbon dioxide reduction relative to the current powertrains available in this market. Future work should be undertaken to further demonstrate this technology through customer field test as well as increased market available of the fuel through adoption of more E85 fueling stations.

The total value of grant ARV-10-044 issued for this project was $2.7 million. Cummins and their corporate partners contributed an additional $2.7 million in private match funding.

CHAPTER 1: Project Purpose and Approach

Project Purpose

The most prolific alternative fuel used in motor vehicles today is ethanol. While millions of passenger cars are capable of using high blends of ethanol (so called "flex-fuel" vehicles), no full-production medium-duty or heavy-duty vehicles have this capability. Cummins believes an optimized E85 engine can be highly competitive on a total cost of ownership basis and deliver over 50 percent carbon dioxide (CO2) emissions reduction relative to current production medium-duty gasoline and diesel powertrains. The goal of this agreement is to design and demonstrate a 2.8L E85 powertrain concept for medium-duty truck operation. The integrated system will be capable of meeting California criteria emission levels within the time frame of the agreement and deliver significant CO2 emissions reduction. The results of this project will establish a technology development path for scalability for medium-heavy-duty vehicles.

The total value of grant ARV-10-044 issued for this project was $2.7 million. Cummins and their corporate partners contributed an additional $2.7 million in private match funding.

Approach

The approach to this project includes developing a high-power density E85 optimized combustion system which takes full advantage of the fuel properties to maximize the efficiency of the engine. It also includes integrating a stop/start system with the engine and transmission such that the engine can be shut down during idle events to minimize fuel consumption at vehicle stop events. Cummins worked through a development path that included determining the powertrain requirements through competitive benchmarking and system modeling work. This component identified high level engine requirements such as power, torque, cylinder pressure, and the displacement range that would be best suited to provide this performance. When the displacement range was identified, it was discovered that it would be possible to partner with another Cummins and United States Department of Energy (U.S. DOE) project at a suitable displacement node to leverage some lower cost components during the design phase. The Advanced Technology Light Automotive Systems (ATLAS) project which was then ongoing had set out to design a 2.8L diesel engine which had similar cylinder pressure and speed range requirements as this project. With requirements defined, the Ultra-Low Carbon Powertrain project entered the design phase along with the ATLAS design team.

While hardware was being procured for the test cell engine build, controls algorithms were rapidly being developed in the prototype space using bench testing and specific rig development. These algorithms were then carried over to the test cell for further refinement on engine.

A test cell engine was built up to develop the engine calibration required to operate this high efficiency engine. Steady state testing was done initially to develop the calibration response surface. This surface was then used for further fine tuning of the controls algorithms to achieve the desired transient response and emissions performance. Once proper emissions control was demonstrated, the stop/start system components were added to the test cell engine to fine tune their integration and operation algorithms.

While test cell testing and development of the E85 engine was ongoing, Cummins used and available production diesel vehicle to develop the stop/start algorithms needed for this project. Both the diesel engine and the E85 engine are controlled by the same Cummins Electronic Control Module so the algorithms could be developed in the diesel environment and then applied almost directly to the E85 engine.

Finally, a demonstration vehicle was put together with the full E85 powertrain. This demonstration vehicle was used for chassis dynamometer testing to compare to similar gasoline and diesel baseline vehicles. It was then used for a real-world demonstration in Sacramento to confirm that the chassis dynamometer results would translate to real world results.

CHAPTER 2: Activities and Results

Engine System Modeling and Analysis

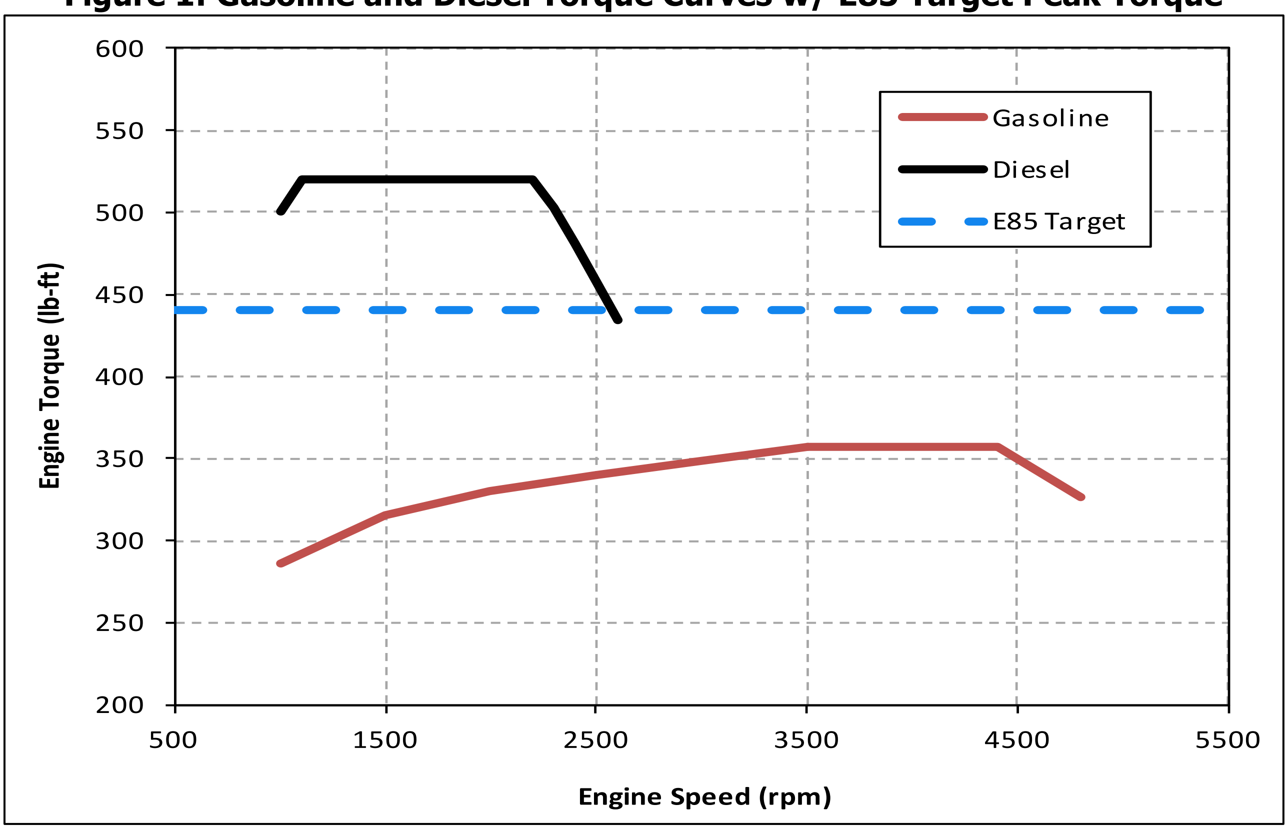
The objective of this task is to determine the major subsystem specifications for an optimized E85 engine targeted at the medium duty step-van market. The output of this task is a detailed technical profile and specific engine architecture that will be used to guide the design effort in Task 3.

Target Application and Maximum Torque and Power Requirements

This demonstration program is focused on the medium duty step-van pick-up and delivery application. Vehicles are available for this application with gross vehicle weight ratings from 14K to 24K pounds covering from Class 4 through Class 6.

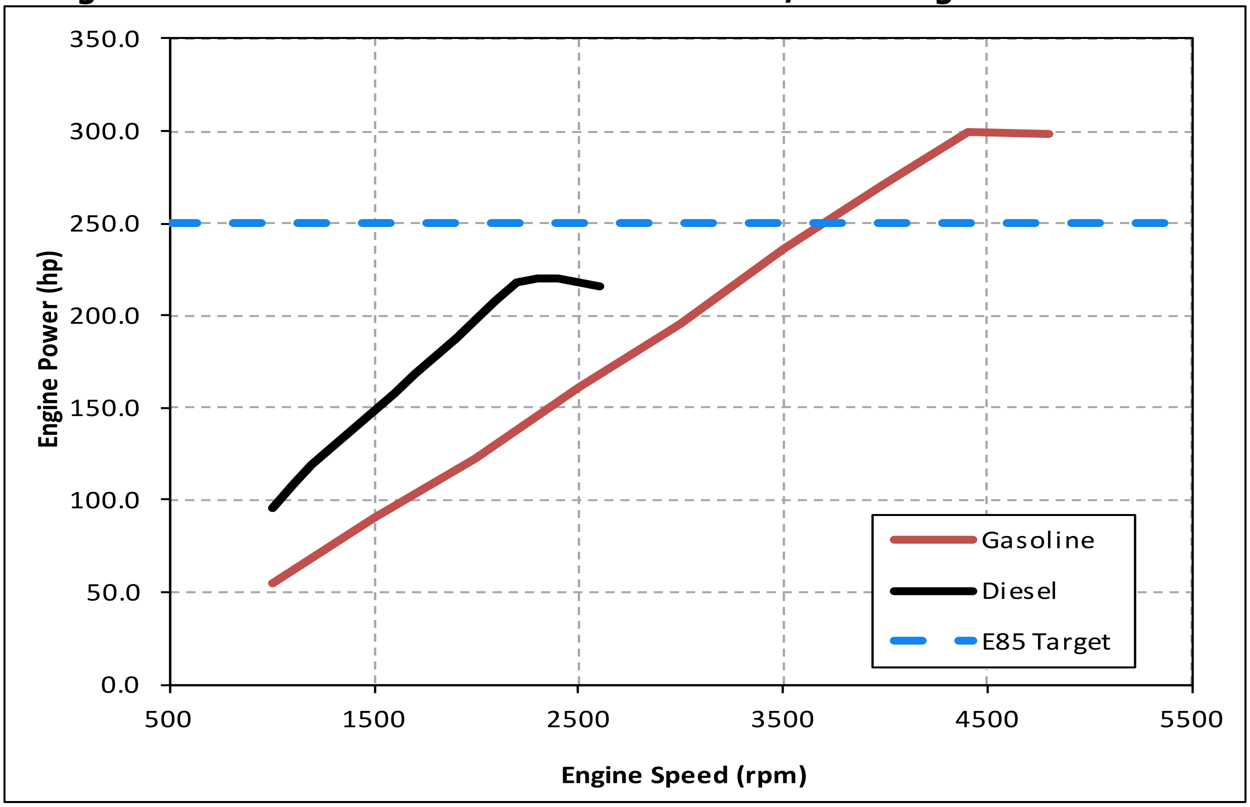
These vehicles are sold with both gasoline and diesel engine variants today. The torque and power curves that represent the most common diesel and gasoline options are shown in Figure 1 and Figure 2 below. As noted in the figures, there is a significant difference in the performance of these two engines although they serve the same market. This is primarily due to the fact that the engines used in step -van applications also serve other applications which may mean that they are not optimized for this particular application. Cummins collected voice of the customer data to determine what engine requirements would be optimal based on their needs. Through this process, we learned that customers would prefer an engine that has a peak torque of 440 pound-feet and a rated power of 250 horsepower, as shown by the target lines in Figure 1 and Figure 2.

Figure 1: Gasoline and Diesel Torque Curves w/ E85 Target Peak Torque



Source: Cummins

Figure 2: Gasoline and Diesel Power Curves w/ E85 Target Rated Power



Source: Cummins

These targets require a peak torque and rated power that are both between that of current diesel and gasoline options.

Optimal Engine Displacement to Meet the Performance Requirements

The goal is to select an engine displacement that will meet the peak torque and rated power requirements while remaining efficient at the operating conditions typically seen by the application. Medium-duty step-van applications have the potential to burn a significant fraction of fuel at high loads making it important to maintain efficiency up to and including the torque curve.

To enable the highest efficiency possible, it is typically desirable to select the smallest displacement possible. This concept is called 'down-sizing'. To enable 'down-sizing' for the optimized E85 engine, a turbocharger will be utilized to increase torque and power capability for a given engine displacement. For a turbocharged E85 fueled engine, there are two primary factors that limit engine efficiency along the torque curve. The first limiting factor is engine knock, and this is what limits maximum torque at low speeds (<2500 revolutions per minute (rpm)).

Engine knock is when fuel auto-ignites causing a rapid rise in cylinder pressure which is detrimental to engine durability. The goal is to design a combustion system that can run as efficient as possible without inducing engine knock at low speeds. There are three primary levers that that can be optimized to achieve this goal:

- Compression Ratio – Optimal range for an E85 optimized engine is 12:1 to 14:1
- Combustion phasing (spark ignition timing) – Desire to maintain optimal phasing (advanced)

- Intake boost – Increased boost increases torque but also increases the likelihood of knock.

For an E85 engine, the three levers can be optimized in multiple ways to create a scatter-band defining the optimal area where an E85 engine's torque curve can operate most efficiently at low speeds.

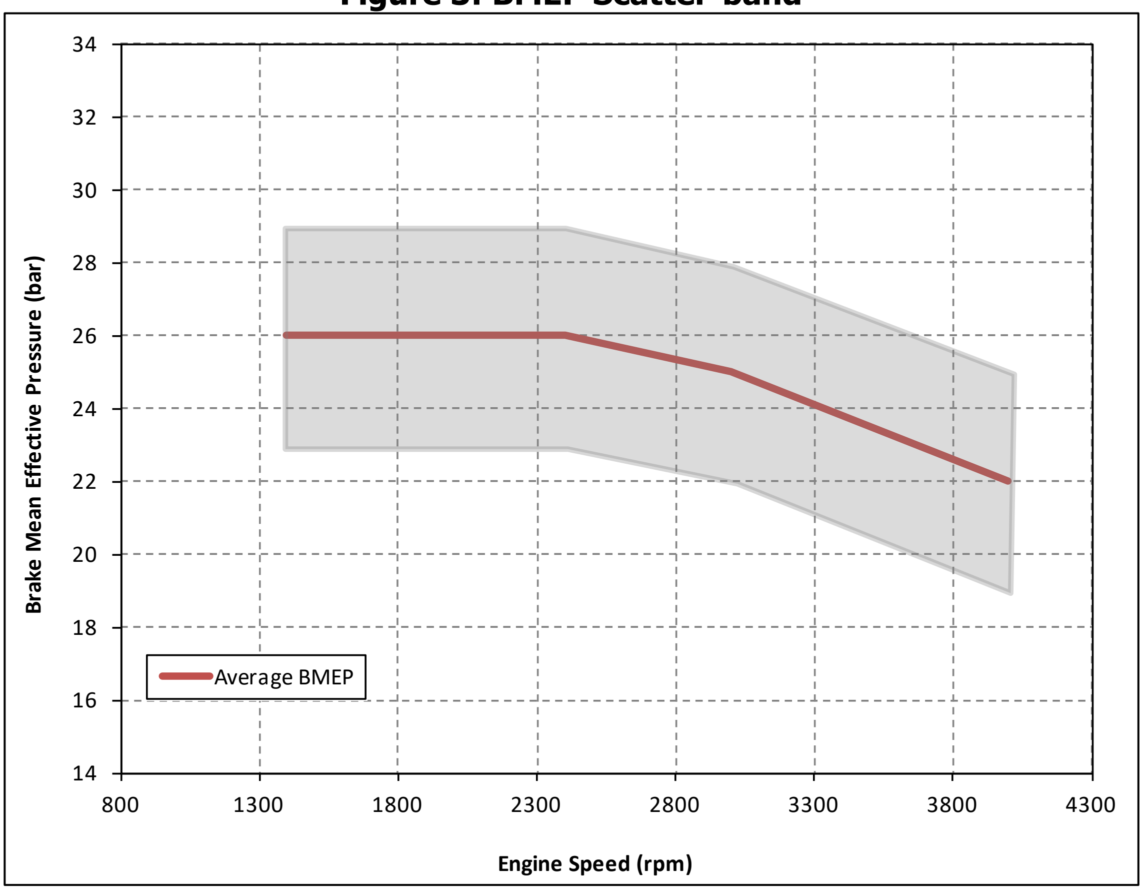
The second limiting factor is turbine inlet temperature. As power increases, more energy is going into the exhaust which raises turbine inlet temperatures. This becomes an issue at higher speeds (>2500 rpm). There are two primary levers that can be used to address turbine inlet temperature issues:

- Use fuel enrichment – This technique would result in a significant fuel consumption penalty and would make the exhaust after-treatment ineffective.
- Reduce engine power – This technique selects the maximum torque at a given speed achievable at the temperature limit without fuel enrichment.

For medium duty applications, fuel consumption and emissions along the torque curve are critical so the only viable option is to reduce engine power. Because we must determine the optimum displacement of the engine, we should consider a range of loads in terms of Brake Mean Effect Pressure (BMEP). BMEP is a measure of load (torque) that is normalized by engine displacement.

Figure 3 depicts the range of possible maximum BMEP as a function of engine speed for an E85 engine. The 'bend' in the BMEP scatter-band at higher speeds is due to power reduction to maintain turbine inlet temperatures within the design limit.

Figure 3: BMEP Scatter-band



Source: Cummins

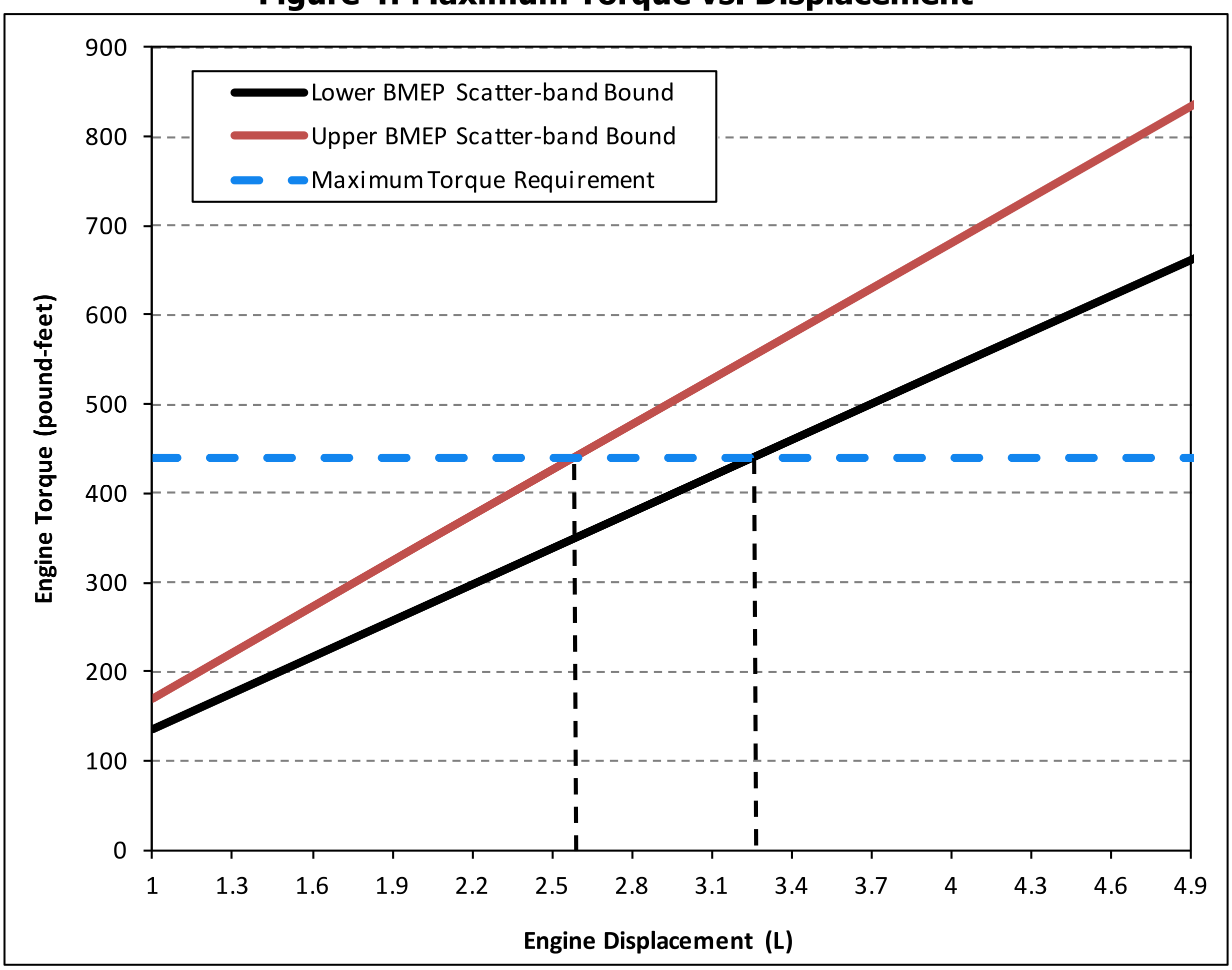
Because there are both peak torque and rated requirements, both can influence the engine displacement that is selected. First, the optimal displacement to achieve peak torque will be considered. Second, the optimal displacement determined for peak torque will be checked to determine if the rated power requirement can be meet with that same displacement.

Peak (Maximum) Torque

It is desirable to achieve the torque peak in the engine speed range of 1300 rpm to 1600 rpm for medium duty applications. This engine speed range is ideal because it minimizes engine friction and provides for improved vehicle drivability (via good transient response). Torque peak speeds of less than 1300 rpm (for further friction reduction) are typically not practical for two reasons. First, high torque at speeds less than 1300 rpm increases vehicle drive-train vibration to a level not acceptable to customers (drivers). Second, it is often cost prohibitive to match an air-handling solution (turbocharger, etc.) that can deliver the required airflow to achieve high torque at speeds less than 1300 rpm while still being capable of delivering rated power at higher speeds.

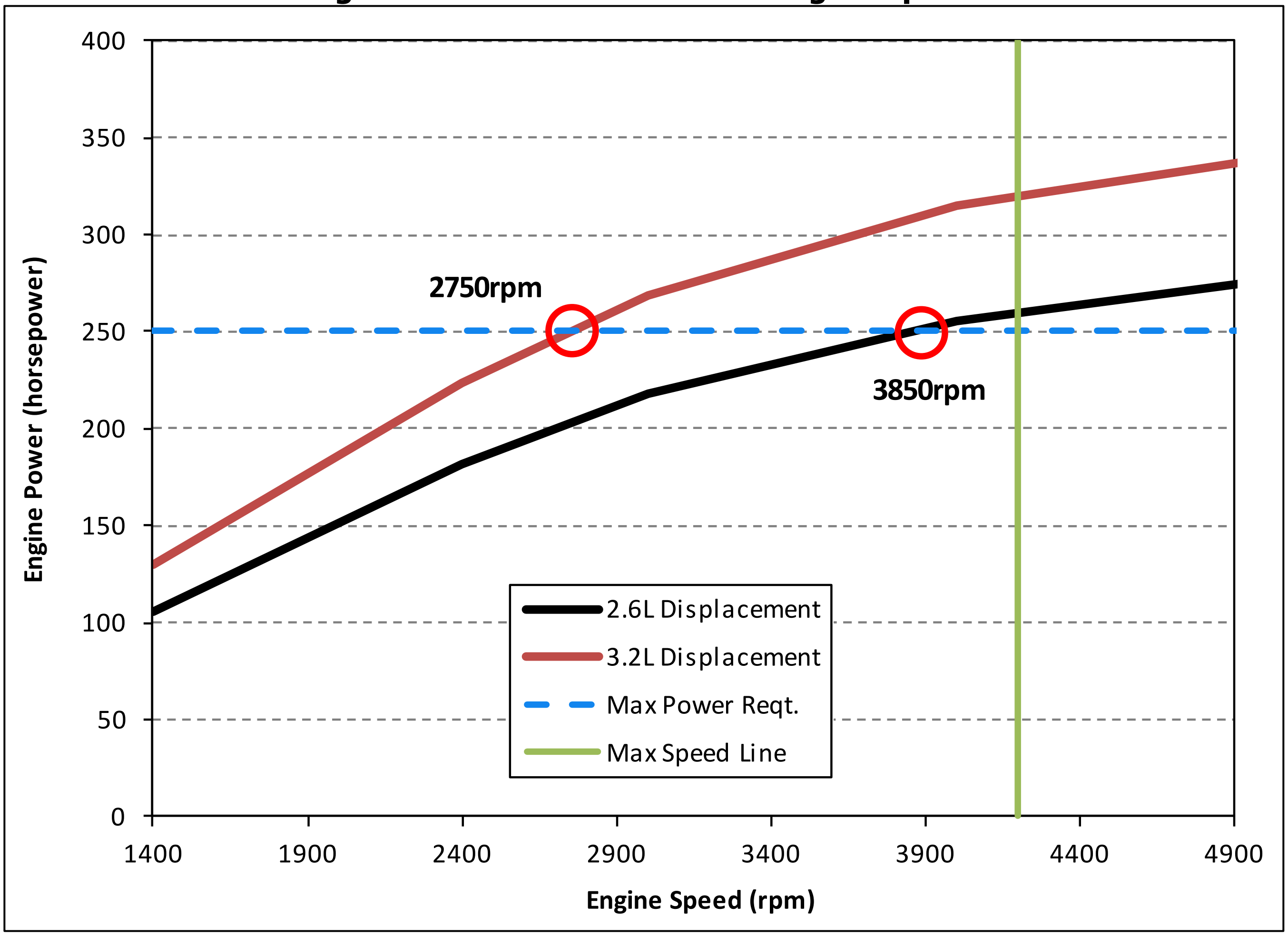
For this application, we are targeting a torque peak speed of 1400 rpm which provides a good compromise between friction reduction and air-handling hardware (which will be discussed in a later section). By combining the BMEP scatter-band bounds at 1400 rpm (from) with a range of displacements, a curve of displacement versus torque can be computed, as shown in Figure 4 below.

Figure 4: Maximum Torque vs. Displacement



Source: Cummins

Figure 5: Maximum Power vs. Engine Speed



Source: Cummins

This analysis shows that a displacement between 2.6 to 3.2 liters is optimal to meet the 440 pound-foot at 1400 rpm peak torque requirement while maintaining high engine efficiency.

Rated (Maximum) Power

With the optimal displacement for peak torque defined, the analysis should be turned to evaluating the rated power requirement. Using the average BMEP line in Figure 3, the power curves for the two extremes of engine displacement (2.6L and 3.2L) range can be calculated as shown in Figure 5.

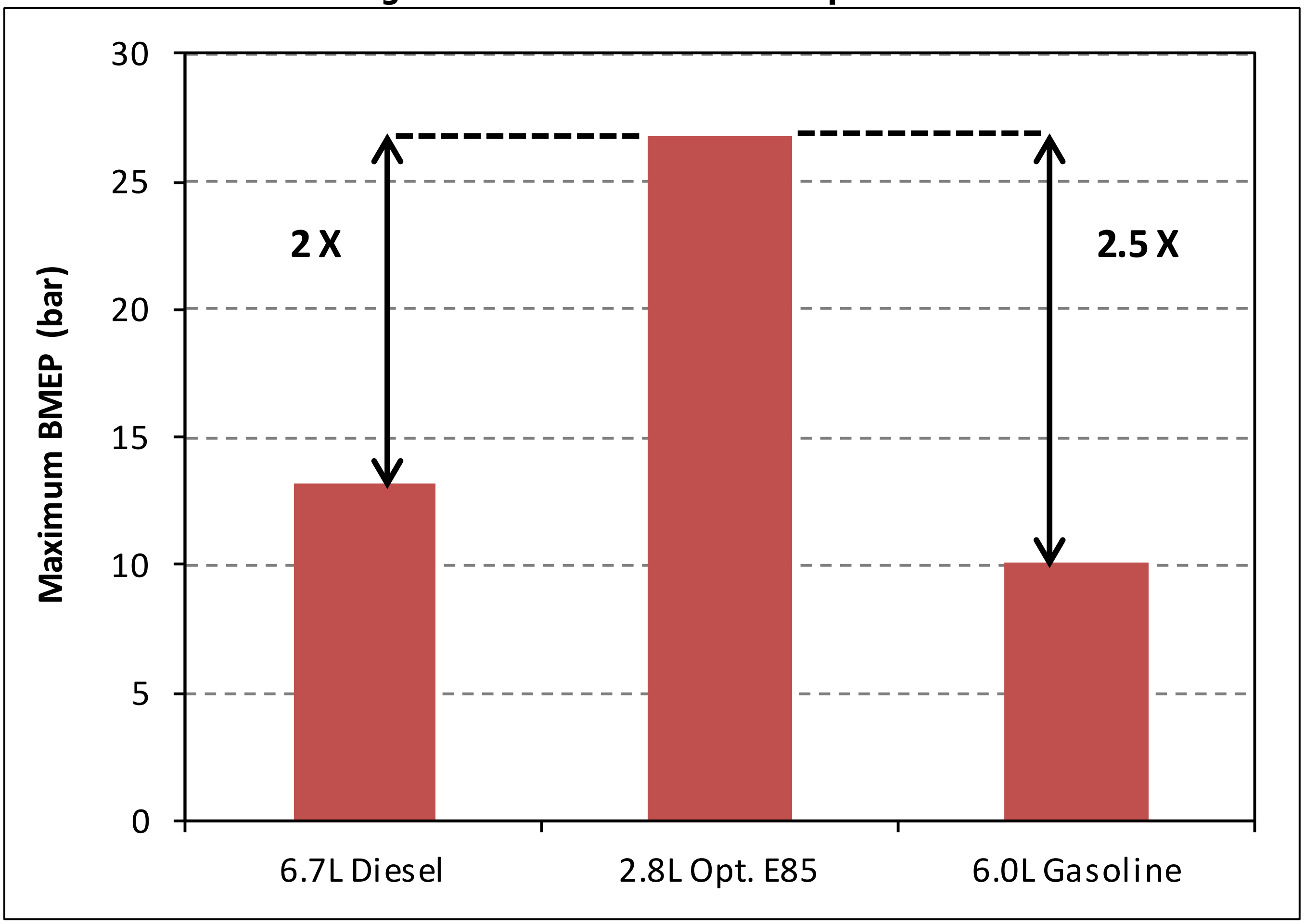
For this range of engine displacements, the rated power can be allowed to occur at speeds up to 4200rpm. This speed range limit is based on hardware limitations that occur when designing an engine capable of high torque at low engine speeds and high cylinder pressures. As can be seen in the figure above, both engines can meet the rated power target at a speed less than the maximum speed range allowed for the engine.

This analysis shows that both the maximum torque and power requirements can both be met by an engine with a range of displacements from 2.6 to 3.2L. For the optimized E85 engine, a displacement of 2.8L was selected to enable commonality with a clean sheet engine design that is currently active at Cummins as part of a US Department of Energy program. Because many of the design criteria for an optimized E85 engine match those of an optimal diesel engine, there are several advantages of basing the optimized E85 engine on a diesel platform. This base engine sharing will be discussed in more detail in the following two sections.

2.8L E85 Engine Comparison to Currently Available Engines

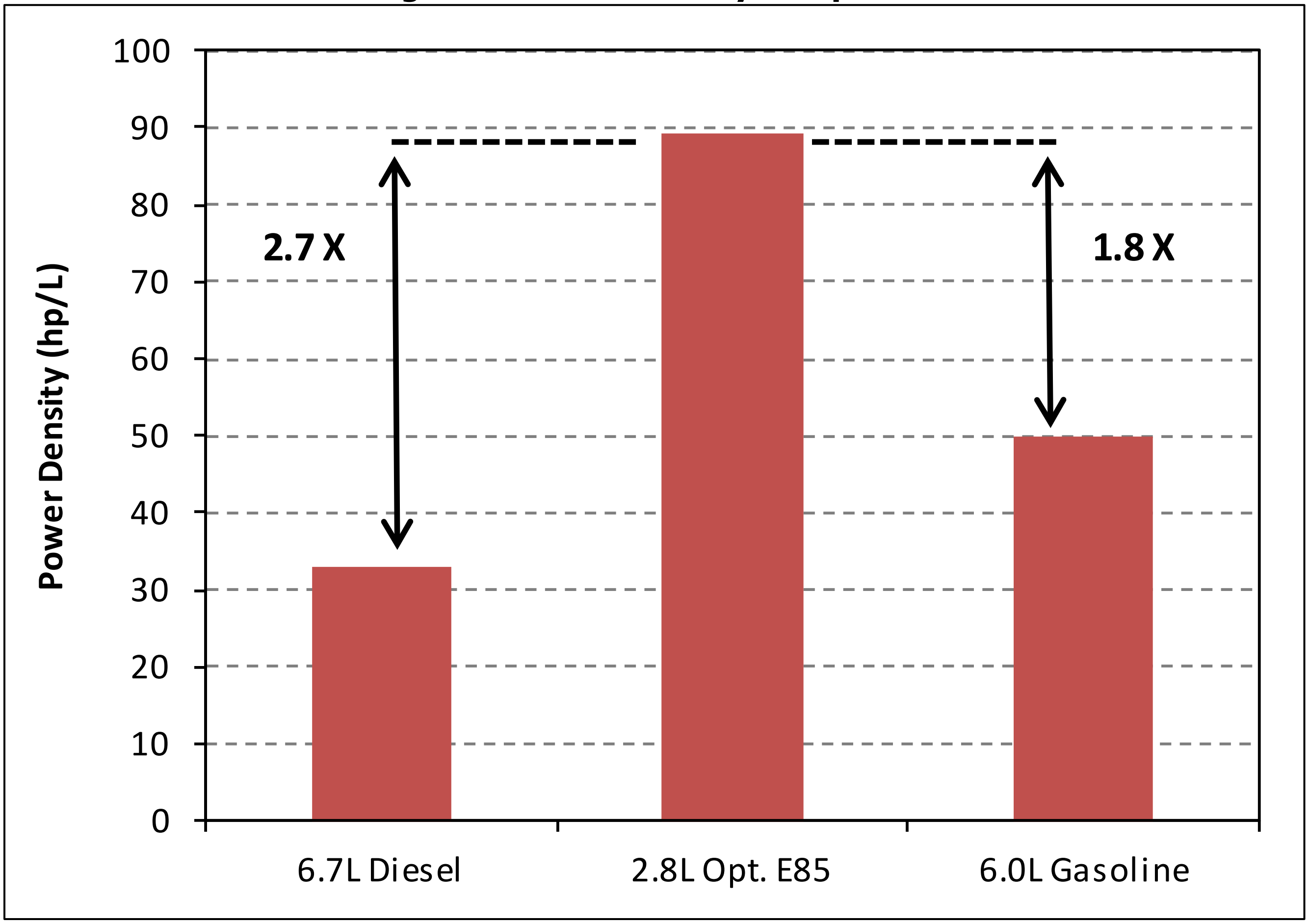
Figure 6 and Figure 7 compare the maximum BMEP (torque normalized by displacement) and power densities (power normalized by displacement) of the optimized E85 engine to the currently available diesel and gasoline options.

Figure 6: Maximum BMEP Comparison



Source: Cummins

Figure 7: Power Density Comparison



Source: Cummins

The 2.8L optimized E85 engine has a significantly higher maximum BMEP and power density when compared to the diesel and gasoline engine options. It is important to note that the power density increase compared to the gasoline engine could be higher. However, the E85 engine is targeting a lower maximum power based on customer requirements which results in a smaller power density increase. In summary, these results show that the E85 engine is significantly 'downsized' which enables the greatest reduction in greenhouse emissions possible while still meeting engine performance requirements.

Engine Architecture Similarities between a Diesel and Optimized E85 Engine
As we evaluate all of the similarities between a diesel and an optimized E85 engine with respect to medium duty applications two primary similarities stand out:

Maximum Cylinder Pressure Requirement - The cylinder pressure requirements for a diesel and E85 engine are similar in the range of 175-200 bar. This pressure range is significantly higher than state-of-the-art gasoline engines that have a pressure capability of 90-130 bar. Without the increased pressure capability, it would be challenging to realize the efficiency potential possible with E85 fuel.

Durability - Because both the diesel and optimized E85 engine are designed for the medium-duty market, they are designed to meet the durability requirements. Gasoline engines that are used in the medium-duty market are typically not designed for the same durability. With the gasoline engines used in the medium-duty market today, it is common for one or two engine re-powers to be required before the end of the vehicle's useful life.

For these reasons, it is advantageous to use an architecture that is more similar to a state-of-the-art diesel engine than a modern gasoline engine as the base for an optimized E85 engine. This why this program is partnering with U.S. DOE sponsored 2.8L clean sheet diesel engine program at Cummins for base engine design. With an engine displacement of 2.8L, Table 1 shows the torque peak and rated power targets.

Table 1: 2.8L Optimized E85 Engine Torque Peak and Rated Power Targets

	Speed (rpm)	BMEP (bar)	Torque (lb-ft)	Horsepower (hp)
Torque Peak	1400	26.8	440	126
Rated Power	3200	24.9	410	250

Source: Cummins

The speed at which rated power is achieved was selected based on selecting a BMEP in the middle of the scatter-band and selecting the speed at which this BMEP would achieve the maximum power target.

Additional Synergies Associated with using the Diesel Platform
As discussed earlier, the optimized E85 engine will be based on a 2.8L diesel engine. This engine is a clean sheet design being developed as part of a US Department of Energy demonstration program. Many of the key technologies necessary to meet the engine performance requirements to be met are common between the diesel and optimized E85 engine variants:

Cast Aluminum Cylinder Head – For the diesel variant, aluminum is used for weight reduction. For the E85 variant, there is an additional advantage to using cast aluminum in place of cast iron. Aluminum improves heat transfer in the combustion face surface which reduces the propensity for knock. While increased heat transfer by itself reduces efficiency, the optimization of compression ratio and combustion timing enabled by the reduced likelihood of knock outweigh the heat transfer losses resulting in a net gain in efficiency.

Cast Aluminum Block with Steel Cylinder Liner Inserts – The aluminum block construction significantly reduces weight while the steel cylinder liner inserts maintain the durability requirements of the medium duty market.

Cylinder Pressure Capability – The common crank-train and power cylinder design is capable of supporting peak cylinder pressures up to 200bar.

Dual Overhead Cam Valve-Train – Will be utilized to implement variable valve timing independently on both the intake and exhaust cams for the optimized E85 engine. This allows for improved torque via increased cylinder burned exhaust scavenging at low speeds and minimized pumping losses through-out the entire operating map. Can also be used during transient operation to improve torque response.

Weight Optimized Design – The engine is designed to be a light-weight platform through the use of lightweight materials (such as the aluminum block) and optimal design of components. The optimized E85 engine will be approximately 700 pounds lighter than the 6.7L diesel engine option which contributes to this program's overall reduction in vehicle greenhouse gas emissions for the target duty cycle.

While many of the requirements of the diesel and E85 engine are common, there are requirements that are not. By developing this new engine with both diesel and E85 combustion requirements being considered at the same time, it is possible to protect the common elements of the design for features that need to be different for each variant to achieve optimal performance and efficiency.

High Pressure E85 Direct Injection Fuel System

This engine will utilize direct injection of E85 fuel into the cylinder. By injecting the fuel directly into the cylinder, full advantage of the high charge cooling capability of ethanol fuel can be realized. The charge cooling of E85 is significantly greater than gasoline due to the higher latent heat of vaporization. This charge cooling reduces in cylinder temperatures which in turn mitigates engine knock. This charge cooling effect is required to achieving the BMEP scatter-band shown in Figure 3. The specifications for the fuel system are:

Common Rail Injection System with 200 bar maximum injection pressure.

A High Flow Capacity high pressure fuel pump.

Multi-Hole Design Direct Injectors with a maximum flow rate of 19.0 cc/sec.: The spray pattern of the injector was optimized to meet two criteria. First, to provide good air-fuel mixing in the cylinder to improve combustion stability and emissions. Second, to minimize fuel impingement on the cylinder walls to minimize fuel in oil dilution and the potential for cylinder liner wear issues.

Spark Ignition System - A spark ignition system with a single spark plug per cylinder utilizing a coil-on-plug ignition coil is being integrated into the engine. The spark plugs are a 12mm iridium design with a heat range and gap selected for E85 combustion. Leading technology ignition coils with maximum ignition energy of 100 megajoule (MJ) have been selected.

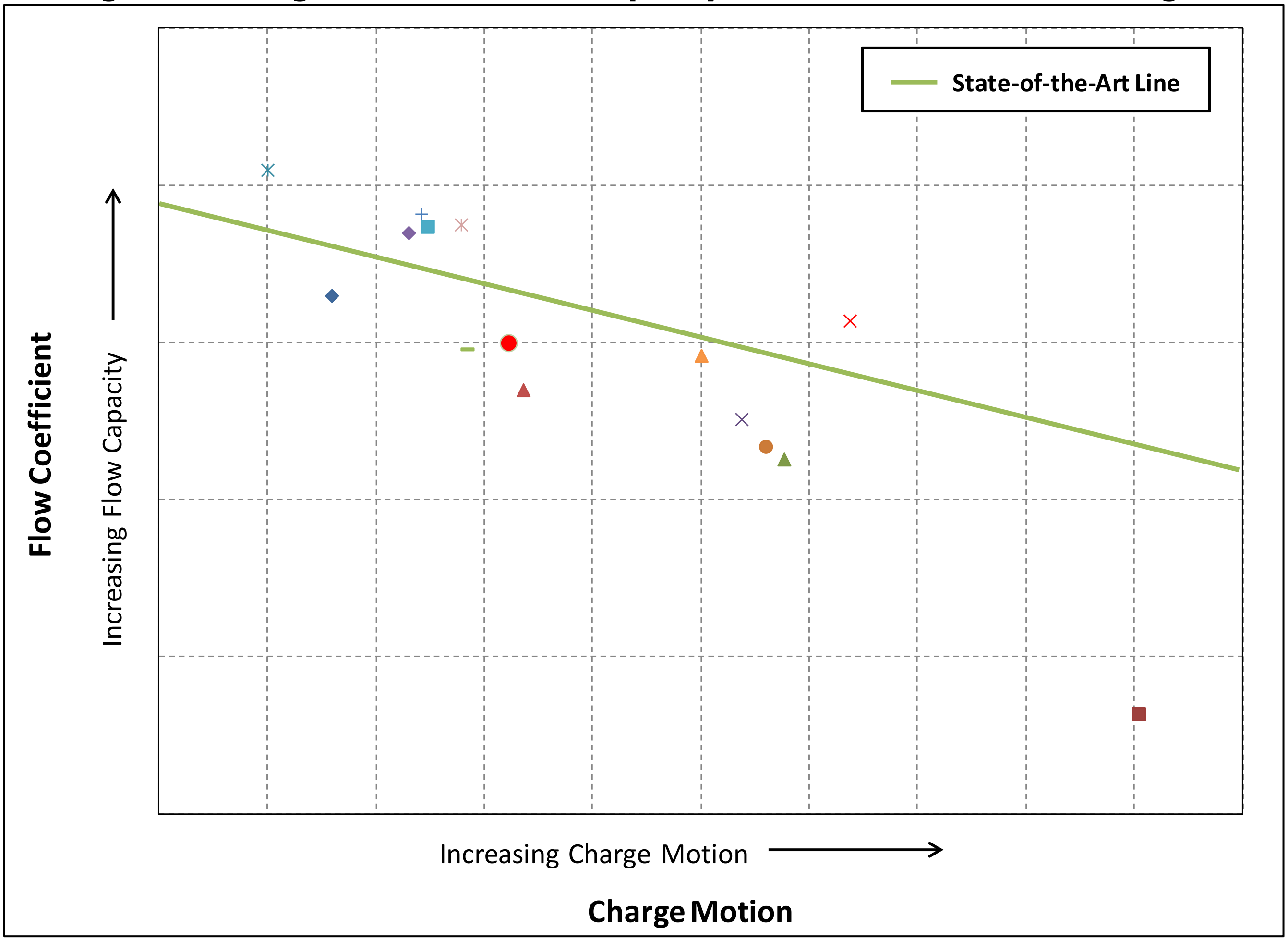
Compression Ratio - The compression ratio of the initial E85 engine is being set at 12:1 with the flexibility to specify any value within the range of 10:1 to 14:1 (if found to be needed during engine testing). The compression ratio of the E85 engine is lower than the diesel variant for two reasons. First, the lower compression ratio enables higher maximum BMEP levels to be achieved (higher than the diesel variant) while staying with-in cylinder pressure limits. Second, the lower compression ratio also allows high BMEP operation at optimal combustion phasing without knock occurring. While a higher compression ratio alone would result in better engine efficiency, the higher compression ratio would require non-optimal combustion phasing resulting in a net reduction in efficiency.

Cylinder Head - The cylinder head for this engine is a unique design and optimized for E85 combustion. There are three primary design features that are unique to this cylinder head when compared to the diesel variant:

Intake Port Design – The intake ports will be designed to create the in-cylinder charge motion, created during the intake process, which is optimal for E85 combustion. The optimal motion target is based on previous Cummins experience and industry literature reviews.

Having established the target charge motion, the target discharge coefficient is determined using the state-of-the-art charge motion vs. discharge coefficient line, shown in Figure 8. The target integrated discharge coefficient of 0.35 is specified.

Figure 8: Charge Motion vs. Flow Capacity for Various Intake Port Designs



Source: Cummins

Fuel Injector Mounting – the E85 direct injector is located with the nozzle located in a pocket within the combustion phase surface between the two intake ports (i.e. side or laterally mounted). This location was selected to meet injector cooling requirements within component packaging constraints.

Center-Mounted Spark Plug – The cylinder head is being designed to accommodate a center mounted spark plug. A 12mm plug is specified and the water-cooling jacket is optimized to provide proper cooling to mitigate spark plug "hot-spots" that can lead to engine knock.

Piston - The piston for the E85 optimized engine will also be constructed of aluminum for improved heat transfer for the same reasons as the cylinder head. The piston bowl geometry is optimized to achieve the 12:1 compression ratio target while maintaining the desired in-cylinder charge motion during the compression process.

Stoichiometric Combustion with 3-Way Catalyst - The combustion system of this engine is designed for stoichiometric operation over the entire engine operating map, including the torque curve. The ARB's 2010 heavy-duty emissions requirements will be met using three-way catalyst (TWC) technology. The after-treatment will consist of both a close-coupled unit mounted on the turbocharger outlet and a chassis mounted unit under the vehicle body. This arrangement provides the best compromise between system efficiency and packaging constraints. Cummins is working with a catalyst supplier to optimize the catalyst wash-coat for this application.

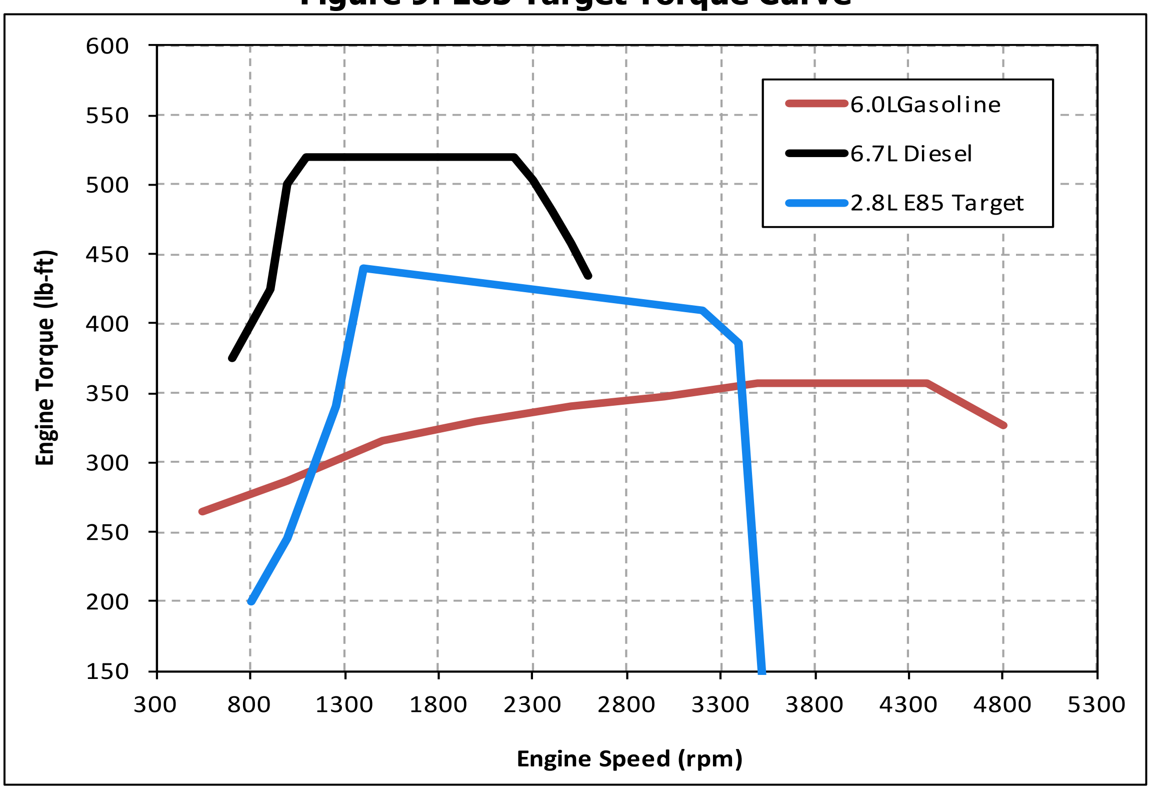
While it is true that increased efficiency may be possible by implementing a dual mode combustion system (lean at light loads, stoichiometric at high loads), the project decided not to pursue this path due to the increased level of catalyst technology development required to meet the 2010 ARB emissions targets. However, since dual mode combustion application represents additional improvements in CO2 emissions, Cummins considers this an excellent candidate for a future follow on program.

Target Torque Curve – Optimized E85 Engine

Having established the turbocharger match and valve lift profiles, the GT-Power model was used to determine the maximum torque that can be achieved at speeds less than 1500rpm. Figure 9 shows the results of this analysis.

It is worth noting that the torque of the E85 engine at speeds less than 1100rpm is significantly less than both the gasoline and diesel engines currently used in this application. For this application, single turbo was selected instead of a more complicated two-turbocharger or turbocharger-supercharger configuration to minimize engine cost. With a single turbo, compromises are required at low engine speeds limiting the available boost which in turn limits the maximum torque possible. With limited boost, torque becomes a function of engine displacement and with the E85 engine having less than half the displacement of the other two engines it will always be at a disadvantage.

Figure 9: E85 Target Torque Curve



Source: Cummins

However, the capability to deliver high torque at speeds less than 1100 rpm is only an advantage in applications that use manual transmissions to provide increased clutch engagement torque for easier vehicle launches (i.e. with more torque you don't have to be as 'gentle' with the clutch pedal to launch the vehicle). For medium duty step-van applications, automatic transmissions are the majority of the market share. The majority (well over 70 percent) of the vehicles sold into the medium duty step-van market are equipped with automatic transmissions. Because of this, the E85 engine demonstration vehicle will also be equipped with an automatic transmission. With an automatic transmission, low speed torque is not required because the torque converter curve does not allow the engine to operate at these conditions even if the engine is capable of it, as shown in Figure 9.

Components Optimized Specifically for The E85 Engine

High Pressure E85 Direct Injection Fuel System - The engine is equipped with a direct injection system capable of 200 bar injection pressure coupled with multi-hole injectors with a maximum flow rate of 17.0 cc/sec. The spray pattern of the injector is optimized to provide homogeneous mixing of the charge, minimize liner wall wetting, and to eliminate any interference with open intake valves. By injecting the fuel directly into the cylinder, full advantage of the high charge cooling capability of ethanol fuel is realized. The charge cooling of E85 is significantly greater than gasoline due to the higher latent heat of vaporization and the increased fuel mass required to deliver an equivalent amount of energy. This charge cooling reduces in-cylinder temperatures which in turn mitigates engine knock.

Spark Ignition System - The ignition includes a single spark plug per cylinder utilizing a coil-on-plug configuration. The spark plugs are a 12mm iridium design with a heat range and gap selected for low temperature E85 combustion. A cold heat range spark plug was selected for two reasons. Since E85 does not produce as much soot and smoke, spark plug fouling was not an issue running colder heat range plugs. Also, E85 is more susceptible to pre-ignition caused by hot plug temperatures. Using a colder plug helps mitigate this path of pre-ignition on the engine. The spark plug gap was preset by the manufacturer. The cylinder head is designed to accommodate a center mounted spark plug. A "full wrap" water cooling jacket is optimized to provide proper cooling to mitigate spark plug hotspots that can lead to knock or ethanol pre-ignition.

Engine Design

Table 2 shows engine specifications.

Table 2: Ultra-Low Carbon Powertrain Project Engine Specifications

Main Dimension	Unit	Value
Engine dimensions		
Displacement	L	2.8
Bore	mm	94
Stroke	mm	100
Bore Spacing	mm	104
Performance		
Peak (Maximum) Torque	lb-ft @ rpm	450 @ 1400
Rated (Maximum) Power	hp @ rpm	250 @ 3400
Turbocharger		
Configuration	--	Single entry turbine
Valve train		
Configuration (type)	--	Type I, dual overhead camshaft
Variable Valve Actuation	--	Independent intake and exhaust cam timing adjustment with switching dual intake / exhaust cam profiles
Intake Valve		
Number / Cylinder	--	2
Head Diameter	mm	36
Lift Duration	degrees	160 / 240
Maximum Lift	mm	4 / 9
Phase Adjustment	degrees	70
Exhaust Valve		
Number / Cylinder	--	2
Head Diameter	mm	32
Lift Duration	degrees	200 / 250
Maximum Lift	mm	5.5 / 9.5
Phase adjustment	degrees	55
Fuel System		
Type	--	Multi-hole direct injection
Maximum Pressure	bar	200
Injector Maximum Flow Rate	cc/sec	17.0
Fuel Compatibility	--	E85
Ignition System		
Type	--	Production coil-on-plug
Maximum Energy	mJ	100
Emissions After-Treatment		
Type	--	3-way emissions catalyst
Configuration	--	Close-coupled, on engine
Design Considerations		
Peak Cylinder Pressure	bar	174
Compression Ratio	--	12:1
Cylinder Head	--	Cast aluminum
Cylinder Block	--	Cast aluminum with cast iron liners
Piston	--	Cast aluminum with gallery cooling

Source: Cummins

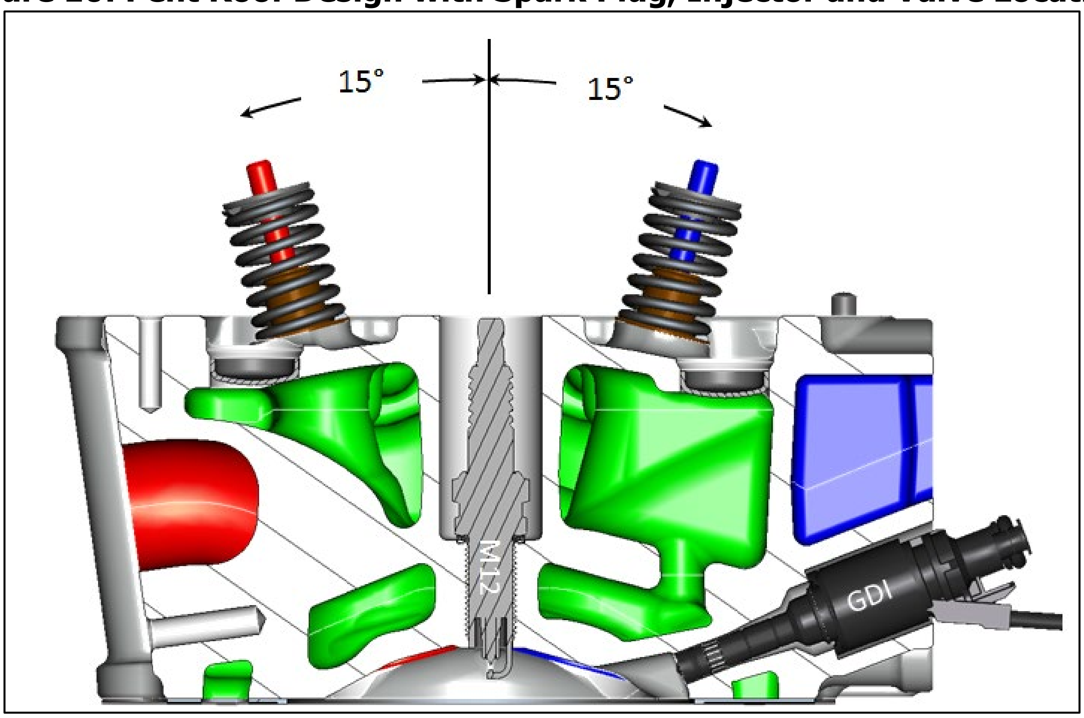
Compression Ratio - Compression ratio of 12:1 is chosen based on 1-D modeling and simulation results. 12:1 compression ratio coupled with high boosting achieves 27 bar BMEP at

1400 RPM while keeping spark advance close to maximum brake torque timing and within peak cylinder pressure limits.

A higher compression ratio alone would result in improved engine efficiency, but the higher compression ratio would require non-optimal combustion phasing which could result in a net reduction in efficiency. For a downsized medium duty application which will spend a significant amount of time on the torque curve, best efficiency is achieved when a lower than maximum compression ratio is used such that torque curve combustion phasing is still near maximum brake torque timing.

Cylinder Head - Air enters each cylinder through twin intake high tumble ports. The target integrated discharge coefficient of 0.35 is used based on competitive benchmarking. The combustion chamber uses a pent roof design with the intake and exhaust valves inclined 15° from the center to achieve reasonable flow capacity while still keeping cylinder head volume low such that a flat piston top could be used at 12:1 compression ratio. This allows for a good surface area to volume ratio tradeoff even at slightly higher or lower compression ratios. Figure 10 illustrates a cross-section of the cylinder head along bore centerline, perpendicular to the crankshaft axis.

Figure 10: Pent Roof Design with Spark Plug, Injector and Valve Location



Source: Cummins

Fuel Injector Mounting – The E85 direct injector is located between and below the two intake ports (i.e. side or laterally mounted). This location was selected to meet injector cooling requirements within component packaging constraints. A recess is provided in the combustion chamber surface at the injector nozzle tip for adequate clearance to the fuel spray pattern.

Variable Valve Timing and Actuation – A Type I (i.e. direct acting bucket tappet) valve train is used in conjunction with dual overhead camshafts. This enables a state of the art variable valve actuation system capable of independent valve phasing for intake and exhaust valves. The bucket tappets are also configured to enable the use of two different valve lift profiles for the intake and exhaust valves. Electronically actuated hydraulics are used to control the various valve train functions using engine oil as the working fluid.

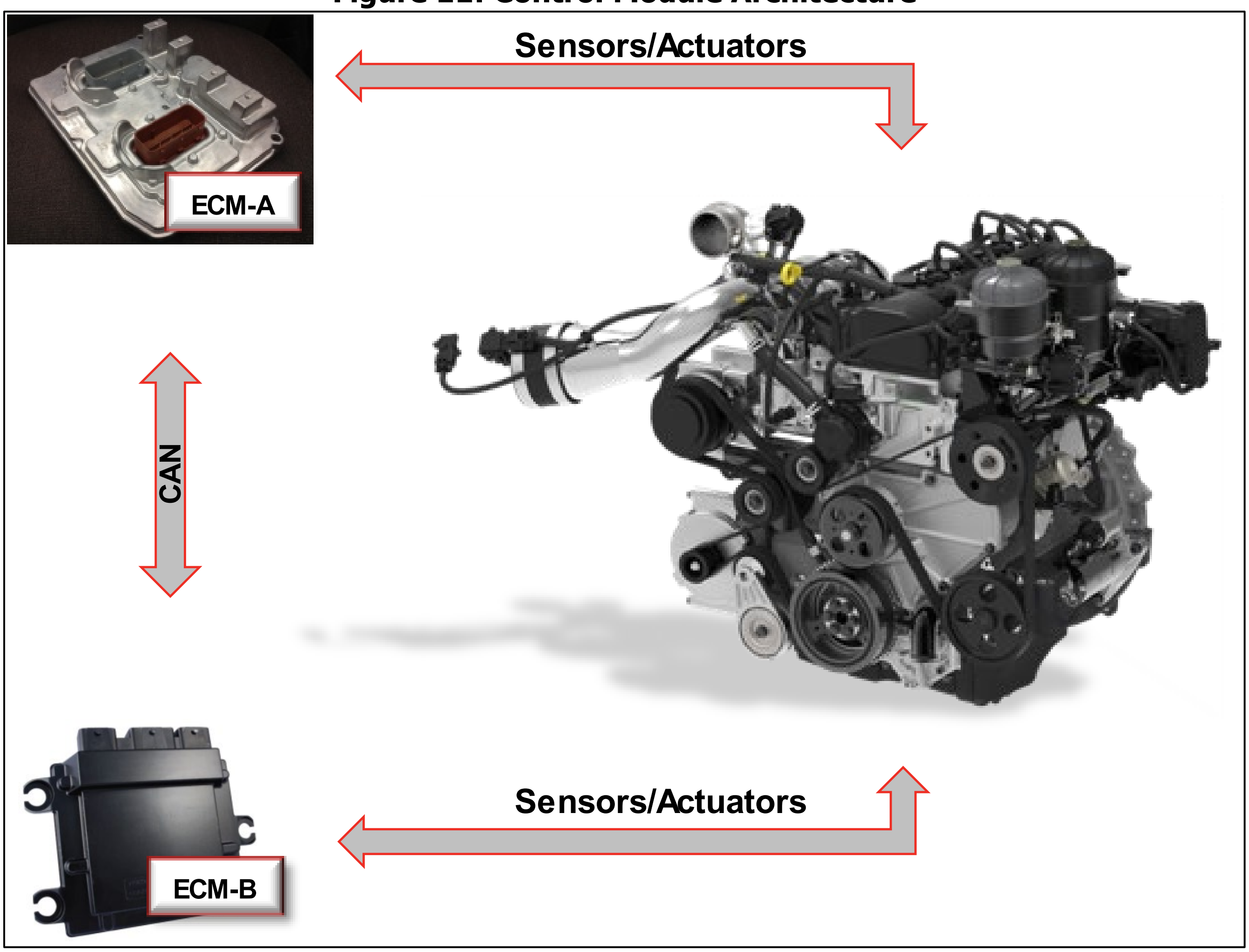
Piston - Gallery cooled cast aluminum pistons are used for maximum knock mitigation at high loads and high cylinder pressures. A flat top piston with valve pocketing for all 4 valves to allow for nearly 6mm of valve opening at top dead center is used which enables high valve overlap operation with minimal surface area increase. A steel top ring carrier is employed to allow for continued operation at high cylinder pressure with minimal top ring groove wear.

Stoichiometric Combustion with 3-Way Catalyst - This combustion system is designed for stoichiometric operation over the entire engine operating map, including the torque curve. The exhaust catalyst system consists of a close-coupled unit, engine-mounted on the turbocharger outlet.

Engine Controls

The engine control hardware consists of two Electronic Control Modules (ECM) and a suite of sensors and actuators which communicate with the control units, which are shown in Figure 11. The majority of the control functions reside in ECM-A, while ECM-B provides expanded I/O required for some of the additional functions unique for this engine such as variable valve timing. The modules communicate bi-directionally via a CAN link.

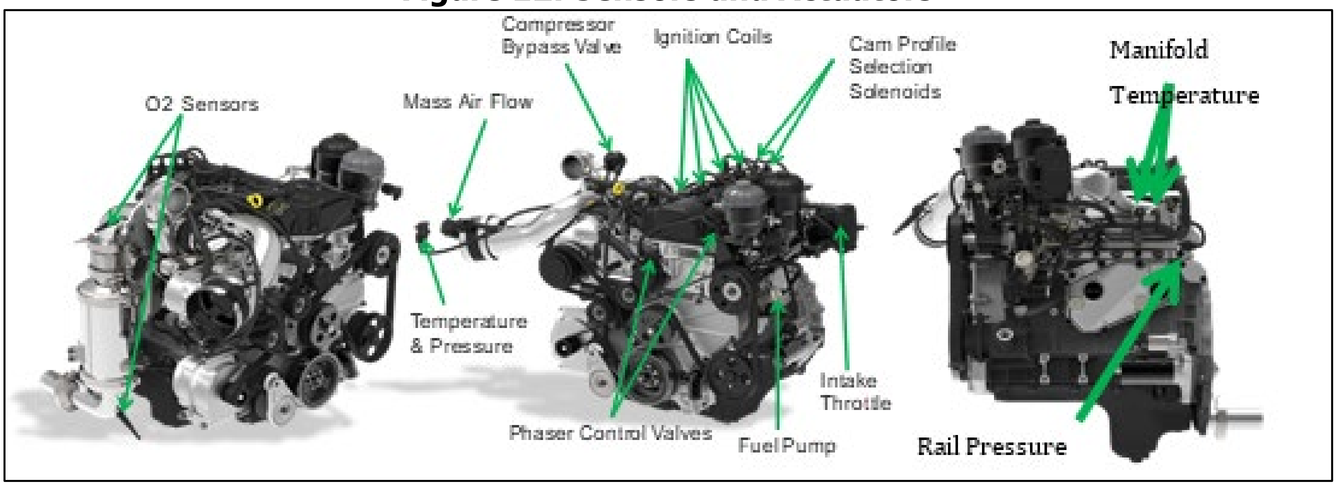
Figure 11: Control Module Architecture



Source: Cummins

Numerous sensors and actuators are used for controlling the engine, identified in Figure 12. An air flow sensor provides a direct measure of air at the entrance to the air box. Various pressure and temperature sensors are used to estimate air flow at points along the air flow pathway and are particularly useful in transient conditions such as when the engine is accelerating. These sensors provide feedback to the control module for controlling air flow and fuel quantity for proper combustion. The primary means of controlling air flow into the engine is the intake throttle which is electronically actuated via motor control. A secondary airflow control means is through the mechanical waste gate on the turbocharger. The waste gate is not electronically controlled but actuates mechanically as a function of boost pressure. The waste gate opening sequence was chosen to achieve the desired torque curve and transient response for this project.

Figure 12: Sensors and Actuators



Source: Cummins

The variable valve system on the engine consists of dual, independent cam phasers with two shuttle valves to regulate hydraulic pressure to the phaser actuators. Hall-effect position sensors (not shown) located in proximity to the cams are used to measure the angle of the cams relative to the crank shaft angle. The engine is also equipped with two discrete cam profiles (both intake and exhaust) which are toggled using a hydraulic circuit regulated by the four solenoids on top of the engine. This level of cam shaft control over the engine breathing characteristics allows for reduced CO_2 emissions relative to the lower cost fixed cam timing strategy.

The fuel system consists of four direct, in-cylinder fuel injectors (not shown), and a high-pressure pump which delivers fuel to a high-pressure accumulator or "rail". A pressure sensor mounted on the rail provides feedback to the control module which is used to regulate to the desired pressure using an electronically controlled valve on the pump. Fuel pressure is controlled to achieve optimum fuel droplet size and air/fuel charge homogeneity while minimizing the power consumption of the fuel delivery system.

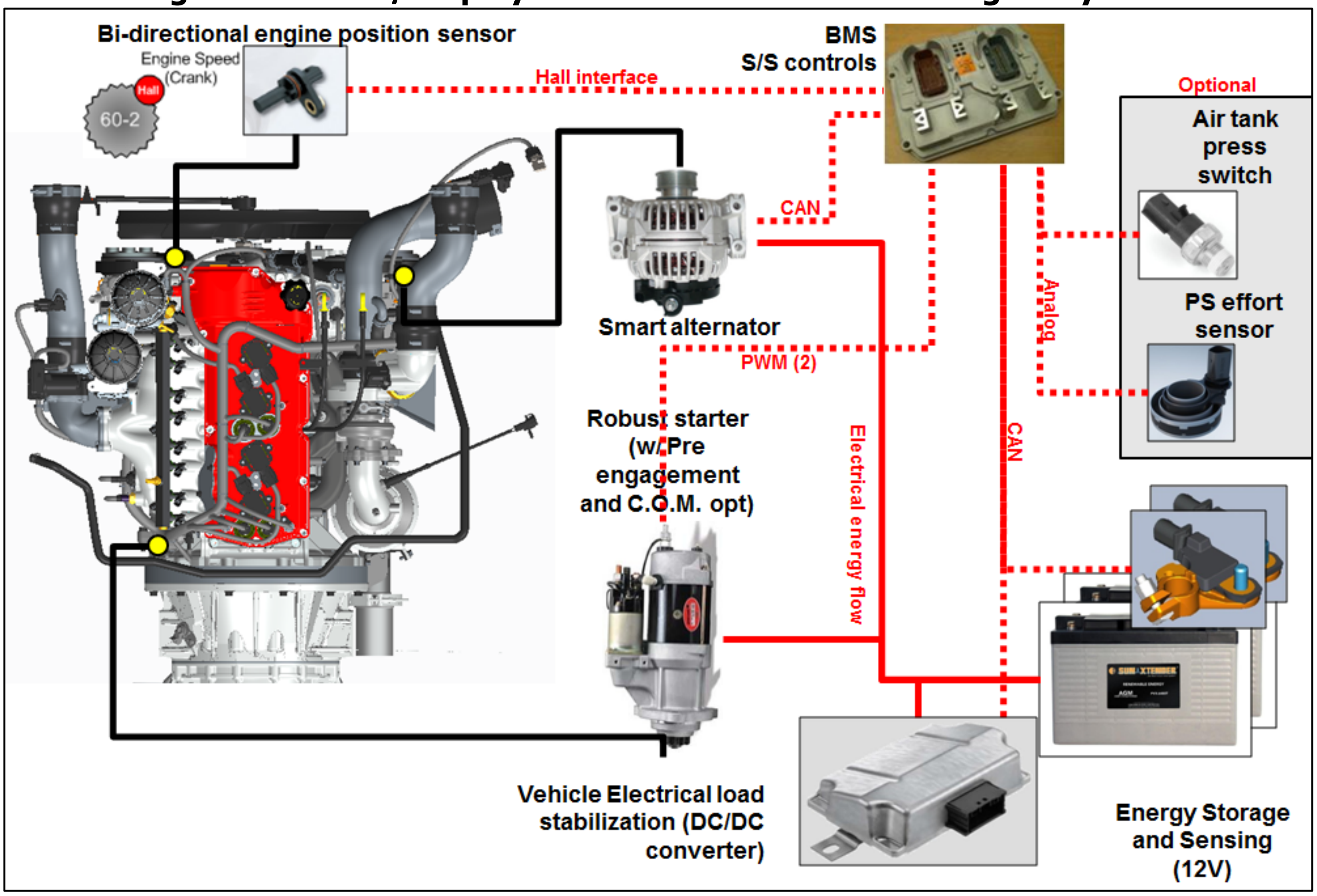
The ignition system consists of four coil-on-plug units which are independently controlled by the control modules by commanding variable dwell times synchronous with engine rotation to ignite the respective cylinder fuel/air mixture at the desired time. A high energy ignition system with a breakdown voltage limit of 40 kilovolt was chosen due to the increased cylinder pressure at which this high compression engine will operate. This allows for improved ignitability of the dense air/fuel charge providing smoother engine operation.

The exhaust system is equipped with two O2 sensors, which provides feedback to the control modules for maintaining the desired air-to-fuel ratio for optimal combustion and high conversion efficiency through the catalyst.

Stop/Start System Controls

The start/stop architecture for this program includes proven technology integrated with proprietary controls to offer a cost competitive solution for fuel consumption and carbon emissions reduction. This system provides excellent value with a payback period acceptable to customers using vehicles in classes 4-6 to allow for good market acceptance of this system. A schematic of the Start/Stop system architecture is found in Figure 13.

Figure 13: Start/Stop System Architecture for E85 Engine System



Source: Cummins

Ideally, a stop/start system allows the engine to shut down at every idle event. In order to accomplish this goal, the batteries must be maintained at a specific level of charge such that vehicle electrical loads can be supplied while the engine is shut down and there is enough battery charge to crank the engine for a restart at the end of the shutdown event. The role of the alternator in this system is to maintain the state of battery charge such that the maximum amount of engine off operation can be achieved and the vehicle electrical accessories can always be powered.

A conventional alternator achieves the function of keeping the batteries charged by always maintaining a voltage output slightly higher than the battery voltage. The conventional alternator has no concern for engine operation; it only provides some level of charge to the batteries at all time. The smart alternator being used for this project provides an intelligent charging strategy. The smart alternator is a small cost increase above a conventional alternator but allows the engine computer to control the output voltage of the alternator to increase or decrease the battery charge rate. The alternator integrated into this system can be varied from a 10V output (no charging of the batteries) to a 16V output (more rapid charging of the batteries). By applying software control algorithms to the alternator charging strategy, increased fuel economy can be achieved.

Two prime mechanisms of intelligent alternator control have been applied to this project. The first is to manage the battery/alternator system so as to allow regenerative energy capture during deceleration events such as braking and/or coasting. This allows the system to recapture some of the vehicle kinetic energy normally going to the vehicle brakes during the deceleration and applying the energy to the batteries. The second application of intelligent control is to manage the battery/alternator system load during engine propulsion mode.

The engine shutdown logic is used to shut down the engine during as many idling opportunities as possible when the batteries are charged enough to allow for a shutdown. The engine will restart when the driver releases the brake pedal to begin accelerating the vehicle or when the batteries become discharged to a level that requires a restart. If the batteries are excessively discharged, the life of the batteries may become degraded or a restart of the engine may be impossible. Additional shut down logic is included to prevent the engine from shutting down if the driver is performing a maneuver such as creeping up to a stop sign in a line of cars such that the inconvenience of a rapid series of shut down and start up events is not encountered.

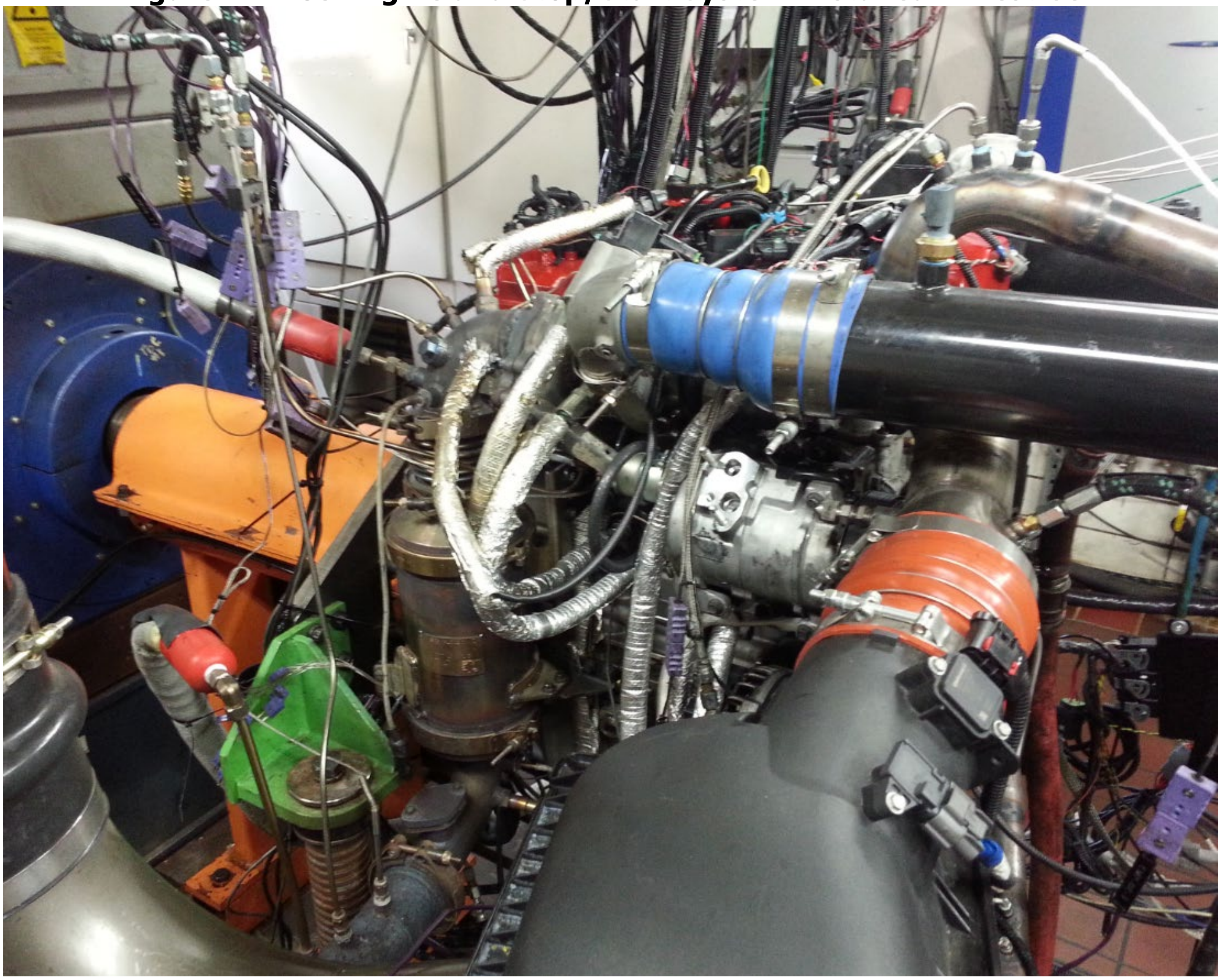
Test Cell Testing

A test cell engine was built up according to the design specifications presented earlier and installed in the test cell to start performance development. A rapid prototype control system was used initially to control the engine and test out the control algorithms. The control was ultimately turned over to the same ECM configuration presented earlier and also planned to be used in the vehicle. The test cell engine can be seen in Figure 14.

Three (3) sizes of single-entry turbochargers were specified for this configuration to allow for determining of the best match to the engine hardware. Certain difficult-to-model parameters affect turbocharger sizing such as evaporation rate for ethanol in this type of combustion system. Since the fuel is injected through a direct injector into the cylinder during the intake stroke, the ease with which air enters the cylinder is also affected. This effect interacts with the turbocharger system by changing the intake manifold pressure required for a particular speed and load condition. The three turbochargers were tested on the engine configuration to obtain the best turbocharger match to the requirements of the target torque curve. This turbocharger was used to perform fuel mapping and optimization exercises which demonstrated it is a good match across the speed and load range of interest.

Due to the modeling uncertainty around the turbocharger matching, turbochargers made from readily available materials were specified for this matching work to minimize the cost spent on hardware which would not ultimately be used for the final engine configuration. Upon completion of the turbo matching exercise, the appropriate high temperature capable turbocharger was ordered to allow for stoichiometric engine operation at high power.

Figure 14: E85 Engine and Stop/Start System Installed in Test Cell

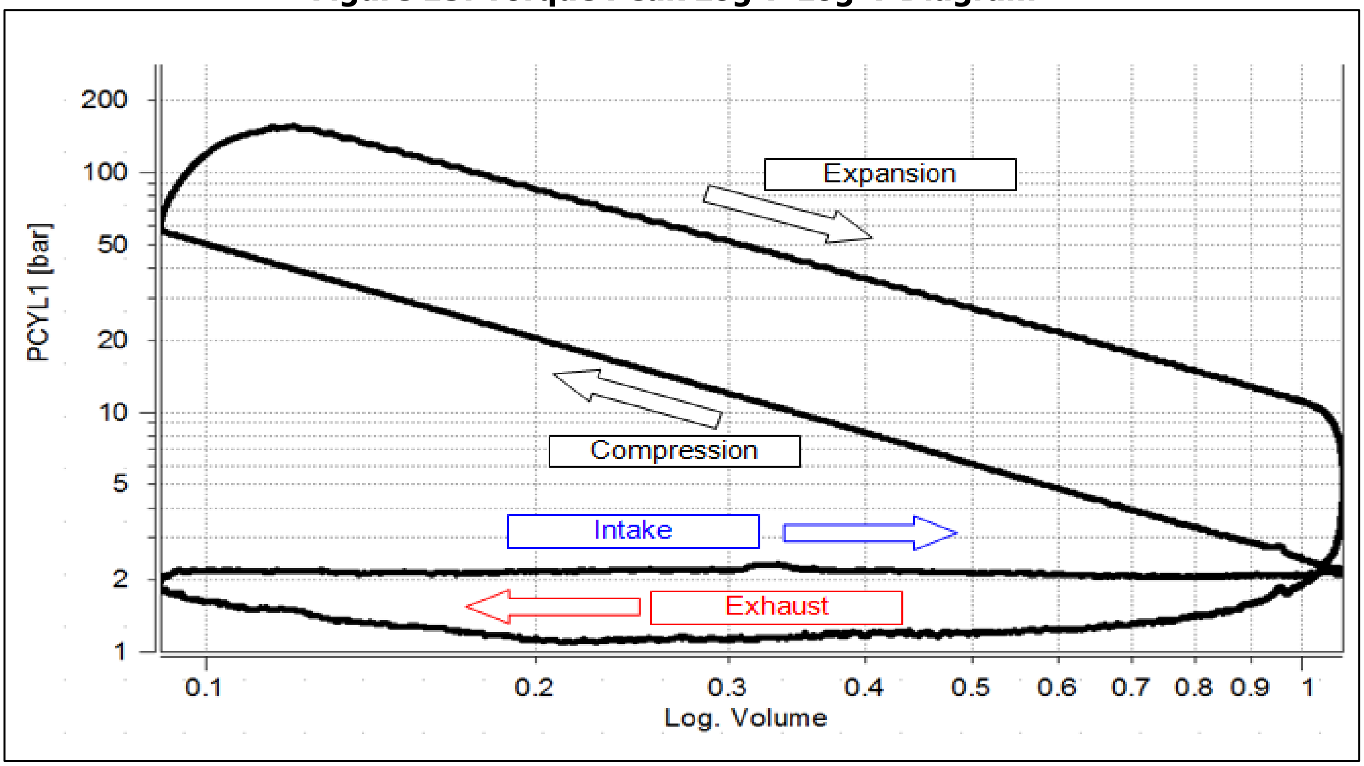


Source: Cummins

A high energy ignition system was incorporated into this engine configuration due to the challenge of igniting a very dense air-fuel charge at high cylinder pressure due to the 12:1 compression ratio. This ignition system choice has proved to be very reliable on the test engine and gives good performance results. One challenge encountered during this testing phase is the very low pre-ignition temperature of ethanol fuels. Typical spark plugs for gasoline applications are designed to operate with a tip temperature around 800 degrees Celsius to avoid carbon fouling on the electrodes and resulting misfire events. Ethanol has a pre-ignition temperature threshold, which is much below 800 degrees Celsius. The pre-ignition phenomenon where the air-fuel charge is ignited before the spark event is particularly prominent at the lower engine speeds and high loads of this high specific power platform. Cummins has worked with an ignition system supplier to specify spark plugs which minimize the occurrence of pre-ignition at these conditions to allow for attainment of the target torque curve.

A significant lever for producing good engine efficiency on this project has been the integration of an efficient turbocharger with a cylinder head that exhibits excellent gas flow characteristics. Using a turbocharger capable of higher temperatures than a diesel turbocharger typical to this medium duty application allows for good exhaust energy recovery at low turbine inlet pressure. In this situation with high turbocharger inlet temperature and low required boost pressure due to the high volumetric efficiency of the combustion system design, some of the waste heat can be recovered from the exhaust and used to push the piston down during the intake stroke. This is contrary to many turbocharged or naturally aspirated engine designs which require the piston to pull air into the cylinder. Figure 15 shows an example cylinder pressure trace taken near torque peak speed and moderate load, which shows a very good pumping loop where turbine inlet pressure is much lower than intake manifold pressure. This higher intake manifold pressure pushing the piston down during the intake stroke improves engine efficiency at high load.

Figure 15: Torque Peak Log-P Log-V Diagram



Source: Cummins

Mule Vehicle Development

In order to meet the delivery timeline for this project, it was not possible to wait on a functioning E85 engine to begin component testing and controls development for the stop/start system. The project team decided to use a production Freightliner M2 vehicle owned by Cummins and powered by a Cummins ISB 6.7L diesel engine. Since the stop/start system for the E85 engine was planned to be controlled by the Cummins ECM, the ISB engine was a good fit already being controlled by a similar ECM. This allowed nearly all the controls algorithms and software developed for stop/start on the diesel engine to be carried over directly and implemented in the ECM for the E85 engine.

Figure 16: Stop/Start Mule Development Truck

Source: Cummins

The mule truck showing in Figure 16 used identical stop/start components except for the engine starter, which had to be larger than the E85 engine to accommodate the starting requirements of the diesel engine. A standard production starter for this engine was used for stop/start development purposes. The mule vehicle was used to develop the communication strategy and controls for the starter, smart alternator, battery sensors, and transmission interface. This vehicle allowed for stop/start development to be completed and ready to implement on the E85 engine when the E85 demonstration vehicle was first assembled.

Vehicle Integration

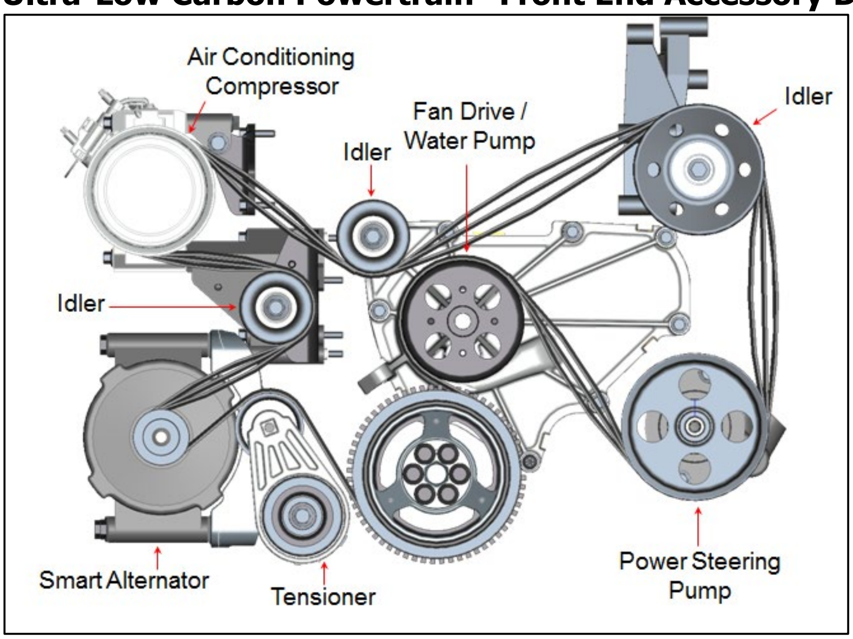
Under Task 5.1 of the Statement of Work, Cummins completed the vehicle selection, powertrain layout in the vehicle, and the design work required for integration of the powertrain into the vehicle. The goal of this project is to demonstrate this "Ultra-low Carbon Powertrain" in a medium duty step van type vehicle which is commonly used in pickup and delivery type applications. Cummins worked with Freightliner Custom Chassis Corporation, who is a well-known chassis supplier for this type of vehicle, to procure a vehicle suitable for this project development and demonstration. Freightliner Custom Chassis Corporation has provided an MT45 chassis with a Morgan Olson body which can be seen in Figure 17. The MT45 chassis can be loaded to 19,500 lbs. to allow development work at the average vehicle weight for classes 4-6.

Figure 17: Demonstration Vehicle

Source: Morgan Olson

Freightliner Custom Chassis Corporation was also willing to provide models of the frame rails, cab area extending into the engine bay, and work with Cummins to understand integration requirements for the various vehicle subsystems. These integration requirements were used to size several of the Front-End Accessory Drive components which are mounted on the engine but used to supply various vehicle subsystem needs. The sizing of these components affects the belt routing and design as well as the space claim for the front end of the engine. The Front-End Accessory Drive components include the air conditioning compressor, alternator, power steering pump, water pump, and fan drive. The completed Front-End Accessory Drive layout can be seen in Figure 18.

Figure 18: "Ultra-Low Carbon Powertrain" Front End Accessory Drive Layout



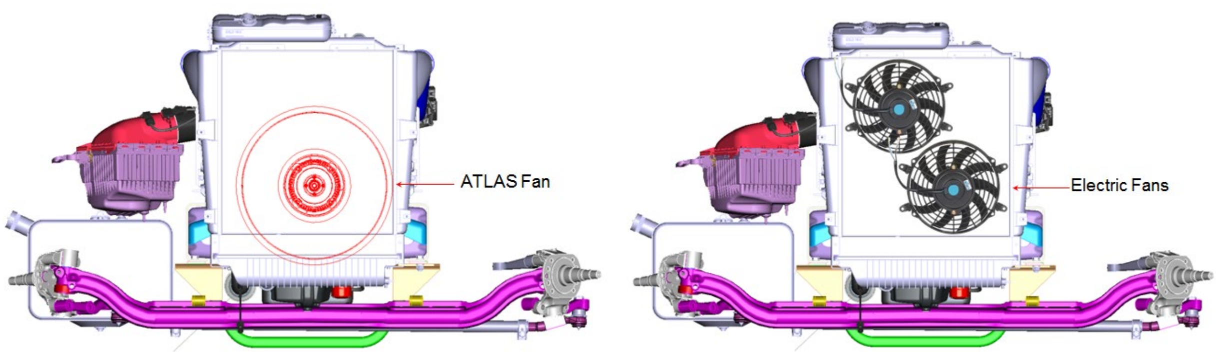
Source: Cummins

The Front-End Accessory Drive layout in Figure 18 appears to have three belts in the graphic. There is only one belt used to drive the Front-End Accessory Drive, but this view of multiple belts is used to show the expected belt deflection in operation due to the torsional vibration of the crankshaft. This view is used to ensure the belt does not cause wear on any unintended components when it moves outside the nominal belt path.

The agreed budget for this project was minimized by using common components with the U.S. DOE funded light-duty diesel project, ATLAS, when possible. This alignment is favorable due to the cylinder pressure requirements of this E85 engine lining up very well with the diesel engine cylinder pressure requirements, thereby reducing the development expense for many engine components. However, one place where the commonality U.S. DOE and CEC-funded programs has added a challenge is in the fan hub location. The current fan hub location was chosen because it centers the fan in the ATLAS demonstration vehicle. When the same fan hub location is placed into this program's demonstration vehicle, the fan does not line up optimally with the vehicle cooling package as shown in the left picture of Figure 19.

It is not easy to change the fan hub location on the engine because it is integrated into the water pump location which is integral to the block water passages. The method chosen to address this issue was to use electric cooling fans on the demonstration vehicle as can be seen in the right picture of Figure 19. This allows additional flexibility in the battery management strategy for the start/stop system integration.

Figure 19: Vehicle Cooling Package Front View



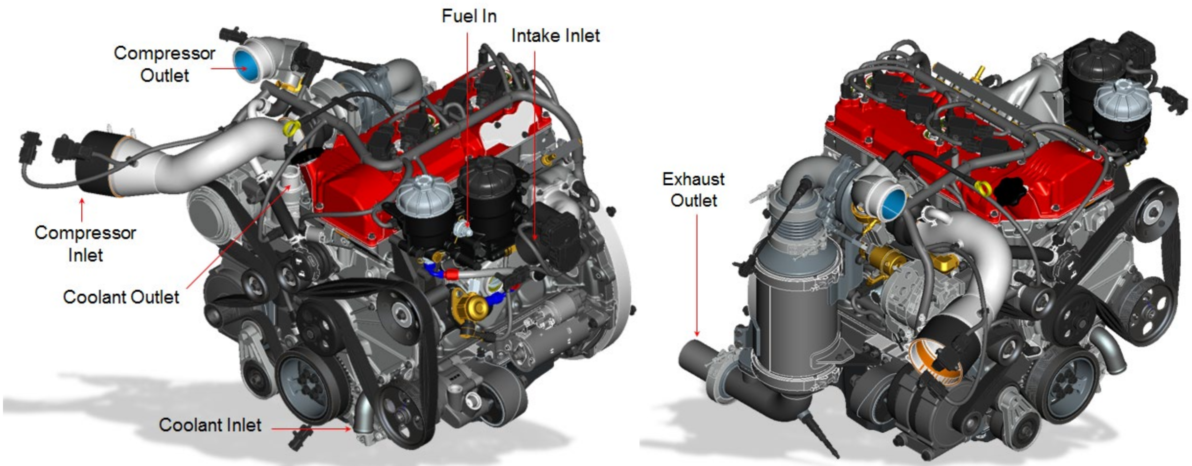
Source: Cummins

In addition to changing the fan location on the engine and fan type in the vehicle, the power steering pump needed design consideration for this application. The block was designed with mounting of the ATLAS light-duty vehicle power steering pump, prior to this program's vehicle selection. This Class 4-6 demonstration vehicle requires a much larger power steering pump because the vehicle weight is much higher than the ATLAS vehicle, and the power steering pump will additionally be used to supply brake assist pressure for the braking system. Modification of the mounting bracket to fit the larger power steering pump and relocation to a lower position on the engine was required for this application.

There are four primary "fluid" connections for running the engine which must be connected to and supplied from the vehicle. These are the air, coolant, fuel and exhaust systems. The connections for these three systems were designed to integrate with this demonstration vehicle and can be seen in Figure 20, which depicts the completed engine. The left picture in

Figure 20 also shows the oil and fuel filter locations which have been designed for easy access in the vehicle by locating them on top of the engine.

Figure 20: Isometric Right and Left Side Views of the Engine

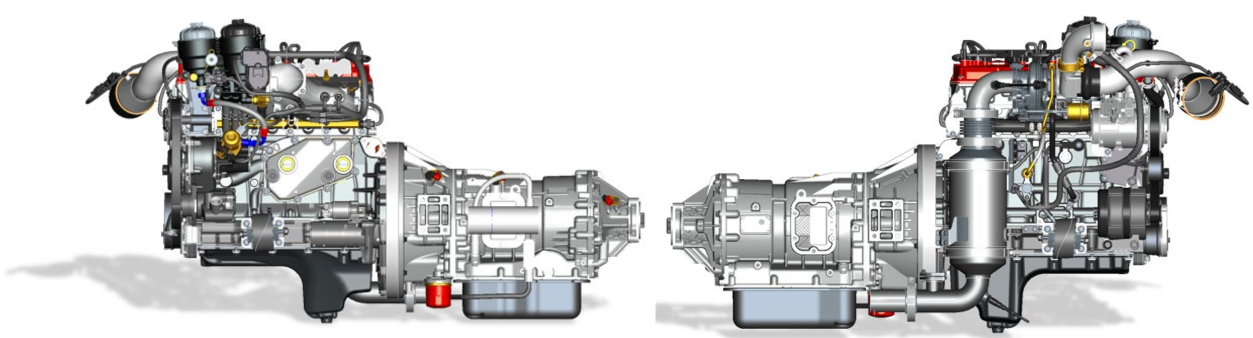


Source: Cummins

Transmission integration to the engine required design of an adapter plate to mate the engine integrated flywheel housing with the bell housing of the transmission. The engine block design incorporates a flywheel housing cast into the back end of the block. This flywheel housing was designed to integrate with the transmission used in the ATLAS vehicle, and requiring only a small spacer, roughly one inch in length, to join the engine to transmission for the heavier class vehicle.

This integration can be seen in Figure 21. Somewhat visible in Figure 21 is the significant amount of downsizing represented by this E85 engine. The engine is only marginally larger than the transmission which could not be downsized due to the high torque and power curves provided by this engine design.

Figure 21: Right and Left Side Views of the Powertrain Package

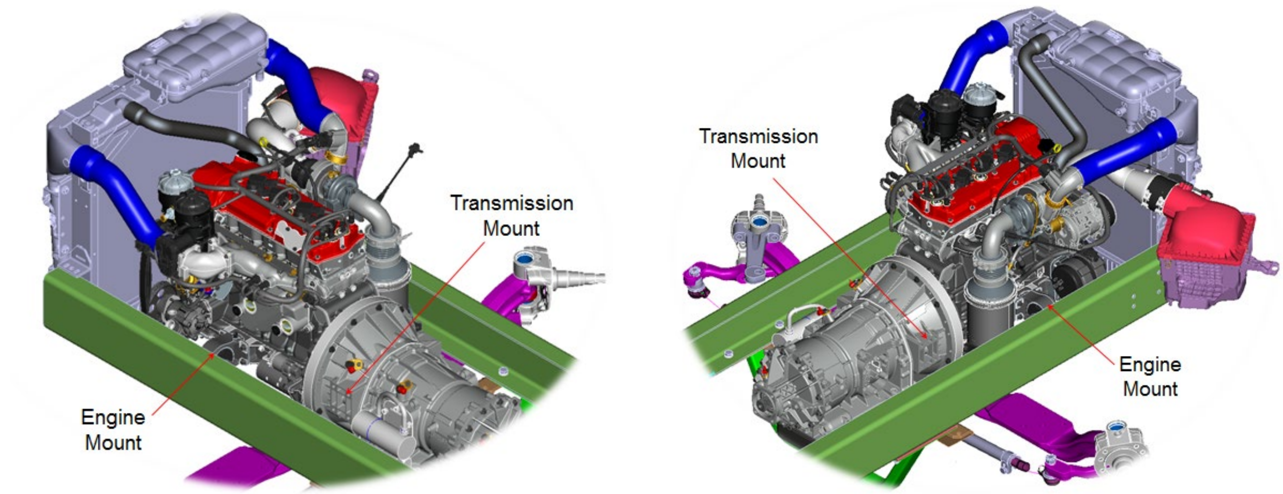


Source: Cummins

With the fluid connection locations defined on the engine and the transmission fitted to the engine, the powertrain could now be located in the frame rails to start making connections to these rails, to the cooling package, and to the other various systems. These various systems and connections can be seen in and Figure 23.

Custom hoses were designed for all the fluid connections as shown in Figure 22. Engine side mounts from the ATLAS vehicle can be used but frame side engine mounts were required to be designed from scratch for this application. The transmission mounts and driveshaft length also had to be designed specifically for this application. The fuel lines from the baseline gasoline vehicle will be used with minor modifications. The fuel tank, low pressure fuel pump, and evaporative emissions system remains located over the rear axle of the vehicle as it is in the baseline configuration.

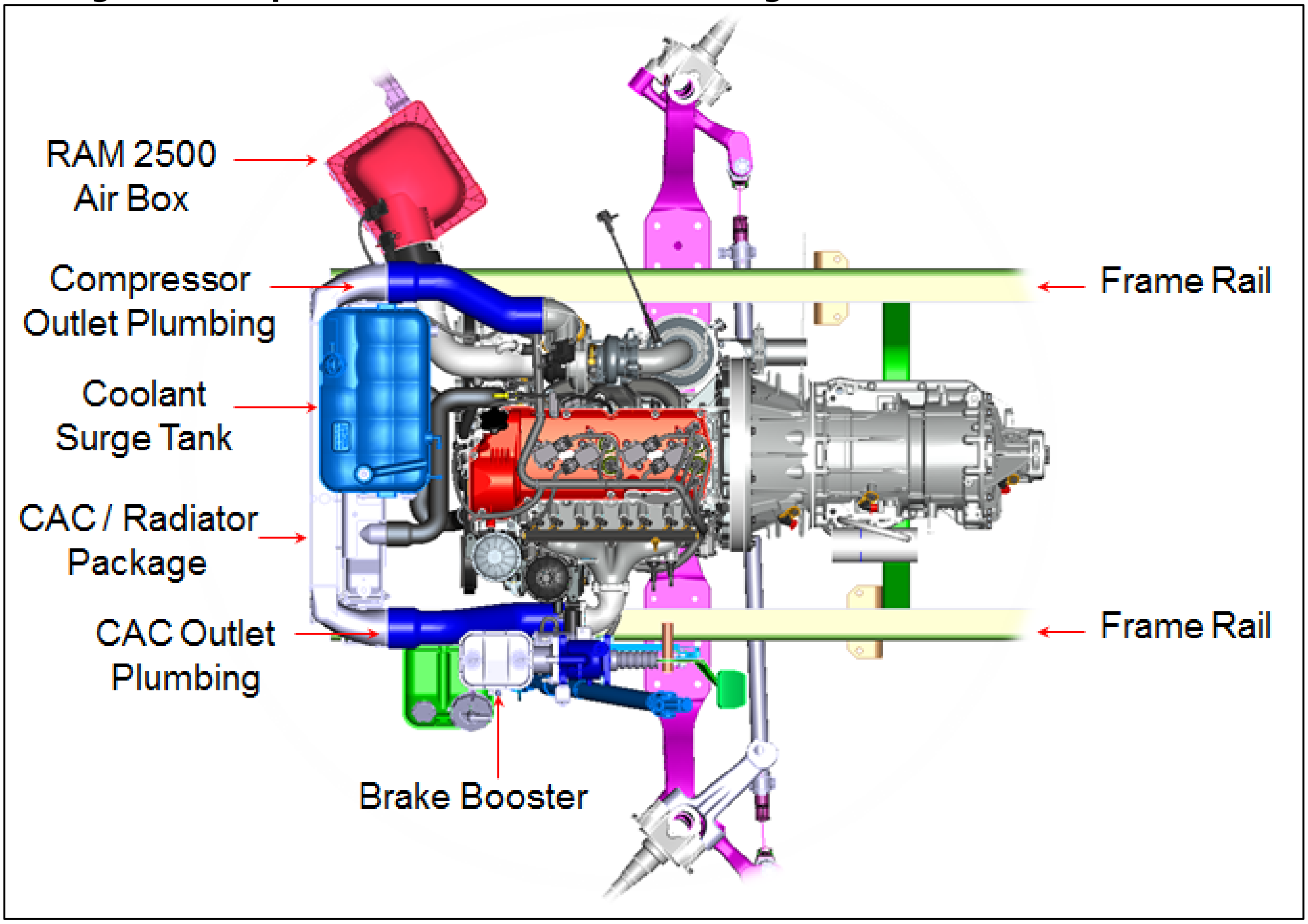
Figure 22: Isometric Right and Left Side Views of the Powertrain Package Installed in the Frame Rails



Source: Cummins

The powertrain package had to be located slightly further forward in the vehicle than ideally desired due to interference from the driver foot well of the vehicle cab. The gasoline baseline engine typical in this configuration has a central air intake system which does not interfere with the vehicle foot well. With this 4-cylinder configuration, the intake manifold is on the driver's side of the engine and there is little clearance between the air inlet to the intake manifold and the foot well. This forced the engine forward of the desired location but did not drive any additional design changes.

Figure 23: Top View of the Powertrain Package Installed in the Frame Rails



Source: Cummins

The exhaust connection from the engine to the vehicle was designed to connect in a similar location to the gasoline baseline engine location. The small engine size allowed for a much closer coupling of the after treatment to the engine than in the base gasoline vehicle configuration. This allows for improved emissions reduction potential during the cold start emissions testing. A short extension pipe is used to connect the exhaust outlet on the engine to the vehicle piping which replaces the section of exhaust where two catalysts were required for the V8 gasoline baseline.

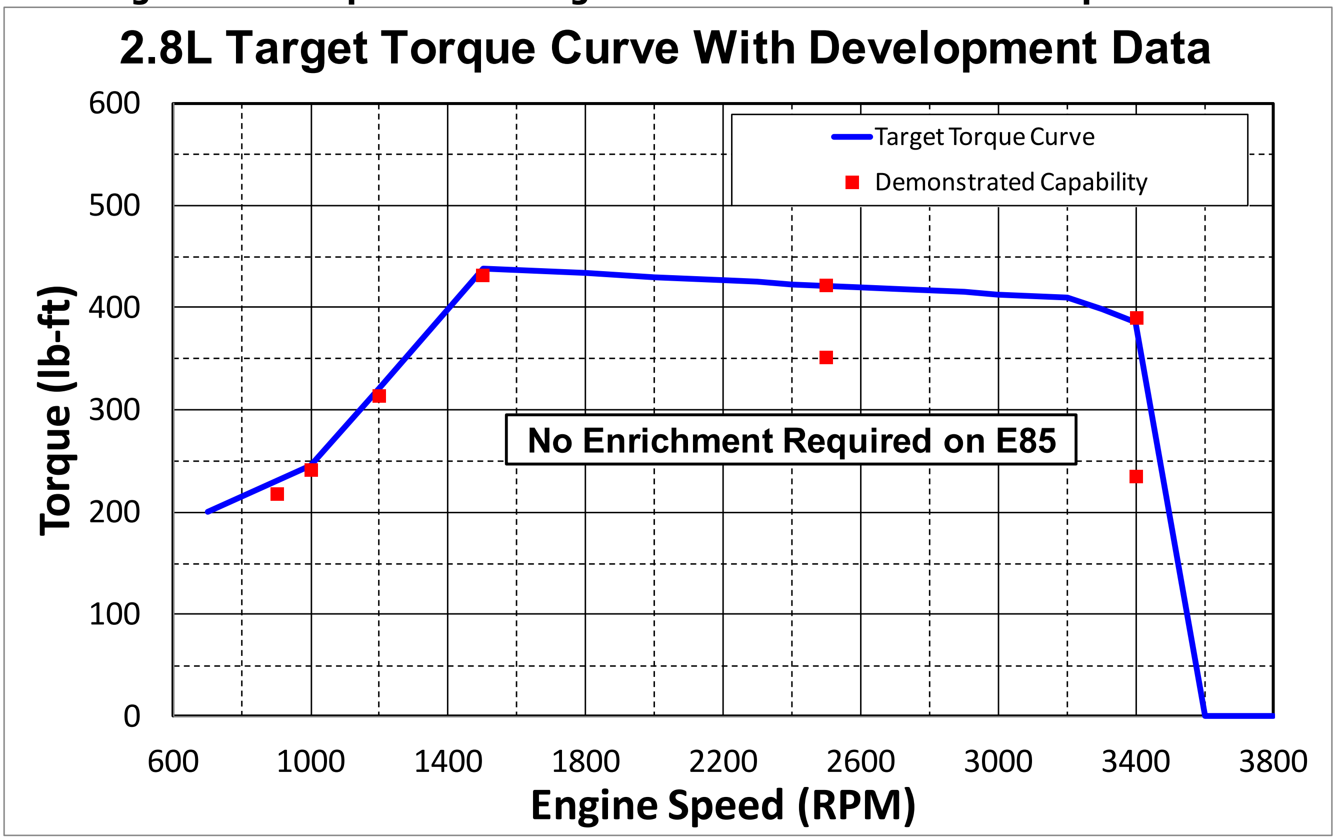
The air inlet plumbing for the engine was designed using the air filter box from a Ram 2500 pickup due to Cummins familiarity with this system. The air box houses the mass air flow meter for incoming fresh air sensing. The accuracy of this sensor is very important for proper engine air fuel ratio control. This sensing strategy is also very sensitive to air box mounting, type and configuration. Using the Ram air filter box provides excellent accuracy of the mass airflow sensing system for this program based upon the previous development of that system for production application.

Results

Steady State Engine Efficiency

Steady state engine mapping was completed using the previously described hardware and the high temperature capable best fit turbocharger. This turbocharger achieved the full target torque curve while maintaining acceptable turbocharger inlet temperature without fuel enrichment. The work completed while demonstrating the full torque curve showed that fuel economy, combustion performance, and engine stability were excellent over the entire engine map. Figure 24 shows the target torque curve with a few of the test cell demonstration data points. The engine is able to operate with no spark retard and at optimum combustion phasing at rated power due to the knock resistant engine design and fuel properties. The engine efficiency targets established for this project have been exceeded by the running engine over the majority of the engine operating space.

Figure 24: Comparison of Target and Demonstrated 2.8L Torque Curve

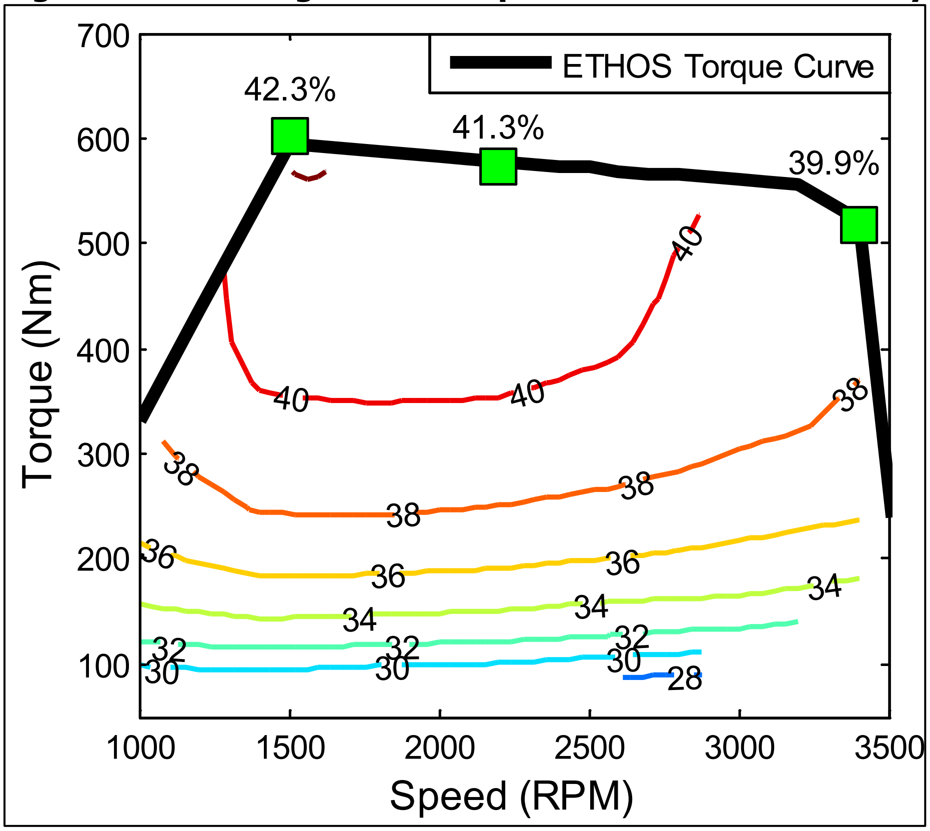


Source: Cummins

While the engine thermal efficiency results (Figure 25) are excellent and generally much better than the gasoline engine available in the class 4-6 market, the fuel properties cannot be ignored. The low lower heating value of ethanol still requires a higher volumetric fuel burn rate (Figure 26) with the good engine efficiency. Of prime concern to a vehicle owner is what the cost to operate the vehicle on ethanol would be relative to a similar vehicle being operated on gasoline.

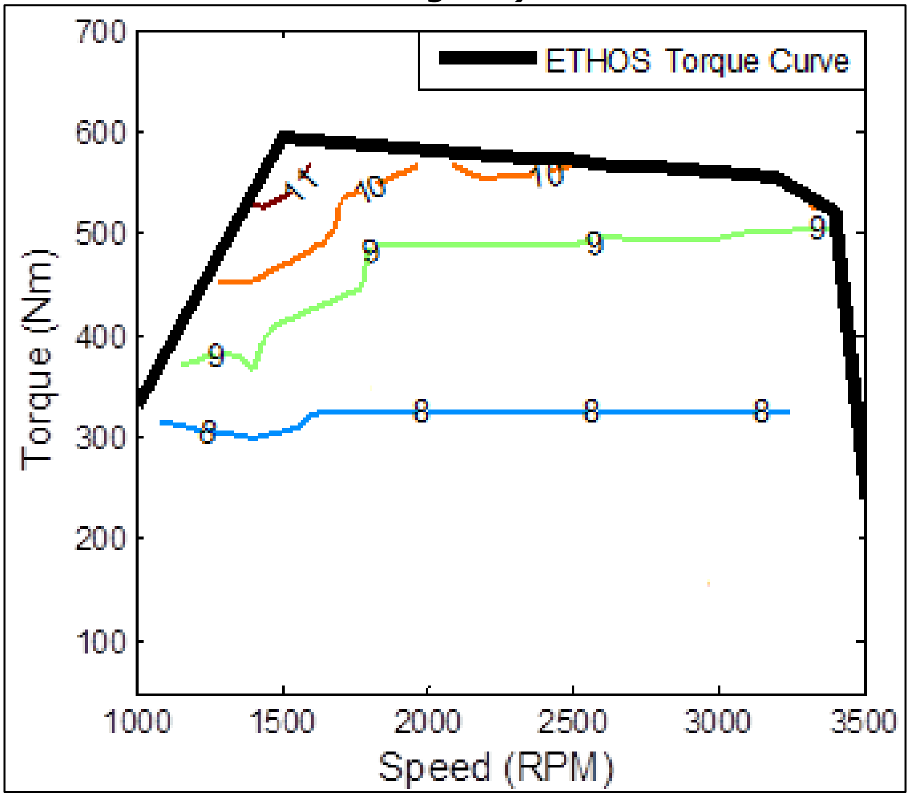
Figure 27 shows a comparison of the January 2014 fuel costs for the Sacramento area between gasoline and E85. Even though the cost of E85 is lower than the cost of gasoline, it's still 12 percent more expensive on an energy basis.

Figure 25: 2.8L Engine Fuel Map Brake Thermal Efficiency



Source: Cummins

Figure 26: 2.8L Engine Fuel Map 50% Mass Fraction Burned Location (Crank Angle Degrees)



Source: Cummins

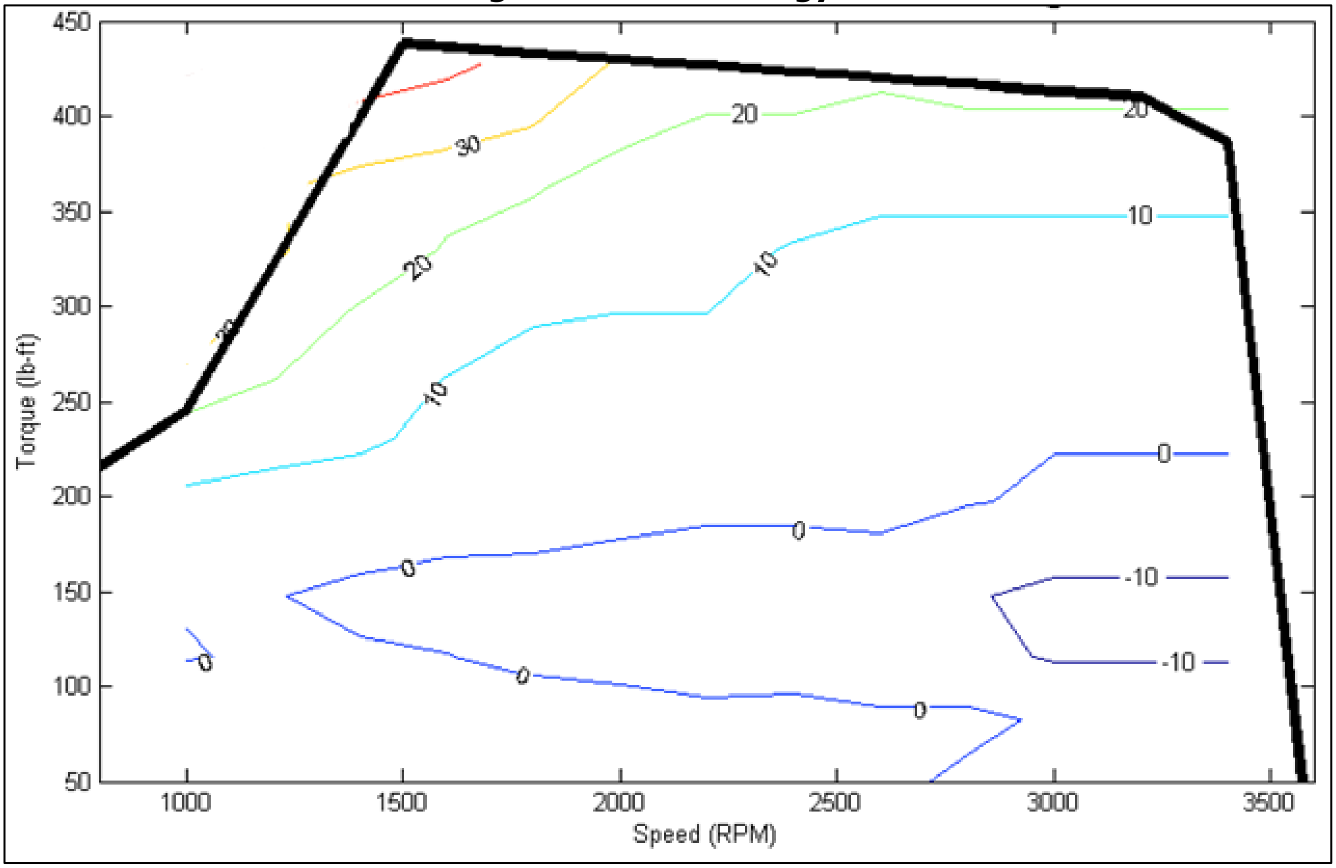
Figure 27: Fuel price comparison for E85 and gasoline in the Sacramento area for January 2014

- Gasoline = $3.80/gallon
  - 109,772 BTU/gallon
  - 28,887 BTU/$
- E85 = $3.19/gallon
  - 81,346 BTU/gallon
  - 25,500 BTU/$
- Diesel = $4.40/gallon
  - 127,464 BTU/gallon
  - 28,969 BTU/$
- E85 is 12% more expensive than both Gasoline and Diesel

Source: Cummins

To assess the "Ultra-Low Carbon Powertrain" efficiency, Cummins compared our obtained E85 results with published data from a state-of-the-art Gasoline Turbocharged Direct Injection engine. Figure 28 has also had the numbers adjusted to account for the fuel cost difference between E85 and gasoline. All E85 engine efficiency values were reduced by 12 percent and then compared to the gasoline engine efficiency values. A positive number in the figure represents the percent improvement of the E85 engine efficiency over the gasoline engine efficiency.

Figure 28: Fuel Cost Improvement Percentage of the E85 Engine Compared to a State-of-the-Art Gasoline Turbocharged Direct Injection Engine Accounting for a 12% Higher Cost E85 Energy Content



Source: Cummins

At low engine load values, the low engine friction capability of the gasoline engine as well as the optimized designs present in the market today make it very difficult to exceed the gasoline engine's efficiency by 12 percent using the optimized E85 engine. The high cylinder pressure requirements of an optimized E85 engine are a penalty in this lower load region. However, this is also an area where the fuel burn rate of either engine is low and a highly downsized engine such as this is unlikely to spend much time in the area of the map where there is a negative improvement relative to the gasoline engine. It is worth noting that the fuel price delta for California appears to be higher than many other areas of the country. Some areas of the country are even reporting E85 pricing to be at parity with gasoline pricing on an energy basis, which would further improve the fuel cost of this platform in those areas of the country.

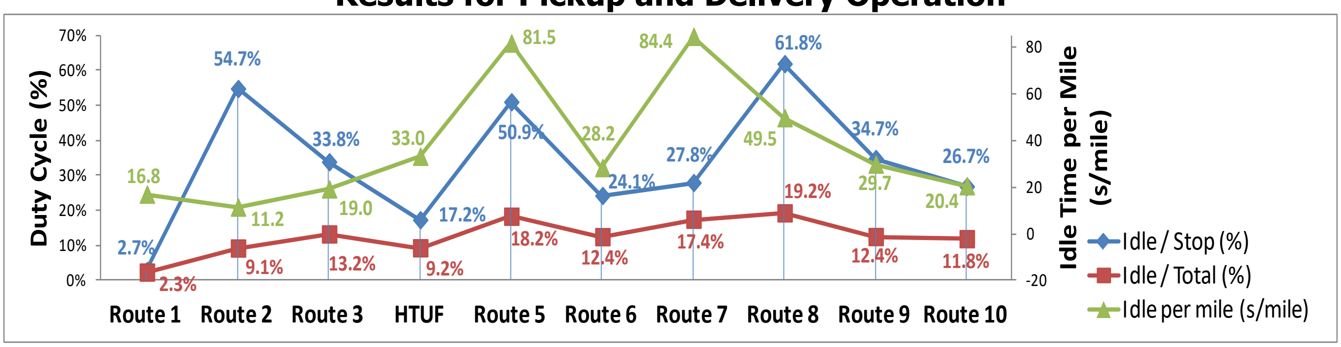
At medium and high engine load values, the optimized E85 engine efficiency exceeds the efficiency of the gasoline engine by 12 percent or significantly more in large regions of the operating space. The knock resistant engine design along with the favorable fuel properties for E85 allow for the good results in this area. The intended market for this engine has been the class 4-6 vehicle where the driver of such a vehicle will regularly operate the engine at high loads. By achieving a favorable efficiency relative to the gasoline engine at high loads where the volumetric fuel burn rate is high, the efficiency benefit of this engine platform will be readily apparent to a vehicle driver or fleet owner. This is all achieved while still delivering a better vehicle acceleration performance than the gasoline engines currently available in this market.

Transient Test Cell Results

The test cell setup described above includes the E85 optimized engine, stop/start components, and a motoring dynamometer to allow for transient cycle testing. The stop/start components included in this testing were the 12V smart alternator, two 12V absorbent glass mat batteries, battery sensors to determine the battery state, a 12V engine mounted starter, and a load bank to simulate the vehicle electrical loads applied to the batteries. Transient drive cycle testing was conducted to simulate a 20,000 lb. step van driving specific routes. The fuel burned during the cycle was measured and totaled for each cycle to determine the vehicle fuel economy for each cycle.

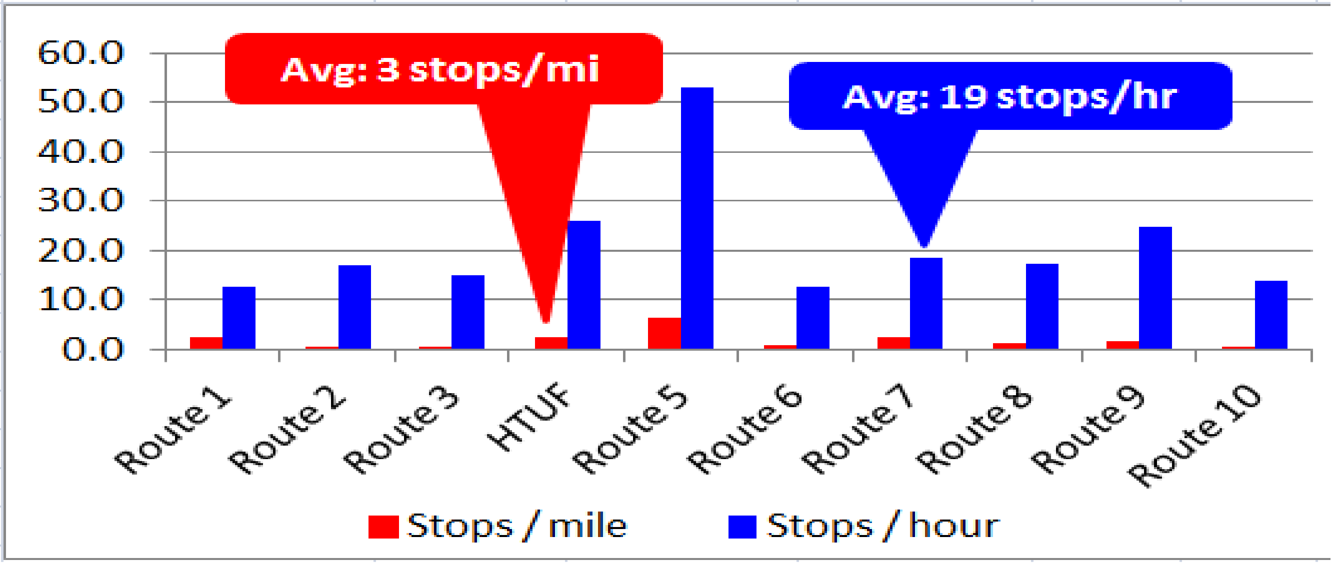
Ten vehicle routes were used during the modeling phase to assess the fuel economy benefits of the stop/start system. Two of those same vehicle routes were repeated in the test cell to validate the fuel economy improvement of the real system. The Hybrid Truck Users Forum Pickup and Delivery Route and Route 5 shown in Figure 29 and Figure 30 were used as they represent a variety of pickup and delivery drive cycles. Route 5 is an internal route from a Cummins field test data logger which has characteristics found to represent a worst-case type of pickup and delivery cycle in terms of fuel burned at idle. As can be observed in Figure 29, Route 5 has a very high amount of idle time measured in either idle time as a percentage of the total time or as idle time in seconds of idle per mile of driving. Cummins believes this represents the entitlement for a stop/start system if all of the idle time on a route such as this can be replaced with engine off operation. The actual amount of idle time that could be eliminated is a function of the electrical load on the batteries during the engine off time, the size of the battery pack, and the charging rate during the driving period of the cycle. For the demonstration vehicle being used for this project, Cummins was able to achieve idle off operation during all of this idle time, so the route was simulated as such in the test cell. This represents best case fuel economy improvement for a stop/start system on this route.

Figure 29: Drive Cycle Route Metrics Used to Simulate and Test Fuel Economy Results for Pickup and Delivery Operation



Source: Cummins

Figure 30: Drive Cycle Route Metrics Used to Simulate and Test Fuel Economy Results for Pickup and Delivery Operation



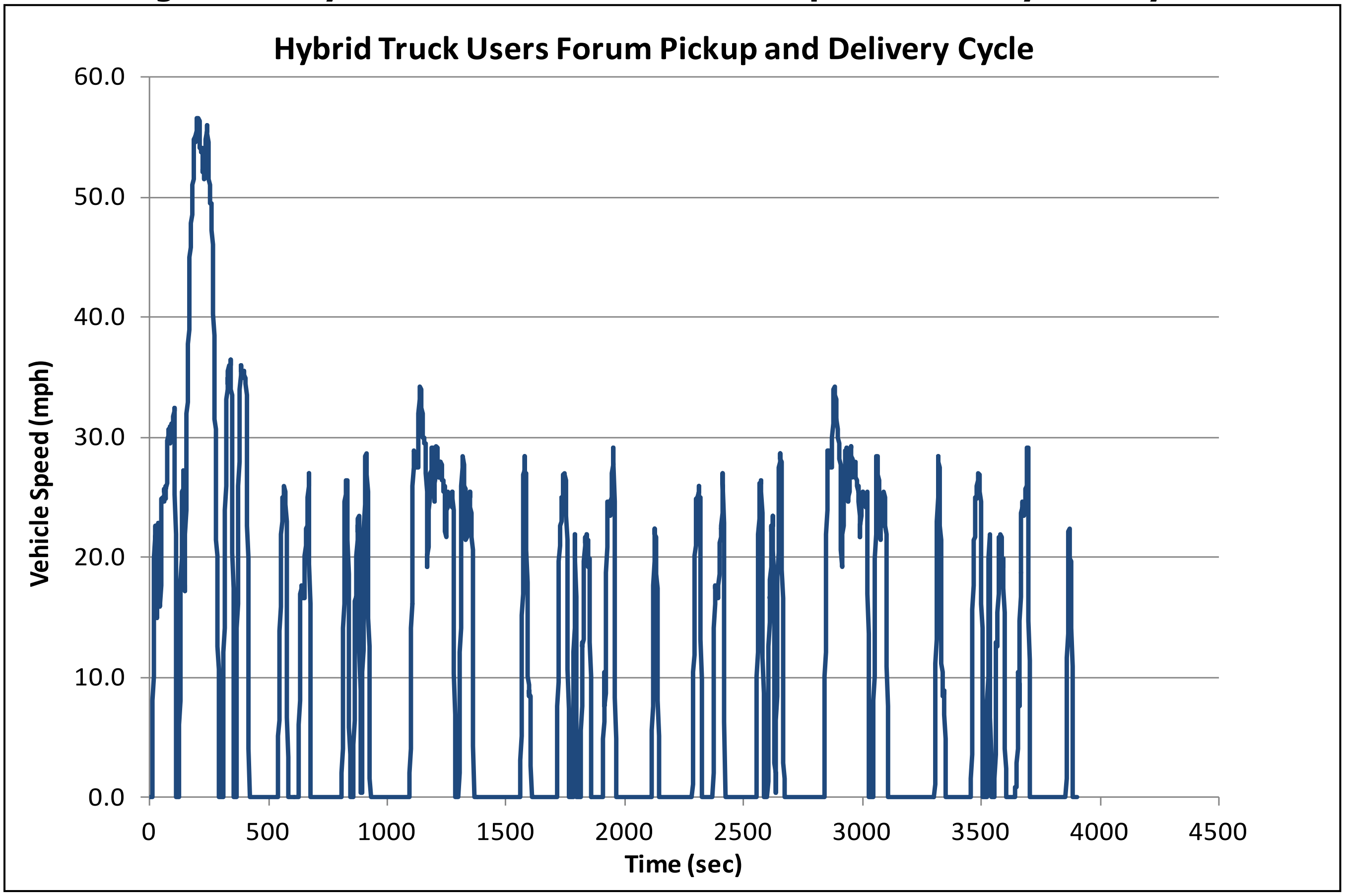
Source: Cummins

The test cell testing focused on validating the improvements which came from smart alternator controls as well as reduced idle time through engine shut down. These two features were evaluated independently and together to measure drive cycle fuel economy improvement. Fuel economy performance was collected as a baseline for each of the two routes. The baseline drive cycle has no idle shutdown events and includes the vehicle electrical load as well as conventional alternator control. A conventional alternator control strategy was simulated using the smart alternator hardware for the baseline cycle so that hardware switching was not required to alternate from the conventional to smart alternator control strategies. Conventional alternator control was simulated by setting the smart alternator hardware to target a system voltage of 13.5V for the entire cycle which is equivalent to the control approach of a conventional alternator.

The engine shutdown events were originally intended to shut the engine down and then start back up using the engine mounted starter. It was found that this operation of starting the engine which was mechanically connected to a dynamometer imposed a much higher starting load on the starter than what would occur in the vehicle. This imposed an unusual load on the batteries and caused the starter to overheat easily. The cycle was then switched to simulate engine shutdown events by allowing the dynamometer to hold the engine to idle speed with the fuel turned off to the engine and the alternator charging disabled. The post processing work for each cycle then included adding in an additional amount of fuel which represents the fuel used to get the engine up to idle speed after each engine restart. This was a small amount of fuel, but it was necessary for characterizing the fuel economy improvements associated with stop/start operation.

The translation of the steady state fuel economy results to a route representing an actual customer drive cycle has been done using the Hybrid Truck Users Forum Pickup and Delivery cycle. This route represents a low speed city delivery type drive cycle common for delivery applications in the Class 4-6 market. As Figure 31 depicts the majority of the cycle occurs below 30 miles per hour. The Hybrid Truck Users Forum route has been setup in the test cell to mimic the speed and load the engine would see when powering a 20,000 lb. vehicle driving this route. As the majority of this drive cycle is below 30 miles per hour the transmission torque converter losses for an automatic transmission factor heavily into the fuel economy results.

Figure 31: Hybrid Truck Users Forum Pickup and Delivery Test Cycle



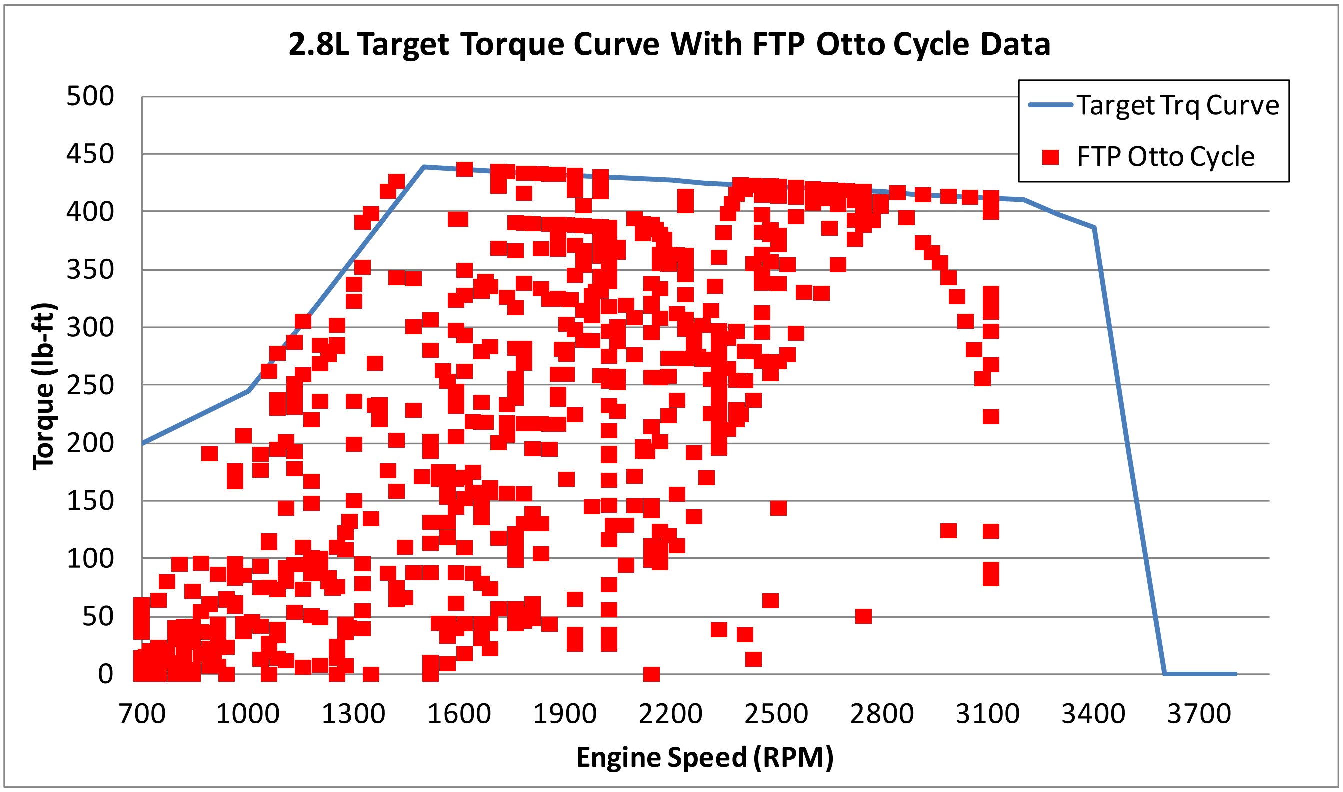
Source: Cummins

Running our optimized E85 engine on the Hybrid Truck Users Forum cycle resulted in a 9 percent mile per gallon fuel penalty compared to the gasoline engine common in this market. This means that the engine achieved 15-20 percent higher efficiency on E85 than the comparable gasoline engine in this market.

Criteria Emissions Feasibility

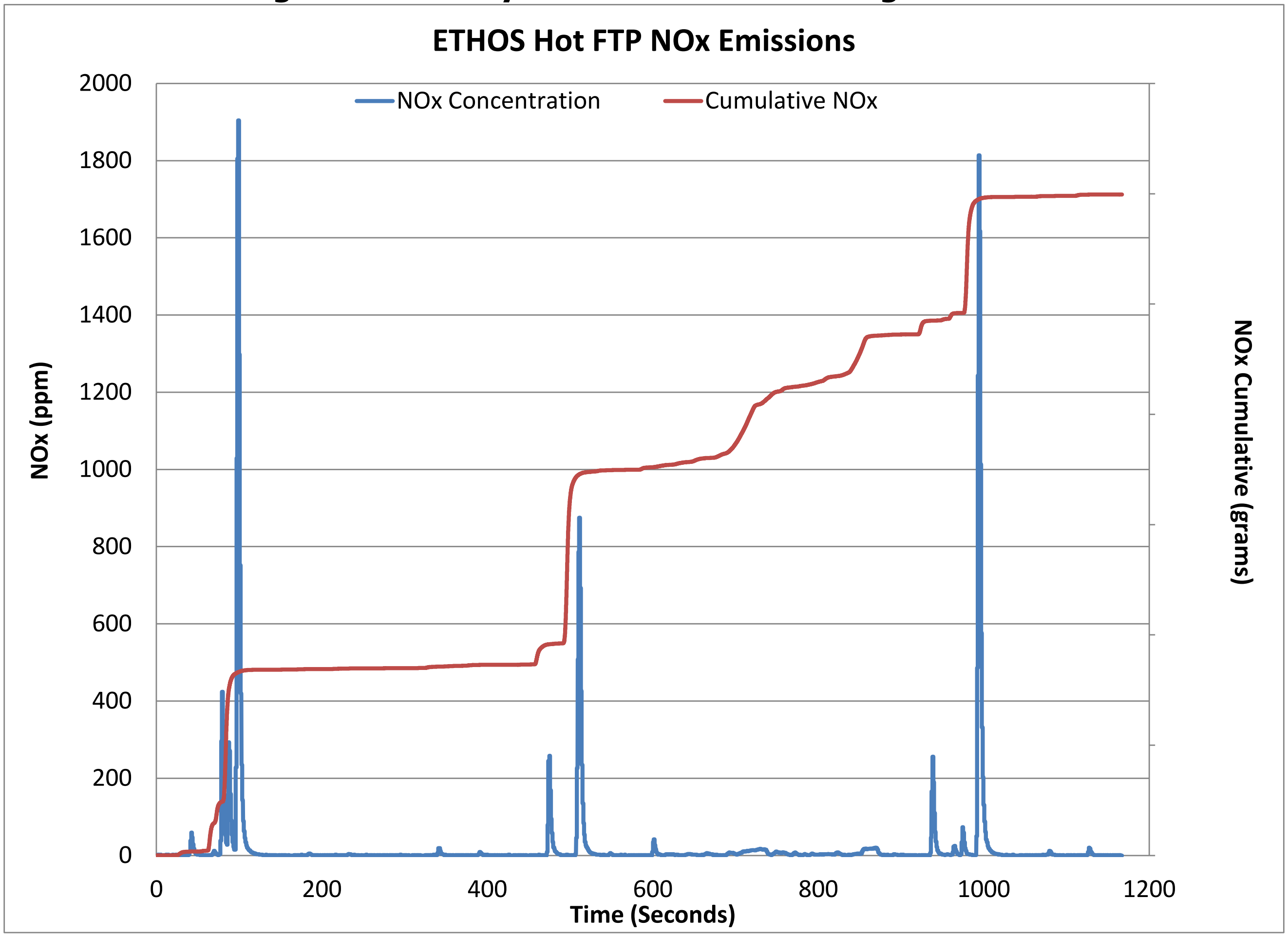
Emissions performance was assessed using the Heavy Duty "OTTO" Cycle on the engine test stand by measuring NOx, CO, and NMHC emissions. A fully warmed engine was operated to fine tune air to fuel ratio control over the cycle as well as to screen several catalysts for best performance. Figure 32 shows the "OTTO" cycle relative to the full torque curve for the 2.8L engine. An air-to-fuel ratio target equal to the stoichiometric ratio was used for the entire engine operating space to achieve the best emissions performance. Using the optimum catalyst emissions performance below the regulated limits was achieved for all measured emissions constituents. Figure 33 shows the NOx emissions over the cycle. Emissions are generally near zero except during a few events in the cycle where further air-to-fuel ratio control work could improve the results further. As can be seen by the cumulative NOx emissions line, these few areas of control challenges generate nearly all the emissions over the cycle. The cumulative NOx axis is zoomed to show the opportunities for improvement in emissions control, however the total emissions results are currently well below the regulated limits. Cold and Hot Heavy Duty "OTTO" cycles were run to generate a composite emissions test with the selected catalyst formulation. All system out emissions were below the 2010 ARB emissions limits for this non-certified test cell measurement. The application of 3-way catalysts to this ethanol optimized system achieves superior emissions results.

Figure 32: Heavy Duty "OTTO" Cycle Operating Points for the 2.8L E85 Torque Curve



Source: Cummins

Figure 33: Hot Cycle NOx Emissions Using E85 Fuel



Source: Cummins

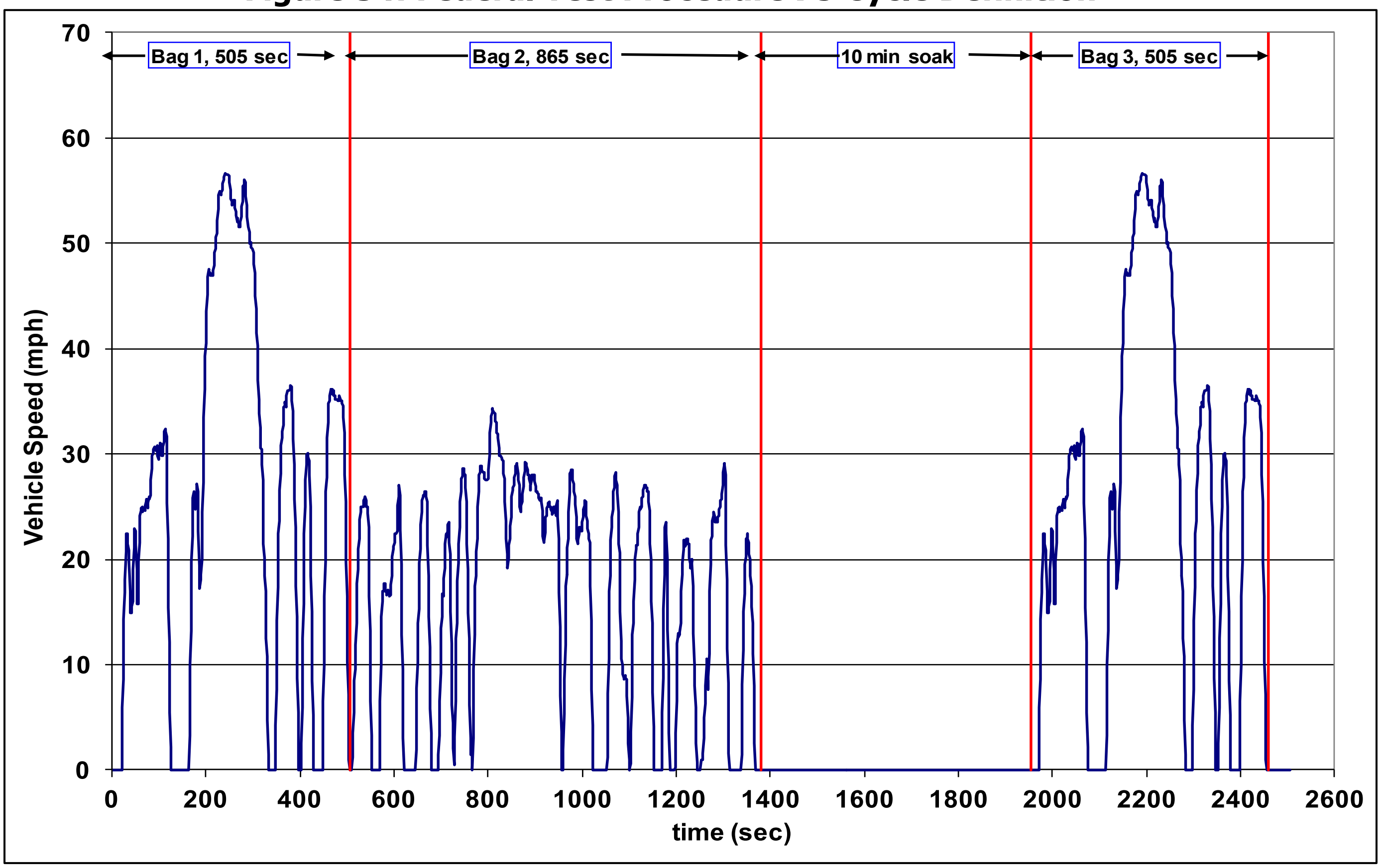
Chassis Roll Fuel Economy

Vehicle testing was conducted on a chassis dynamometer at the Center for Automotive Research at Ohio State University. Coast down coefficients were determined at the Transportation Research Center such that the rolling and aerodynamic characteristics of the step van could be set on the chassis dynamometer to match the real characteristics of the vehicle. The gasoline version of the step van was used to determine the vehicle characteristics. This same vehicle was then used for baseline gasoline testing prior to removal of the gasoline powertrain and installation of the integrated system developed under this program. A diesel version of the same step van was provided for evaluation by Federal Express so that diesel baseline fuel economy and emissions data could be collected.

To cover the full class 4-6 vehicle weight range, simulated vehicle weights of 15,000, 17,500, 20,000, and 23,000 lbs. were run on the dynamometer. Four vehicle drive cycles were selected for evaluation at most of the weights covering Pickup and Delivery and highway drive cycles. The EPA Federal Test Procedure (FTP-75) (Figure 34), Highway Fuel Economy Test (Figure 35), Hybrid Truck Users Forum Pickup and Delivery (Hybrid Truck Users Forum pickup and delivery), and a Cummins Pickup and Delivery cycle (Figure 36) were used to represent real world driving cycles. Several repeats at each vehicle mass and drive cycle were collected to ensure consistent results.

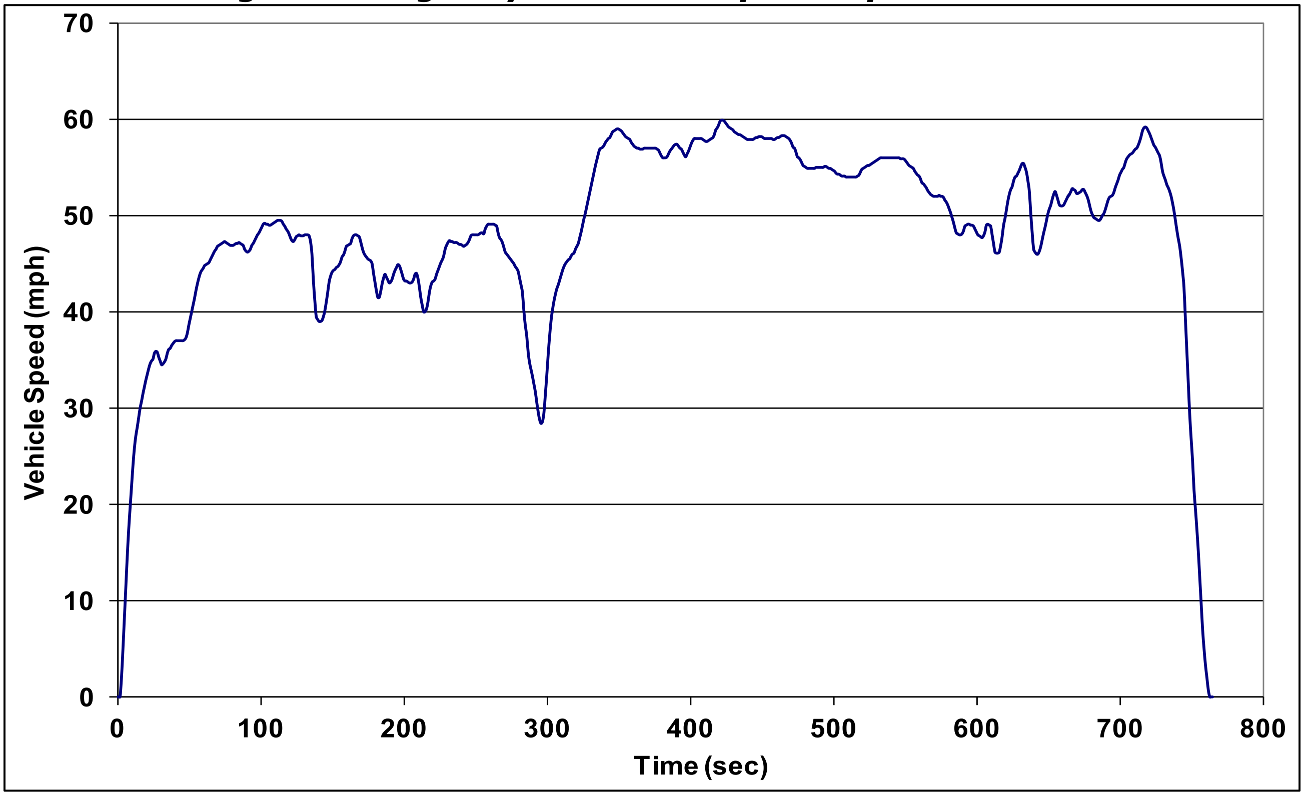
The stop/start system was included for all cycle operation on the E85 powertrain. Neither the diesel nor the gasoline baseline systems included stop/start operation as this option is not available for either of these applications. On the baseline systems the engine idles when the vehicle speed is 0 miles per hour except for key-off events in the Hybrid Truck Users Forum pickup and delivery cycle. The key-off events in the Hybrid Truck Users Forum pickup and delivery cycle were used to represent longer idle events where a conscientious driver would already be turning off the engine to make a package delivery for instance. Both the baseline and stop/start capable systems were shut down during the key-off period. The E85 powertrain with stop/start would shut down during all 0 miles per hour vehicle speed events except during the start of the FTP cycle where engine coolant temperature had not reached the threshold to allow stop/start activity. During stop/start events the key switch is left in the "run" position such that vehicle electrical loads are draining the batteries. If left in this state long enough, the engine would restart due to low battery level. There were no engine-off periods of sufficient length in the tested cycles to cause the engine to restart for low battery level.

Figure 34: Federal Test Procedure 75 Cycle Definition



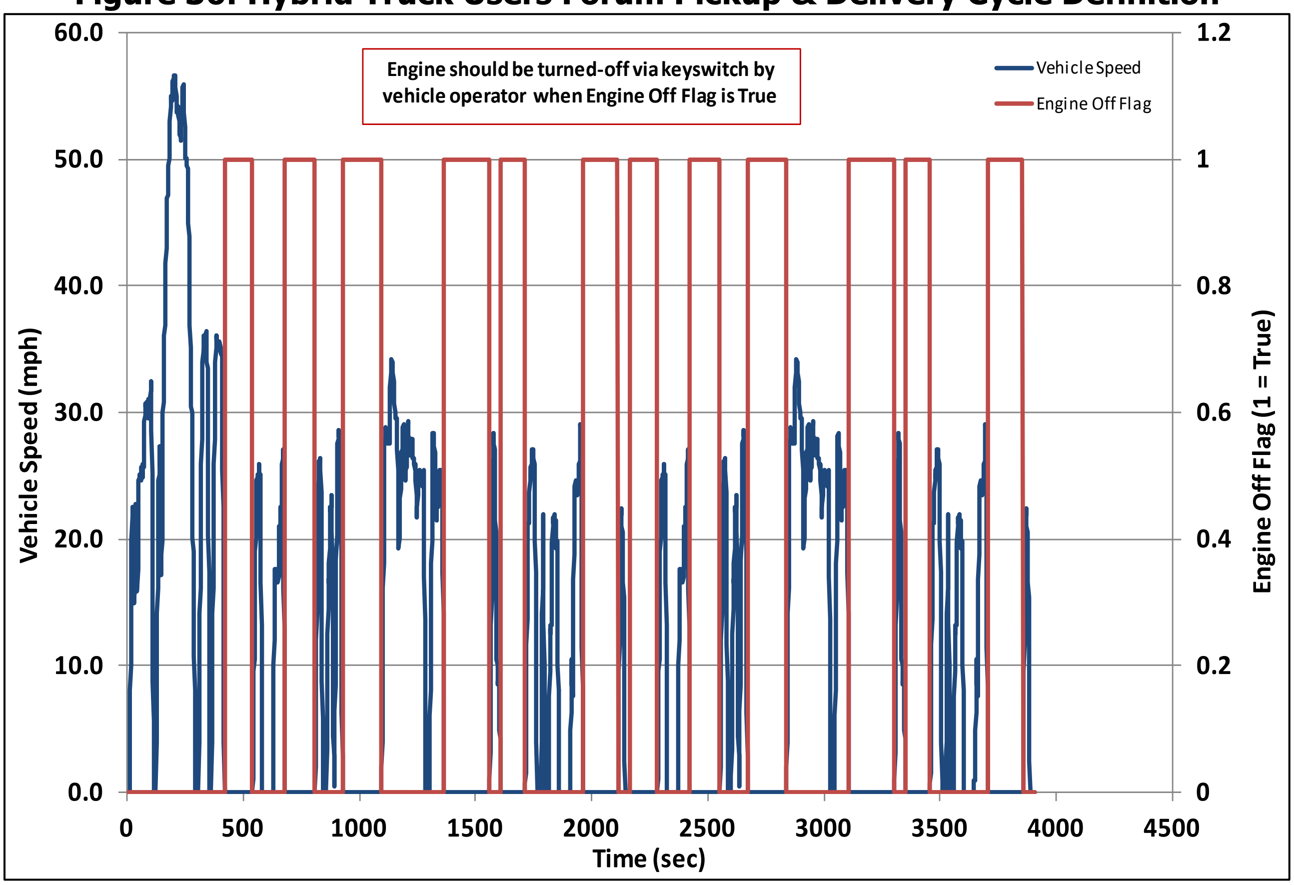
Source: Cummins

Figure 35: Highway Fuel Economy Test Cycle Definition



Source: Cummins

Figure 36: Hybrid Truck Users Forum Pickup & Delivery Cycle Definition



Source: Cummins

The project goal for CO_2 emissions is a 50 percent reduction relative to current powertrains on a well to wheel basis. The project goal for fuel economy was to achieve a level that would allow for good market adoption of the E85 powertrain. This means that the fuel cost should be competitive to the powertrains currently in the market when the fuel economy and fuel cost are looked at together. This target was rolled up in a $/mile measure and compared to the baseline powertrains.

CO_2 emissions results measured during vehicle dynamometer testing represent a partial accounting along the full well to wheel path. The measured CO_2 emissions are a result of the elemental carbon contained in the fuel being combusted and oxidized to CO_2. Table 3 shows the elemental carbon emissions contained in the various fuels being represented in this testing. The elemental carbon values are very similar for the various fuels. The values for Corn Ethanol and Cellulosic Ethanol from Farmed Trees represent the pure ethanol values for both elemental carbon and well to tank carbon. When 15 percent California reformulated gasoline was added to these values the following two rows are generated for an E85 blend. To calculate the full well to wheel CO_2 emissions that can be compared for CO_2 reduction, the tank to wheel CO_2 emissions collected on the dynamometer must be adjusted by the following formula:

Well to wheels CO_2 = (g/mile CO_2 as measured) * (Elemental Carbon / Well to Tank Carbon)

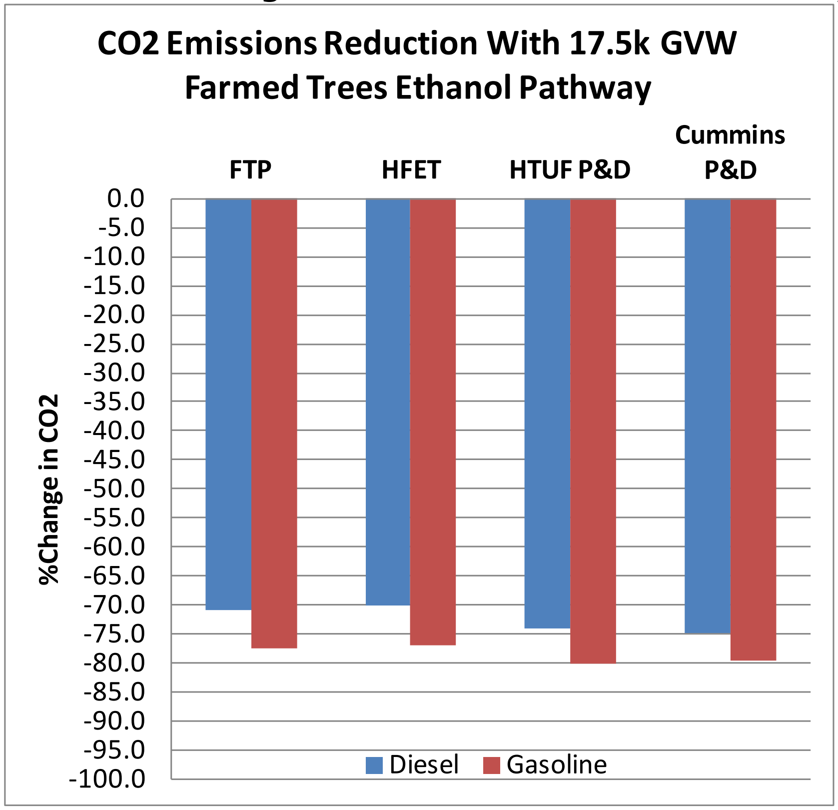
This means that the measured CO_2 while running on the dynamometer only accounts for the elemental carbon contained in the fuel and not the fully burdened fuel pathway. For instance, measured emissions from running the gasoline fuel vehicle would need to be increased by roughly 36 percent to account for all the carbon emission coming from the processing and transportation of diesel fuel allowing it to be available at the pump.

Table 3: Fuel Pathway Carbon Values for Various Fuels

Fuel	Elemental Carbon	Well to Tank Carbon
	G CO2/MJ	G CO2/MJ
California Reformulated Gasoline	72.90	98.95
California Ultra-Low Sulfur Diesel	74.10	98.03
Corn Ethanol	71.02	65.66
Cellulosic Ethanol from Farmed Trees	71.02	21.40
California E85- Corn	71.34	70.65
California E85- Farmed Trees	71.34	33.03

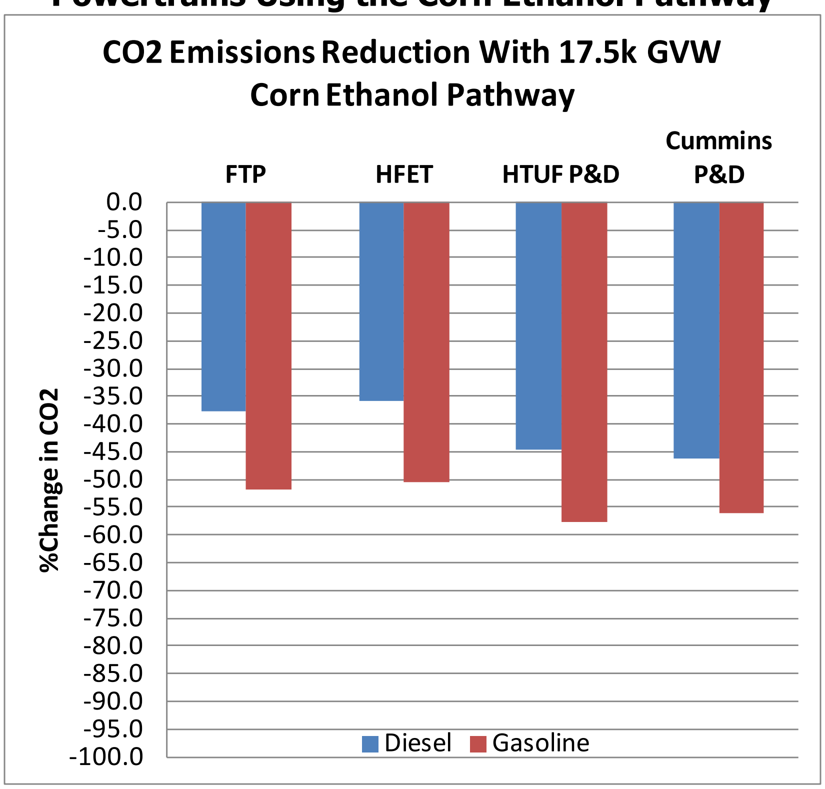
Source: Cummins

Figure 37: CO2 Emissions Reduction for E85 Powertrain Relative to Alternate Powertrains Using the Farmed Trees Ethanol Pathway



Source: Cummins

Figure 38: CO2 Emissions Reduction for E85 Powertrain Relative to Alternate Powertrains Using the Corn Ethanol Pathway

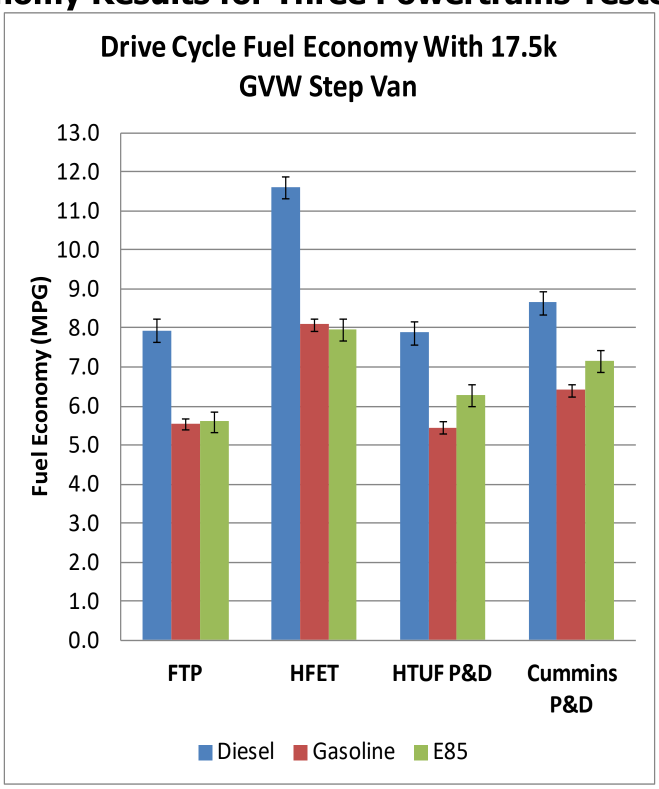


Source: Cummins

Figure 37 and Figure 38 show the well to wheel CO_2 comparisons for the E85 powertrain to the baseline systems assuming a farmed trees pathway in Figure 37 and a corn pathway in Figure 38. This comparison was made at a 17,500 lb. Gross Vehicle Weight (GVW) which falls in the middle of Class 5. When comparing the CO_2 emissions of the E85 powertrain to the diesel baseline, 70-75 percent emissions reduction is seen across the various drive cycles assuming farmed trees ethanol and 35-45 percent reduction assuming corn ethanol. When comparing the CO_2 emissions of the E85 powertrain to the gasoline baseline, 75-80 percent emissions reduction is seen across the various drive cycles assuming farmed trees ethanol and 50-57 percent reduction assuming corn ethanol. These results significantly exceed the project target of 50 percent CO_2 reduction on cellulosic ethanol in an E85 blend. These results also show that nearly 50 percent CO_2 reduction is possible even when considering the corn based ethanol path which is currently the primary path for ethanol in the market.

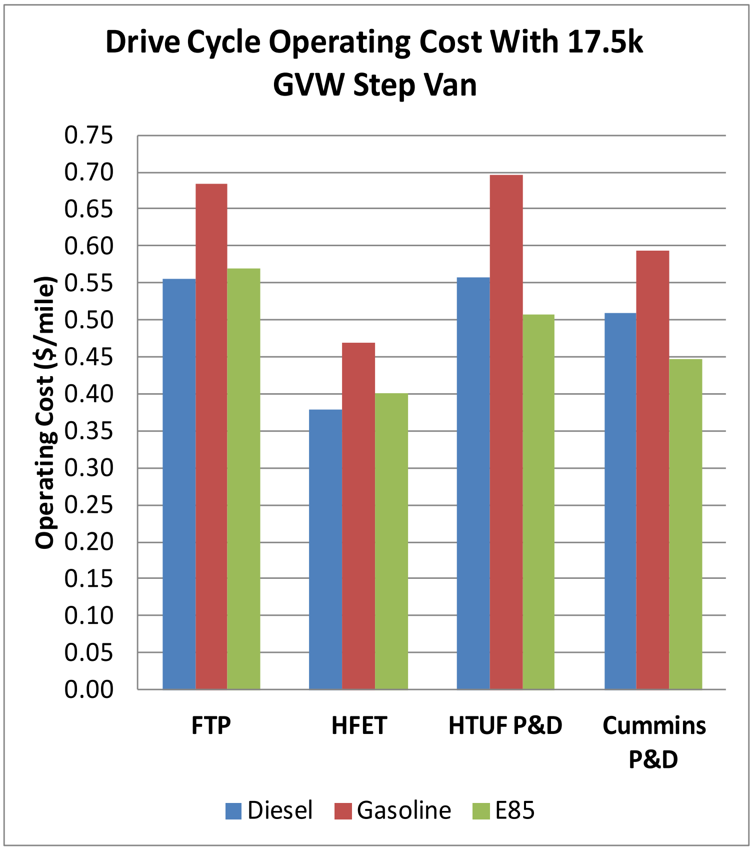
Although it is unreasonable to expect simliar fuel economy to the diesel powertrain considering the diesel engine has both good efficiency and a fuel with a very high energy content per gallon, indeed the mile per gallon results for the diesel powertrain exceed both the converted gasoline and E85 engine in all cycles. The fuel economy comparison of the three powertrains at 17,500 lbs. GVW can be seen in Figure 39. Figure 39 error bars are plus and minus three sigma of cycle results. Cummins is pleased to share that even though E85 has approixmately 30 percent less energy per gallon than gasoline, fuel economy of the E85 powertrain is equal or better than a conventional V8 engine. To compute the fuel cost Cummins assumed values of $4.40/gallon, $3.80/gallon, and $3.19/gallon for diesel, gasoline, and E85 respectively. The E85 powertrain was able to achieve equivalent fuel cost in $/mile compared to the diesel engine and noticeably better operating cost compared to the gasoline engine as seen in Figure 40.

Figure 39: Fuel Economy Results for Three Powertrains Tested at 17.5k lbs GVW



Source: Cummins

Figure 40: Fuel Cost Results for All Three Powertrains Tested at 17.5k lbs GVW



Source: Cummins

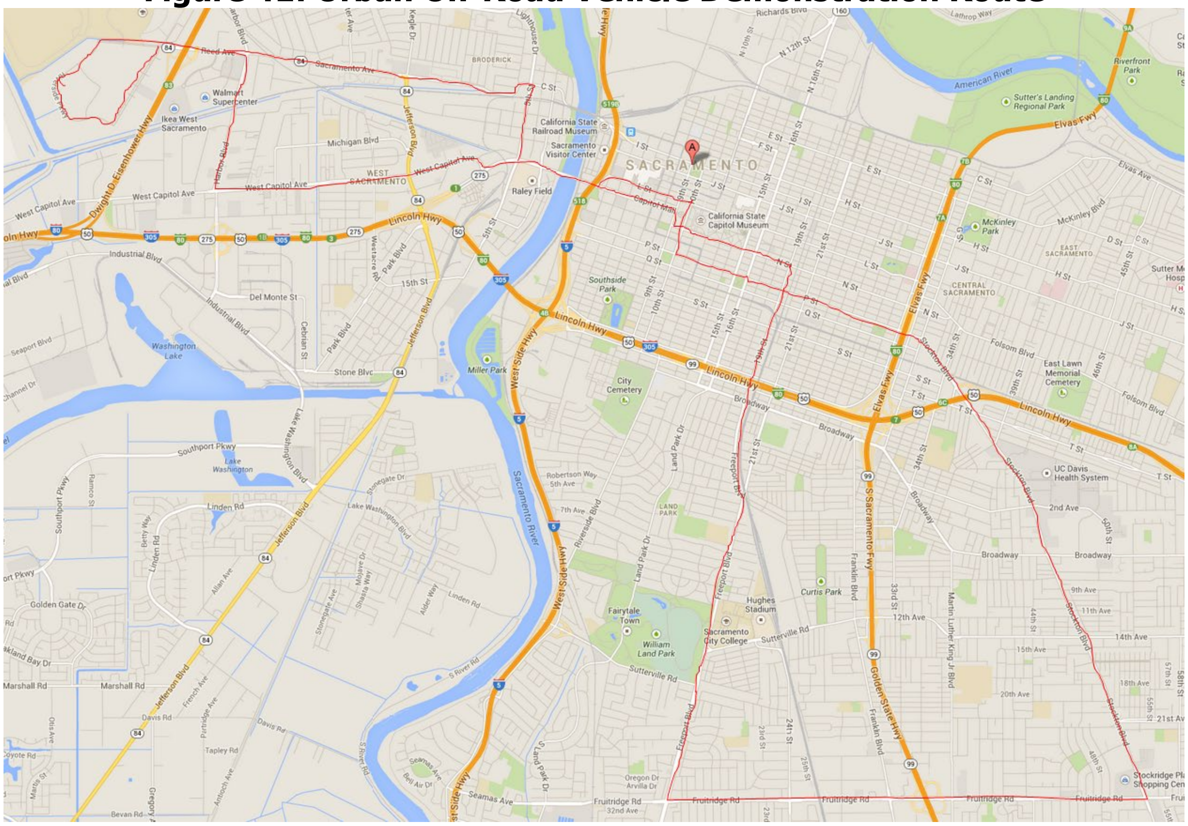
Vehicle Demonstration

The wrap-up of the Ultra-Low Carbon Powertrain project was a full vehicle demonstration to verify the performance of the system in the real world and show publicly the system developed through this project. The demonstration was conducted in the Sacramento area with a GVW of 13,700 lbs. The empty weight of this vehicle was ~9000 lbs such that the additional weight would be slightly more than a typical delivery weight for this type of package delivery vehicle. This weight would be very typical for a highly loaded Class 4 vehicle or a partially loaded Class 5 vehicle. The weight reduction of the powertrain of nearly 700 lbs relative to the diesel powertrain translates directly to lower vehicle weight in real world operation. The demonstration vehicle is shown in Figure 43.

Two routes were defined for the vehicle demonstration phase. An urban route which runs past the Capitol building and through downtown was selected to represent in town drive cycles. This route has a very high number of vehicle stop events and has a low average vehicle speed. A rural route was also selected which runs up through the city of Woodland to represent a rural delivery route. This route has a higher average vehicle speed and includes a lower number of stops. These two routes help span the normal expected operation for typical class 4 and 5 delivery vehicles as well as highlighting the benefits of stop/start in delivery drive cycles. Figure 41 and Figure 42 show the urban and rural routes respectively. 2000 miles were driven total during the 6-week vehicle on-road demonstration. The urban route averaged 7.1 miles per gallon (MPG) while the rural route averaged 8.0 MPG. These fuel economy results align well with the chassis roll fuel economy results and the lower vehicle weight run for the on-road demonstration. The urban route covered 23 miles with a typical vehicle speed between 20 and

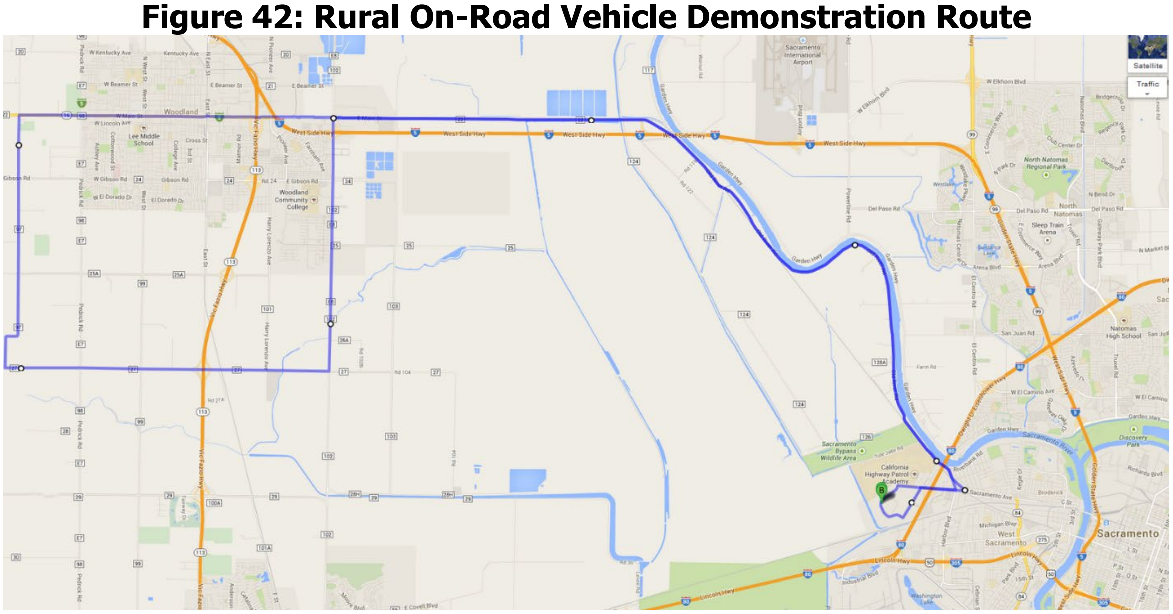
30 miles per hour. This route averaged over 40 stops per hour with around 14 percent of the drive time spent at stop. The rural route covered 48 miles with a typical vehicle speed between 40 and 50 miles per hour. This route averaged nearly 10 stops per hour with only 5 percent of the drive time spent at stop. These cycle metrics fall within the range of the 10 cycles used to investigate the fuel economy potential of a stop/start system during the development phase of this project.

Figure 41: Urban On-Road Vehicle Demonstration Route



Source: Cummins

Figure 42: Rural On-Road Vehicle Demonstration Route



Source: Cummins

Figure 43: "Ultra-Low Carbon Powertrain" Demonstration Vehicle

Source: Cummins

CHAPTER 3: Advancements and Success

Advancements in Science and Technology

The major goal for this project was to deliver a 50 percent CO2 reduction on a well-to-wheels basis for the E85 powertrain compared to current competitive options, while keeping the projected cost good enough to achieve good market penetration with this powertrain concept. This CO2 reduction goal has been greatly exceeded using cellulosic based biofuels. Unfortunately, these fuels are not yet commercially available although this powertrain technology now exists to make good use of these fuels when they become available. While the goal of this project was CO2 reduction on cellulosic biofuels, the powertrain efficiency was good enough that the results are still very interesting when using current production corn-based ethanol.

While many of the components used to achieve this level of powertrain efficiency are not technological developments associated with this project, the combustion system design and arrangement of available components exhibited a step forward in technological capability. The combustion system achieved over 42 percent brake thermal efficiency which is excellent for stoichiometric spark ignited combustion. This allows for the use of a 3-way catalyst which allows for a lower system cost than a diesel engine which would typically achieve this level of efficiency. This low system cost can allow for good market penetration of this powertrain with diesel like efficiency and torque curve, but gasoline like NVH properties. The power density achieved on this engine is significantly greater than currently available in this market. This project achieved 90 hp/L where current offerings are typically less than 60 hp/L.

The stop/start system developed and implemented on this project is novel and a technological step forward for this market due to the adaptation of a heavy-duty automatic transmission. Stop/start systems currently available in other medium duty markets are currently limited to manual transmissions, but manual transmission powertrains are not common in the North American medium duty market space.

Public Assessment of Success and Benefits

Cummins proposed and conducted the grant program by conceiving and developing multiple technologies for greenhouse gas reduction in the context of medium duty truck application. The program focused on the integrated design of these technologies for maximum CO2 benefit. Part of the program involved Cummins putting the vehicle on the road in California as a demonstration of the technology and to begin assessing market interest in this path of low carbon powertrain development. Furthermore, Cummins has made public announcements with planned presentations on the primary program results as an additional method to raise awareness of the benefits demonstrated under the program. Additional briefings with regulatory agencies are also scheduled to provide updates on the program successes and to inform additional stakeholders. However, at the time of preparing this final report, it is not yet clear if this technology path will gain market acceptance for higher volume penetration. Therefore, Cummins endeavors to project the larger benefits potentially available from widespread adoption of this technology. Additional benefits are potentially available by scaling this technology and augmenting it which will also be discussed.

The class 4-6 medium duty truck market in North America is roughly 90,000 units per year. This powertrain was designed and demonstrated capable to power these units and offer a cost and performance competitive powertrain option for this vehicle space. If these units average 20,000 miles per year with 2/3 of these vehicles currently being diesel achieving 7.9 MPG and the other 1/3 being gasoline achieving 5.5 MPG, approximately 150 Million gallons of diesel fuel consumption and 110 Million gallons of gasoline consumption could be avoided. Complete changeover of these 90,000 units would require 290 Million gallons of E85 with an assumed 6.2 MPG. This changeover in fuel would save 1.7 Million tons of CO2 emissions if the ethanol for the E85 production came from corn or 2.6 Million tons of CO2 emissions if the ethanol was produced from farmed trees.

If the class 4-6 medium duty truck market in California were 15,000 units per year with an even split between diesel and gasoline powertrains, the same assumptions would mean approximately 19 Million gallons of diesel fuel consumption and 27 Million gallons of gasoline consumption could be avoided. Complete changeover of these 15,000 units would require 48 Million gallons of E85 with an assumed 6.2 MPG. This changeover in fuel would save 308 thousand tons of CO2 emissions if the ethanol for the E85 production came from corn or 459 thousand tons of CO2 emissions if the ethanol was produced from farmed trees.

While CO2 emissions are easily demonstrable as a result of this project work, criteria emissions reductions are less easily quantified. Both diesel and gasoline engines currently have very similar criteria emissions regulations and the E85 powertrain was designed to target similar emissions levels. The project worked conducted under this program confirmed the expectation that engine dyno certification to 2010 emission levels is cost effectively feasible. Further it is worth considering the difficulty in improving emission of this E85 powertrain, to emissions levels lower than the currently applicable 2010 standards, compared to a diesel powertrain. Based upon current three-way catalyst equipped engine dyno certified products and data collected in this program, it is possible to project that lower emission certification levels are possible and possibly more easily than on a lean combustion system such as diesel. One clear emissions benefit of the Ultra-Low Carbon Powertrain project E85 powertrain, including stop/start, is the elimination of idle emissions, which are the highest in urban areas where frequent stops are common. The stopping of the engine at idle is also a driver and pedestrian benefit by reducing the minimizing the vehicle noise at idle. This makes it easier for drivers to be more alert to pedestrians and for pedestrians on the street experience less noise pollution at traffic lights.

Cummins believes the “Ultra-Low Carbon Powertrain Program” provided a clear vision of a technology pathway that can make significant contribution to air quality programs throughout the United States and particularly California. If the market acceptance of E85 with the performance this powertrain demonstrates can grow, and if the fuel volume growth to support it grows from second generation fuels, it is possible that this technology pallet, and evolutions of it, will grow to be a measurable contributor to the future of transportation technology.

CHAPTER 4: Conclusions and Recommendations

Conclusions

This E85 powertrain project has exceeded the goal of 50 percent CO2 reduction on cellulosic ethanol relative to currently available gasoline and diesel powertrains. The fuel economy demonstrated by this platform will match current gasoline offerings to achieve the same tank mileage and a much lower cost to operate based on the state fuel cost assumptions. The efficiency of this platform allows for a fuel cost very similar to a diesel platform, but it is expected the E85 powertrain cost would be lower than the diesel system cost based on the integration of lower cost 3-way catalyst after treatment technology. The torque and power capabilities of this platform achieve the targets required for good vehicle acceptance. The stop/start system integration with quick vehicle launch capability should also meet customer drivability requirements. The cost and performance of this platform should allow for good market penetration if this platform were to go to production assuming the volume of E85 filling stations continues to increase. This project shows that the current increase in E85 filling stations and the research currently being done to bring cellulosic ethanol to market could be paired with this powertrain technology to deliver a cost-effective solution for reducing CO2 emissions while still providing good customer value.

Recommendations for Future Projects

While the results demonstrated during this project are very good on their own, several options to further this work have been considered.

Increasing awareness of the current results, capabilities, and future possibilities for the powertrain as it exists today would be a good next step. This could include refinement to allow for longer term demonstration with potential customers and demonstration at industry trade events. These activities would help generate market pull for this type of technology which could allow for business case development for production by Cummins, as well as to continue roll out of E85 fuel stations to make the fuel more readily available. Market demand and fueling availability would be two big hurdles that could be better understood by taking the technology to a next level of vehicle demonstration.

A second option is to evaluate the fuel economy and CO2 reduction benefits at the other end of the displacement and compression ratio trade-off. One goal of this project was developing a highly downsized engine to maximize drive cycle BMEP and therefore operate at high efficiency. By operating at high BMEP, a lower compression ratio is required to achieve peak torque requirements while staying within the cylinder pressure limits. It is possible to go to much higher compression ratio on a larger displacement engine that would run at lower BMEP and achieve the same torque and power requirements. This engine could also be combined with cooled exhaust gas recirculation (EGR), a known efficiency improver in knock-limited engines. EGR acts as a displacement reducer and would be an air handling challenge to include on the current high BMEP engine. The combination of cooled EGR and higher compression ratio could result in a further step change increase in fuel economy improvement and CO2 reduction.

A third option for future project work could be to increase the amount of electrification on the current engine. Because the E85 engine has a significantly lower after treatment cost compared to a current diesel engine some amount of increased electrification could be added and still have a powertrain that is cost competitive with currently available powertrains. A good next step would be to add torque assist with increased energy recovery during decelerations and increase energy storage. System optimization down this path could provide a powertrain with similar cost to the current diesel system but much better fuel cost and further improved CO_2 reduction relative to the current project.

GLOSSARY

BRAKE MEAN EFFECT PRESSURE (BMEP)—Mean effective pressure calculated from measured brake torque.[1]

BRITISH THERMAL UNIT (BTU)—The standard measure of heat energy. It takes one Btu to raise the temperature of one pound of water by one degree Fahrenheit at sea level. MMBtu stands for one million Btu.

CALIFORNIA AIR RESOURCES BOARD (ARB)—The "clean air agency" in the government of California whose main goals include attaining and maintaining healthy air quality, protecting the public from exposure to toxic air contaminants, and providing innovative approaches for complying with air pollution rules and regulations.

CARBON DIOXIDE (CO2)—A colorless, odorless, nonpoisonous gas that is a normal part of the air. Carbon dioxide is exhaled by humans and animals and is absorbed by green growing things and by the sea. CO2 is the greenhouse gas whose concentration is being most affected directly by human activities. CO2 also serves as the reference to compare all other greenhouse gases (see carbon dioxide equivalent).

E85—E85 motor fuel is defined as an alternative fuel that is a blend of ethanol and hydrocarbon, of which the ethanol portion is 75-85% denatured fuel ethanol by volume and complies with the most current ASTM specification D5798.[2]

ELECTRONIC CONTROL MODULE (ECM)—A system that controls a series of actuators in the diesel engine to ensure optimal engine performance through electronic control. Modern diesel engines have a number of sensors within the engine and machine, which provide readings to the electronic control unit (ECU).

EXHAUST GAS RECIRCULATION (EGR)—An emissions reduction technique used in gasoline and diesel engines wherein a portion of an engine's exhaust gas is recirculated back to the engine cylinders.

FEDERAL TEST PROCEDURE (FTP)—A series of tests defined by the United States Environmental Protection Agency (U.S. EPA) to measure tailpipe emissions and fuel economy of passenger cars (excluding light trucks and heavy-duty vehicles).

GROSS VEHICLE WEIGHT (GVW)—The maximum operating weight/mass of a vehicle as specified by the manufacturer including the vehicle's chassis, body, engine, engine fluids, fuel, accessories, driver, passengers, and cargo, but excluding that of any trailers.

HORSEPOWER (HP)—A unit for measuring the rate of doing work. One horsepower equals about three-fourths of a kilowatt (745.7 watts).

MEGAJOULE (MJ)—A joule is a unit of work or energy equal to the amount of work done when the point of application of force of one newton is displaced one meter in the direction of the

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[1] Mean Effective Pressure (https://en.wikipedia.org/wiki/Mean_effective_pressure)

[2] E85 Fuel Definition (https://afdc.energy.gov/laws/6210)

force. It takes 1,055 joules to equal a British thermal unit. It takes about one million joules to make a pot of coffee. A megajoule itself totals one million joules.

MILES PER GALLON (MPG)—A measure of vehicle fuel efficiency. Miles per gallon or MPG represents "Fleet Miles per Gallon.” For each subgroup or "table cell," MPG is computed as the ratio of the total number of miles traveled by all vehicles in the subgroup to the total number of gallons consumed. MPGs are assigned to each vehicle using the EPA certification files and adjusted for on-road driving.

NITROGEN OXIDES (OXIDES OF NITROGEN, NOx)—A general term pertaining to compounds of nitric oxide (NO), nitrogen dioxide (NO2), and other oxides of nitrogen. Nitrogen oxides are typically created during combustion processes and are major contributors to smog formation and acid deposition. NO2 is a criteria air pollutant and may result in numerous adverse health effects.

REVOLUTIONS PER MINUTE (RPM)—The number of turns in one minute. It is a unit of rotational speed or the frequency of rotation around a fixed axis.

THREE-WAY CATALYST (TWC)—A three-way catalyst oxidizes exhaust gas pollutants — both hydrocarbons (CmHn) and carbon monoxide (CO) — and reduces nitrogen oxides (NOx) into the harmless components water (H2O), nitrogen (N2), and carbon dioxide (CO2).

UNITED STATES DEPARTMENT OF ENERGY (U.S. DOE)—The federal department established by the Department of Energy Organization Act to consolidate the major federal energy functions into one cabinet-level department that would formulate a comprehensive, balanced national energy policy. DOE's main headquarters are in Washington, D.C.

VOLT (V)—A unit of electromotive force. It is the amount of force required to drive a steady current of one ampere through a resistance of one ohm. Electrical systems of most homes and offices have 120 volts.