CALIFORNIA
ENERGY COMMISSION

ENERGY RESEARCH AND DEVELOPMENT DIVISION

FINAL PROJECT REPORT

High Efficiency Ultra-low Emissions Heavy-duty Natural Gas Engine Research and Development

September 2025 | CEC-500-2025-042

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Agreement Number: 500-18-003

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PREFACE

The California Energy Commission's (CEC) Energy Research and Development Division manages the Gas Research and Development Program, which supports energy-related research, development, and demonstration not adequately provided by competitive and regulated markets. These natural gas research investments spur innovation in energy efficiency, renewable energy and advanced clean generation, energy-related environmental protection, energy transmission and distribution and transportation.

The Energy Research and Development Division conducts this public interest natural gas-related energy research by partnering with RD&D entities, including individuals, businesses, utilities and public and private research institutions. This program promotes greater gas reliability, lower costs and increases safety for Californians and is focused in these areas:

- Buildings End-Use Energy Efficiency
- Industrial, Agriculture and Water Efficiency
- Renewable Energy and Advanced Generation
- Natural Gas Infrastructure Safety and Integrity
- Energy-Related Environmental Research
- Natural Gas-Related Transportation

High Efficiency Ultra-low Emissions Heavy-duty Natural Gas Engine Research and Development is one of the final reports for the Developing Innovative Low Emission Natural Gas Engine and Vehicle Technology for Medium- and Heavy-Duty Vehicles project (500-18-003) conducted by the National Renewable Energy Laboratory. The information from this project contributes to the Energy Research and Development Division's Gas Research and Development Program.

For more information about the Energy Research and Development Division, please visit the CEC's research website (www.energy.ca.gov/research/) or contact the Energy Research and Development Division at ERDD@energy.ca.gov.

ABSTRACT

Natural gas engines benefit from a favorable hydrogen-to-carbon ratio of the fuel molecule and relatively lower fuel cost per unit energy compared to diesel or other liquid petroleum fuels. However, their lower engine thermal efficiencies reduce the greenhouse-gas-emission and total-cost-of-ownership advantages for most heavy-duty vehicle applications, making the adoption of natural gas vehicles challenging. Hence improvements in spark-ignited natural gas engine efficiencies and base engine cost reductions can further the beneficial use of natural gas in heavy-duty vehicle applications. This project aims to drive simultaneous improvements in fuel efficiency and cost while achieving ultra-low emissions.

Cummins, Inc. undertook this research effort to develop a high efficiency, low emission 15-liter heavy-duty natural gas engine designed specifically for natural gas operation. Various design choices to optimize the engine for natural gas operation enabled a greater than 10 percent brake thermal efficiency improvement and a greater than 20 percent engine system cost reduction over commercially available natural gas heavy-duty engines. The engine also achieved diesel-like performance while still being able to deliver extremely low emissions equivalent to current natural gas engine products including 0.02 grams per brake horsepower hour of oxides of nitrogen emissions. This report summarizes the results of the development effort and innovations made to achieve the ambitious targets for emissions, efficiency, performance, and robustness. The project concluded that, with technology additions, a heavy-duty natural gas engine can be nearly as efficient as a modern diesel engine but with the advantage of lower emissions.

Keywords: brake thermal efficiency, NOx emissions, natural gas, engines, research and development, heavy-duty vehicles

Please use the following citation for this report:

Shah, Jay (Cummins, Inc.). 2025. *High Efficiency Ultra-low Emissions Heavy-duty Natural Gas Engine Research and Development.* California Energy Commission. Publication Number: CEC-500-2025-042.

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Executive Summary

While utility-served natural gas plays a small role as a transportation fuel in California, the gas demand for the natural gas vehicle (NGV) sector is expected to grow in the near-term. Programs such as the Low Carbon Fuel Standard incentivize the use of biomethane in NGVs from waste streams such as landfills, wastewater treatment plants, municipal solid waste, and dairies to decarbonize the transportation sector. However, the California Air Resources Board's Advanced Clean Trucks Regulation will phase in zero-emission technologies for heavy-duty vehicles over the next two decades. A decline in fossil gas and biomethane as transportation fuel is expected in the long term as manufacturers comply with this regulation. NGV adoption in interim years may still occur, however, so research to improve the efficiency, cost, and performance of natural gas engines can deliver near-term benefits for applications where zero-emission vehicle options are not yet available.

Project Purpose and Approach

The costs of diesel engine systems have increased over the past several years as on-engine and aftertreatment emissions controls have been implemented to meet regulations. The costs for these systems are expected to increase further to support the capability needed to achieve ultra-low emission levels. By comparison, spark-ignited natural gas (NG) engines can manage emissions with a simpler and lower-cost combustion system and aftertreatment system. This emissions control architecture has remained relatively stable for the past four decades and has already demonstrated its capability to enable NG engines to certify to ultra-low emission standards.

The NG engines currently on the market are not as efficient as diesel engine technologies typically used in commercial vehicle applications. For perspective, a 12–15-liter displacement diesel engine used for Class 8 heavy-duty truck applications is generally more than 44 percent efficient at converting fuel to useful work. An equivalently sized NG engine using current technology is 36–38 percent efficient. This negative trade-off in fuel efficiency reduces the advantage in greenhouse gas (GHG) emissions, practically resulting in only marginally improved GHG emissions for NG engines relative to diesel engines. This project aimed to drive improvements in fuel efficiency to levels like those of conventional diesel vehicles, enable a 20 percent system cost reduction, and maintain ultra-low emissions. With these advancements, NG vehicles can deliver more incremental benefits over diesel in the near term.

The project approach involved system design and analysis, architecture selection and engine design, multi-cylinder engine build and testing, and final engine demonstration with a working prototype in a laboratory setting. Cummins, Inc. led the project, with oversight from the National Renewable Energy Laboratory and a consortium of agencies, including the U.S. Department of Energy, the California Energy Commission, and the South Coast Air Quality Management District.

Key Results

This project resulted in the first purpose-designed heavy-duty NG engine, compared to previous NG engine designs that were modified from diesel engines to achieve improved efficiency while maintaining ultra-low oxides of nitrogen (NOx) emission levels with diesel-like performance and reduced costs. The engine met the project objectives by:

- Demonstrating 42-percent peak brake thermal efficiency against a requirement of 41-43 percent. That is an 11-percent fuel consumption improvement over current NG engine technology.
- Demonstrating 40.2-percent steady state certification cycle average brake thermal efficiency. That is a 13-percent fuel consumption improvement over current NG engine technology.
- Demonstrating the capability of meeting current product heavy-duty NG level emissions, including low NOx, and the readiness to meet future emissions regulations well into the 2030s.
- Demonstrating diesel-like torque curve capability and validating robust operation.
- Achieving an estimated 31-percent engine system cost reduction over current NG engine technology, beyond the target 20 percent, utilizing a reduced cost aftertreatment system. This is an additional capital cost reduction on top of the operating cost reductions from improving fuel consumption.
- Designing and developing a spark-ignited suited pent-roof combustion system for improved closed cycle efficiency.

These achievements will materialize in future heavy-duty NG engine products that can open up NG as a fuel to a new customer base of fleets. Ratepayers benefit from NG use in the transportation sector because it increases utilization of the gas system and enables recovery of more fixed costs from fleets that transition from diesel to NG. Other benefits include incremental GHG emission reductions, especially when coupled with biomethane, and NOx emission reductions.

Knowledge Transfer and Next Steps

Cummins, Inc. advanced the technology readiness level of higher efficiency ultra-low emissions heavy-duty NG engines to technology readiness level 6 (application relevant scale), but additional development is required before commercial adoption, including regulatory dynamometer and in-use emissions certification tests. Cummins plans to use the learnings from this project in its future product launches (2027+) and make the key learnings gained from this project available to the public through technology transfer activities, including conference presentations and technical papers. Interim learnings and updates were shared with industry stakeholders over the course of the project at events such as the 2021 and the 2023 Natural Gas Vehicle Technology Forums. Emissions regulators will be able to use the results to confirm that the next generation of NG engines can deliver a lower CO2 solution while still maintaining the capability to achieve low NOx emissions. This technology also serves as baseline architecture of other technology development projects underway with the U.S. Department of Energy to further efficiency improvements.

CHAPTER 1: Introduction

Background

The cost of diesel engine systems has increased dramatically over the past several years as on-engine and aftertreatment emission controls have been installed to meet regulations. Components like common rail fuel systems, exhaust gas recirculation systems (EGR), variable geometry turbochargers, diesel particulate filters, and selective catalytic reduction systems are the major contributors to the system cost increase. The costs for these systems are expected to increase to support the capability needed to meet ultra-low emissions.

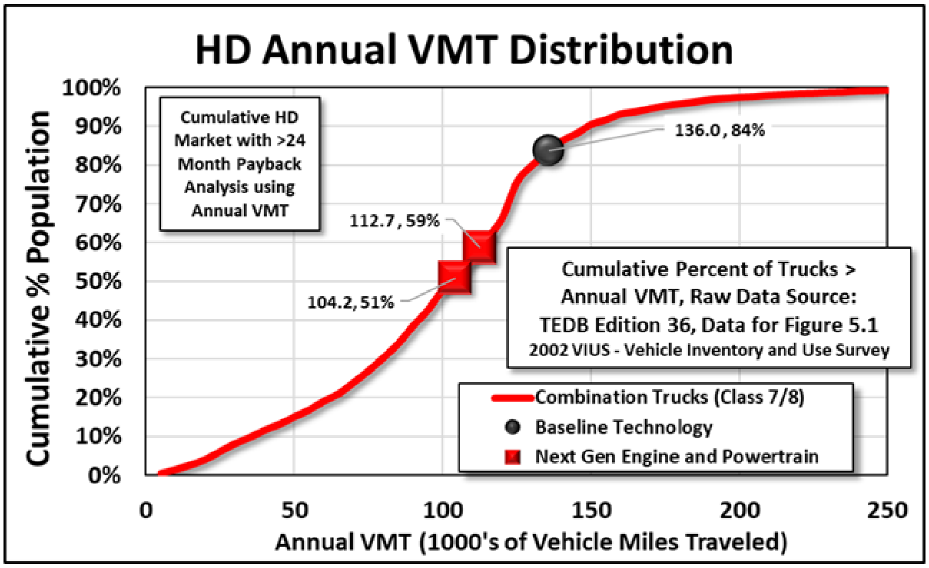
By comparison, spark-ignited (SI) natural gas (NG) engines manage emissions with a simpler and cheaper stoichiometric combustion system and a three-way catalyst (TWC) aftertreatment system. This emissions control architecture has remained relatively stable for the past four decades. As evidenced by the commercial certification of three optional low-nitrogen-oxides (NOx) emissions engines by Cummins, Inc., the stoichiometric combustion and TWC aftertreatment systems emission control is also capable of extremely low emissions capability.

Inherently, all NG engines benefit from a favorable hydrogen-to-carbon ratio of the fuel molecule compared to other liquid petroleum fuels, resulting in significant potential for lower tailpipe greenhouse gas (GHG) emissions from NG engines. Today however, the best NG engines in the market are not as efficient as diesel engine technologies typically used in commercial vehicle applications. For perspective, a 12–15-liter (L) displacement diesel engine used for heavy-duty truck applications will generally be more than 44 percent efficient at converting fuel to useful work. An equivalently sized NG engine using current technology will be 36–38 percent efficient. This negative tradeoff in fuel efficiency reduces the advantage in GHG emissions, practically resulting in only marginally improved GHG emissions for NG engines relative to diesel engines.

The impact of the lower fuel economy of NG engines on operating cost is often more than offset by the relative advantage in fuel cost per unit energy of NG compared to diesel or other liquid petroleum fuels. While significant for high mileage applications, in many commercial vehicle markets the total cost of ownership (TCO) is a more important factor in purchasing decisions than efficiency. A customer common metric of technology viability is the payback period. This is typically expressed as the time (in months) it takes to recover the increase in capital expense (CAPEX) associated with new technology through a reduction in operating expense (OPEX) or to increase revenue enabled by the new technology. In the Class 7 and Class 8 commercial vehicle market, a rough rule of thumb for customer acceptance of new technology is a payback period of 24 months or less. A strong driver in the payback analysis is the annual vehicle miles travelled (VMT). Figure 1 shows a distribution of the annual VMT for a broad range of heavy-duty Class 7 and Class 8 customer applications. Figure 2 shows the more than doubling of market potential using a 24-month payback hurdle with an expected

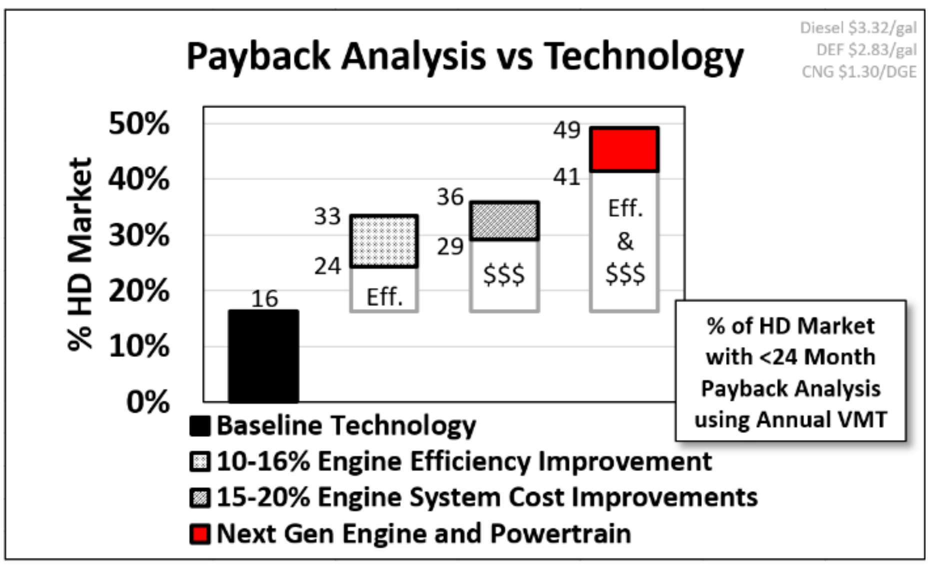
10-percent fuel economy improvement and a modest 15-percent reduction in acquisition cost relative to diesel. In this analysis, residual value has not been considered.

Figure 1: Heavy-duty Annual Vehicle Miles Travelled



Source: Cummins, Inc.

Figure 2: Payback Analysis Versus Technology



Source: Cummins, Inc.

Excluding incentives and intangibles, only very high mileage customers are finding acceptable payback periods for NG over current diesel technologies. Improvements are necessary in the cost and return levers affecting payback to further the penetration of NG in heavy-duty line haul.

This project focused on design, development, and demonstration of the critical technologies required for improving efficiencies of the next generation of heavy-duty NG engines to help increase market penetration. The goal was an improved heavy-duty NG vehicle solution as the lowest cost system solution to achieve NOx and GHG reduction targets. Many of the potential learnings should be scalable to other NG engine sizes and applications.

Current State of the Art

Current state-of-the-art medium-duty and heavy-duty NG engines are diesel engine conversions primarily based upon stoichiometric combustion and TWC. The offerings from Cummins Westport Inc. (CWI) use cooled EGR to improve efficiency and provide moderate power density. However, significant design compromises result from merely converting a diesel engine to operate on NG. These compromises result in reduced reliability (high thermal stresses), relatively low efficiency (low volumetric efficiency and compression ratio), and reduced performance (power and torque density and transient response). While lean-burn NG and dual fuel architectures are feasible options as well, in practice they prove unattractive in terms of efficiency, performance, and cost trade-off at the required emissions levels.

Next Generation Heavy-duty Engine Technology

Several public-private cooperative programs have been executed in the past to improve SI engine fuel economy. Hence, the roadmap for how to improve fuel economy of stoichiometric engines is reasonably well understood for mid-bore engines at 0.2 grams per break horsepower per hour (g/bhp-hr) NOx tailpipe emissions. This project aims to demonstrate the scalability of these learnings to drive simultaneous improvements in fuel efficiency and cost while achieving ultra-low NOx.

Engine technologies like those used in the U.S. Department of Energy's SuperTruck and SuperTruck II programs are considered, but with design and optimization being driven exclusively for the SI stoichiometric engine topology. Engine changes have entitlement capability in combustion cycle efficiency, in air handling management, and in optimized parasitic and frictional losses. Engine technology improvements alone (specifically, combustion cycle efficiency, air handling management, optimized parasitic and friction losses) are expected to improve powertrain efficiency beyond 10 percent. In addition, alignment of global common base engine designs for volume and scale drives advantages in lower costs of the powertrain.

Objective and Goals

The main objective of the project was to reach an improvement in efficiency like that of conventionally fueled vehicles and to reduce emissions to near-zero levels. Specific objectives were to:

- Develop an NG-specific combustion system design that uses high tumble charge motion and cooled EGR that builds upon a proven high cylinder pressure capable heavy-duty base engine platform in the 12–15-L displacement range.
- Demonstrate cycle average brake thermal efficiency (BTE) of 38–40 percent (>10 percent improvement over commercially available NG products on the ramped mode cycle supplemental emissions test [RMCSET]).

- Demonstrate peak BTE of 41–43 percent (>10 percent improvement over commercially available products).
- Maintain 0.02 g/bhp-hr NOx capability.
- Demonstrate a diesel-like torque curve rating of 450-500 bhp and 2100-2500Newton meter (Nm) peak torque.
- Develop an engine integrated on a global platform to enable up to a 20-percent system cost reduction.
- Confirm readiness for a TRL 6 demonstration with a prototype system.

CHAPTER 2: Project Approach

Engine Architecture Selection

Base Architecture/Efficiency Considerations

Today's NG engines include conversions of existing gasoline or diesel engines with modest incremental architecture changes. The market potential for NG suggests that a fundamental assessment of architecture options for heavy-duty applications is needed to establish the roadmap that puts the industry on a path to deliver the highest possible efficiency combined with a cost structure that provides for a high level of market penetration. The appropriate architecture selection must be made in the context of expected market introduction timing, product platform life, and anticipated market and regulatory drivers. In the case of a heavy-duty NG engine for North American applications, expected market introduction timing will be post-2024 with a minimum product platform life (without major required changes) of 10 years. In this context, the expected architecture selection drivers are:

- Efficiency or fuel cost/mile
- Engine system acquisition cost
- GHG emissions: CO_2 and methane
- Criteria pollutants, specifically tailpipe NOx capable of 0.02 g/bhp-hr
- Diesel-like performance: power, torque, and transient response

Given these drivers, general statements can be made for each of the following potential architectures:

- SI Lean Burn:
  - Excessive cost and complexity of delivering diesel-like performance
  - High aftertreatment system cost and complexity
- Dual Fuel:
  - High engine and vehicle costs associated with multiple fuel systems
  - Very high aftertreatment system cost and complexity
  - Potential GHG challenge due to methane emissions
- High Pressure Direction Injection:
  - Complex and high cost system
  - Very high aftertreatment system cost and complexity
  - High efficiency potential

- SI Stoichiometric:
    - Capable of delivering diesel-like power, torque, and transient response
    - Good efficiency improvement potential with cost effective engine systems
    - Cost effective and simple aftertreatment
    - Proven on-board diagnostics strategies
    - Able to meet all anticipated future requirements with competitive efficiency

As the flame propagates in the cylinder of a spark-ignition engine, the unburned, or end gases, are compressed. Under high pressure and temperature conditions, this end gas can spontaneously ignite, resulting in a rapid and extreme pressure rise called knock, which limits the design compression ratio. Stoichiometric conditions have reduced heat capacity driving elevated exhaust temperatures and reduced efficiency. These shortcomings can be tempered by adding diluents to the charge flow via cooled EGR.

For those reasons, the SI stoichiometric, cooled EGR heavy-duty NG engine with TWC architecture is the preferred approach. Recent market introductions of advanced stoichiometric, cooled EGR medium-duty and heavy-duty NG engine systems by Cummins, Inc., Scania, Daimler, and MAN, among others, support the above assessment and conclusion.

Engine Displacement and Platform Selection

One of the primary objectives of this program was to develop an NG-specific combustion system design that uses high tumble charge motion and cooled EGR that is building upon a proven high cylinder pressure capable heavy-duty base engine platform in the 12–15-L displacement range. The selection of engine displacement and platform for the final engine demonstration would be based on design feasibility, reliability, and meeting program requirements for performance, cost, efficiency, and emissions. The engines currently in production that were considered as a base platform are shown in Table 1.

Table 1: Heavy-duty Engine Displacements and Platforms

Engine/ Platform	Displacement (Liters)	Fuel	Advantages	Disadvantages
ISX12N	11.9	NG	- Current production NG engine	- NG-only platform, resulting in high base engine cost due to low volumes. - Limited ability to achieve desired torque/power due to lower displacement.
X12	11.8	Diesel	- Common platform for diesel and NG to drive down cost with increased commonality.	- Limited ability to achieve desired torque/power due to lower displacement.

Engine/ Platform	Displacement (Liters)	Fuel	Advantages	Disadvantages
X15	14.9	Diesel	- Highest displacement on-highway; heavy-duty base engine providing potential for diesel-like power output. - High volume base platform, providing cost saving advantages.	- Low maturity/ experience. First 15-L NG engines will be launched in 2024. - Higher mechanical losses, especially those due to power cylinder friction.

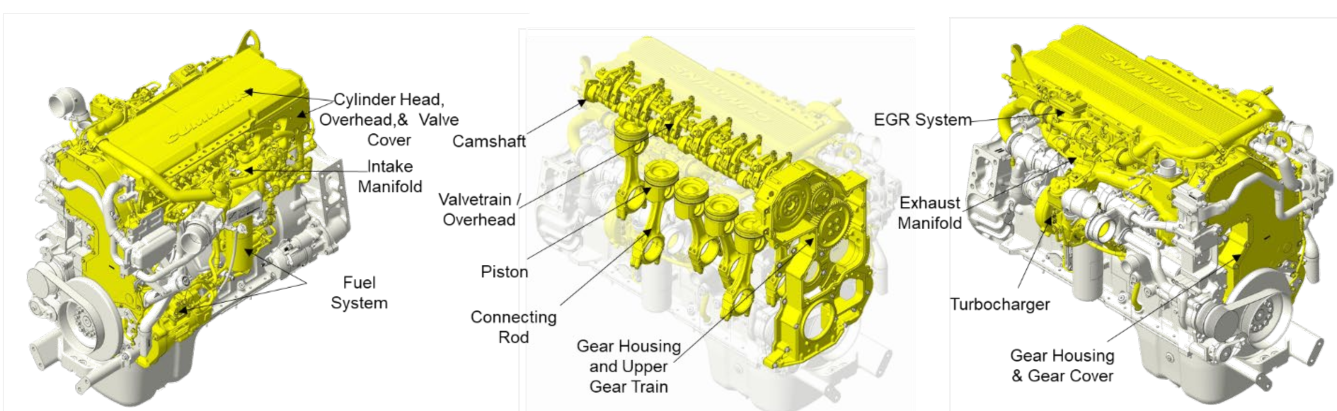
Source: Cummins, Inc.

The X15 diesel platform has a cost advantage over the current X12 diesel and NG platforms, primarily from economics of scale. The lower base engine cost provides a significant system cost reduction benefit. The larger displacement is also more favorable to achieve the desired torque curve with high efficiency.

A significant number of base engine components from the X15 diesel engine can be carried over to the heavy-duty NG engine to achieve lower engine cost by maintaining commonality. The parts highlighted in yellow in Figure 3 represent the unique components that would need to be designed for the heavy-duty NG engine to meet performance and efficiency targets.

The X15 engine also has the design maturity and reliability base engine hardware that is well suited for testing and demonstration of the heavy-duty NG engine. The high volume X15 engine has already been engineered into most North American heavy-duty chassis and has a proliferation of options such as flywheel housings and engine mounts.

Figure 3: NG-specific Unique Components

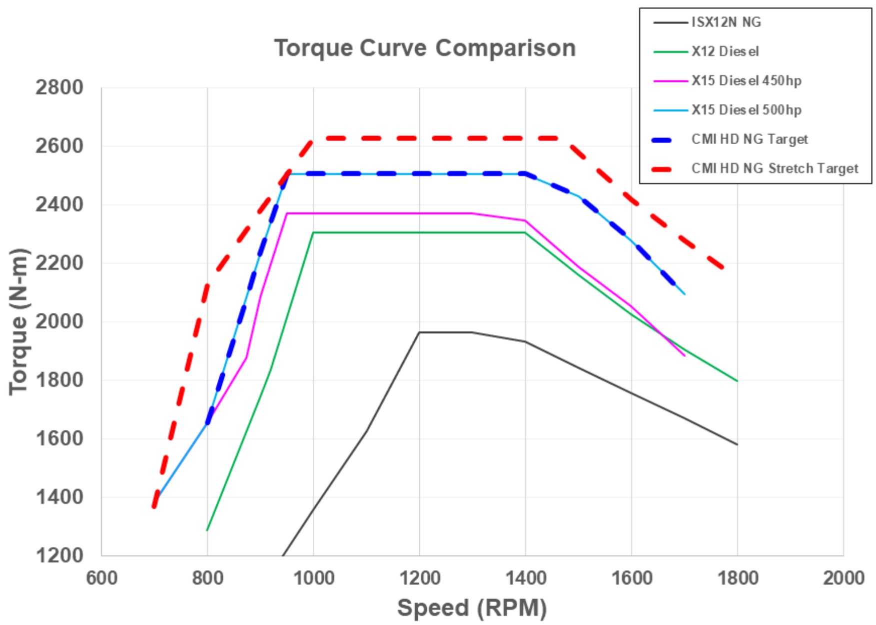


Source: Cummins, Inc.

The target rating for this program is 500-bhp rated power and 2500-Nm peak torque. Figure 4 compares the torque curves for various diesel and NG engines currently in production. Figure 5 shows the same comparison in the brake mean effective pressure (BMEP) space to compare the power density. The target torque curve for final demonstration is set to be the same as the highest volume (by annual sales) of the current production X15 engine with a peak torque of

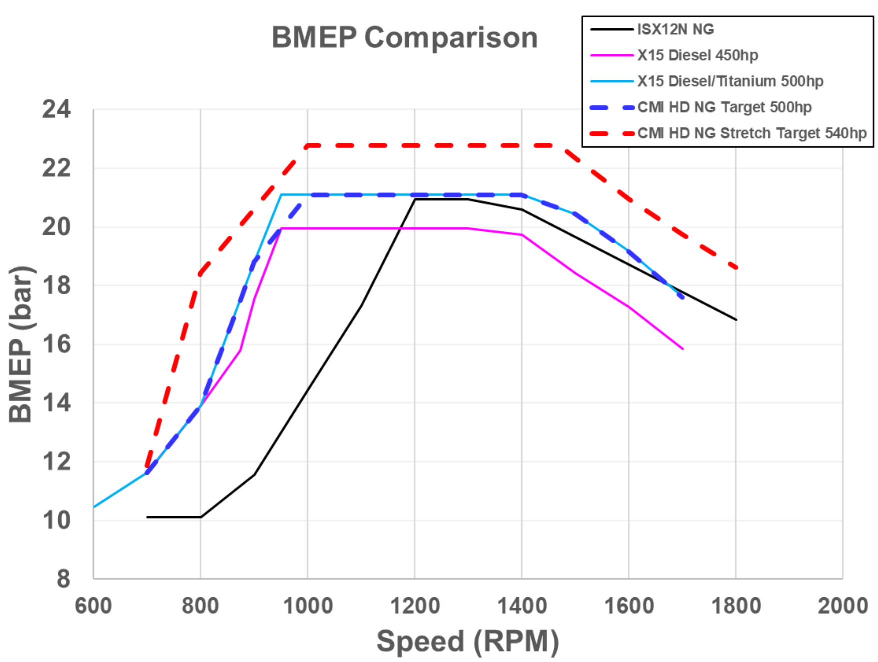
2500 Nm and rated power of 500 bhp to meet the demands of most applications that currently use the X15 diesel engine. The program team also selected rated power of 540 bhp and peak torque of 2630 Nm as a stretch target to design protect the combustion system for higher power density. The air handling solution would be different between the minimum target torque curve and the stretch targets to achieve the best efficiency for the two power density options without a compromise.

Figure 4: Torque Curve Comparison



Source: Cummins, Inc.

Figure 5: Brake Mean Effective Pressure Comparison



Source: Cummins, Inc.

Overall Design Approach

The design improvements took a three-pronged approach based on their contributions to overall efficiency as laid out and illustrated in technology architecture.

1. Mechanical Efficiency – components that can be optimized or designed to reduce frictional losses of the engine and improve mechanical efficiency.

   In addition to friction mean effective pressure (FMEP) reduction from in-cylinder and parasitic losses of the base X15 engine when compared to the ISX12N engine, some other friction reduction technologies were explored in this project for further BTE improvement. The reduction in friction results in improved mechanical efficiency and, hence, improved BTE. Some of the key technologies investigated included, but were not limited to, low compression height pistons with a longer connecting rod, torque plating honing, and low tension piston rings.

2. Closed Cycle Efficiency – combustion system optimization, including required tumble ratio, piston bowl shape, and compression ratio optimization.

   The objective was to develop and demonstrate a pent roof cylinder head that uses tumble charge motion to result in a highly optimized combustion system with high closed cycle efficiency with reduced combustion face temperature. Several intake port designs were analyzed to select the best ones for procurement and evaluation on the single cylinder engine (SCE). Trade-offs between tumble ratio and discharge coefficient were studied. It was important to determine the minimum tumble ratio required for optimum combustion to minimize the flow losses that result in lower volumetric efficiency.

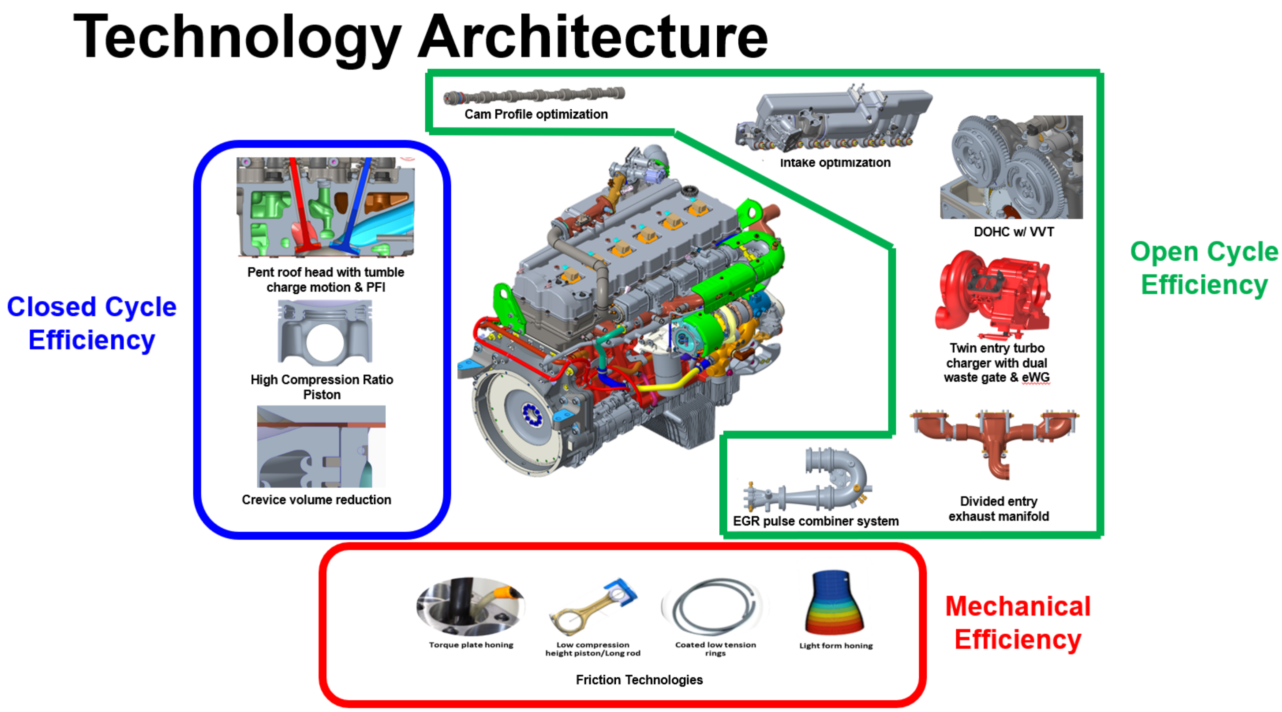
   Different piston bowls with compression ratio ranging from 12:1 to 16:1 were considered using combustion computational fluid dynamics (CFD) analysis and were tested on the single and multi-cylinder engine to estimate and verify improvements in combustion efficiency, heat release rates resulting in maximum BTE improvement. Combustion CFD studies as well as past experimental results on NG engines show an approximate 0.5-percent peak BTE improvement for every 1-point increase in the compression ratio. The benefit to the RMCSET cycle averaged BTE is more complicated to predict due to interaction with air handling and other factors, but it generally trends with peak BTE.

   Further BTE improvements are enabled with reduced top land clearance and reduced crevice volume. A reduction in crevice volume change results in a reduction of a trapped mass of hydrocarbons in the top land crevice region, thereby resulting in lesser deposits and improved combustion efficiency with a reduced knock index. This also reduces the engine out methane emissions, thereby reducing system out methane emissions.

3. Open Cycle Efficiency – optimization of air handling components to meet power and corresponding air flow requirements with the least pumping loop possible.

The components that primarily influence open cycle efficiency (OCE) are the cylinder head, including the ports and valves, cam profiles and valve overlap, turbocharger matching, and the EGR system design. These are collectively called air handling system components. Figure 6 includes the technologies and architecture choices for improving OCE of the engine.

Figure 6: Key Technology and Architecture Options



Source: Cummins, Inc.

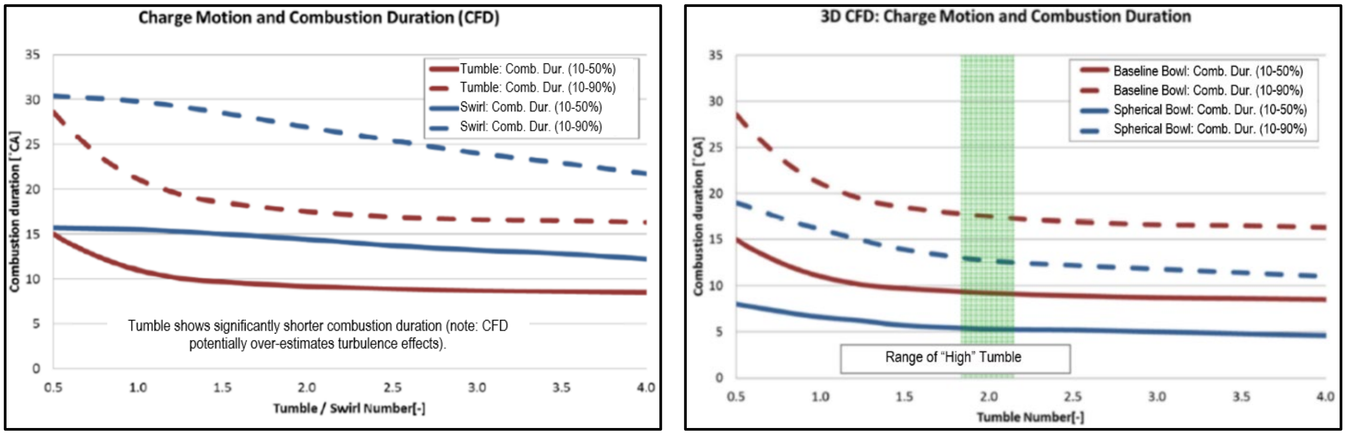
System Design Approach

Combustion System

Diesel-derived NG engines employ a swirl-based combustion system, typically with a reentrant piston bowl designed to achieve the needed compression ratio reduction and accelerate charge motion via squish mechanisms. Tumble-based combustion systems, employed in most light-duty SI engines, offer some intriguing benefits that could transfer to NG engines. CFD simulations for an NG-fueled combustion system indicate that tumble charge motion may also benefit combustion duration. A shorter combustion duration improves EGR tolerance, facilitating higher EGR rates, enhanced resistance to engines' tendency to knock, and greater engine efficiency.

The left side of Figure 7 shows that a tumble-based system can improve combustion duration for the conditions simulated, even at moderate tumble levels. The right side shows the additional benefits that optimizing the piston bowl shape can deliver by comparing a current state-of-the-art bowl shape against a proposed spherical bowl shape. The simulations were performed for a medium-bore engine. Analysis was extended to a large-bore engine under this program. With this program, the compression ratio, charge motion, and piston bowl geometry were optimized for the best heat release rate, combustion stability, high torque, robustness to wide range of fuel composition, and closed cycle efficiency.

Figure 7: Effects of Charge Motion – Simulation Results (2000 rpm/12 bar BMEP)

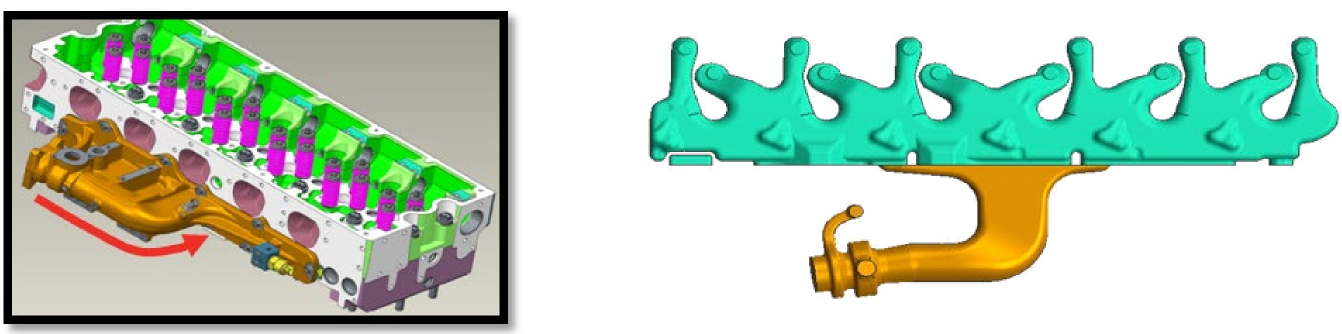


Source: Cummins, Inc.

Cylinder Head and Breathing

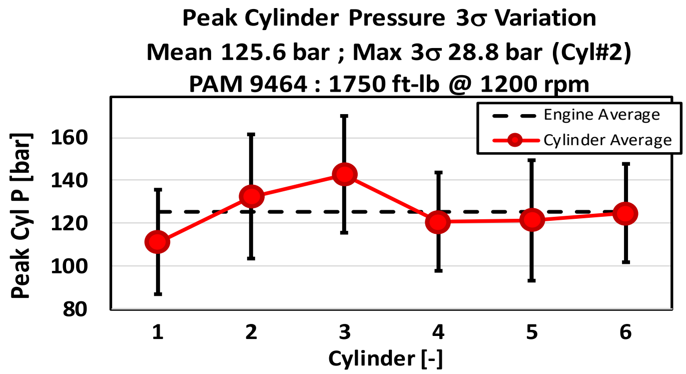
Key focus areas to improve OCE include cylinder-cylinder and cycle-cycle variation reduction, volumetric efficiency improvements, and reduction of in-cylinder residuals (in addition to the previously discussed charge motion). Today's NG engines use a diesel-derived cylinder head and integrated intake plenum such as those shown in Figure 8. This design, while satisfactory for diesel engines, results in flow distribution that is imbalanced and leads to significant peak cylinder pressure (PCP) variation from cylinder to cylinder (Figure 9). This results in compromised efficiency in an NG engine.

Figure 8: Typical Cummins Diesel Cylinder Head and Intake Manifold



Source: Cummins, Inc

Figure 9: Cylinder-to-cylinder Peak Pressure Variation



Source: Cummins, Inc.

The intake plenum, designed for swirl generation, results in relatively high loss factors and less than desirable volumetric efficiency. The intake manifold was designed for optimal charge mixing of air, fuel, and EGR and balanced flow distribution to all cylinders.

Cooling is another area for improvement, for both volumetric efficiency and knock mitigation. Current diesel-derived heavy-duty NG cylinder heads run relatively hot. Given the lower PCP target for NG of 180 bar, compared to current targets of 220 bar and higher for diesel engines, a thinner combustion deck can be employed. This allows the cooling jacket to be located closer to the cylinder head face, which will lower operating temperatures significantly. Achieving cooler combustion face temperatures allows for increased compression ratio while avoiding engine knock, which has a direct and positive impact on engine efficiency. With this approach, improved cooling of the combustion deck was developed.

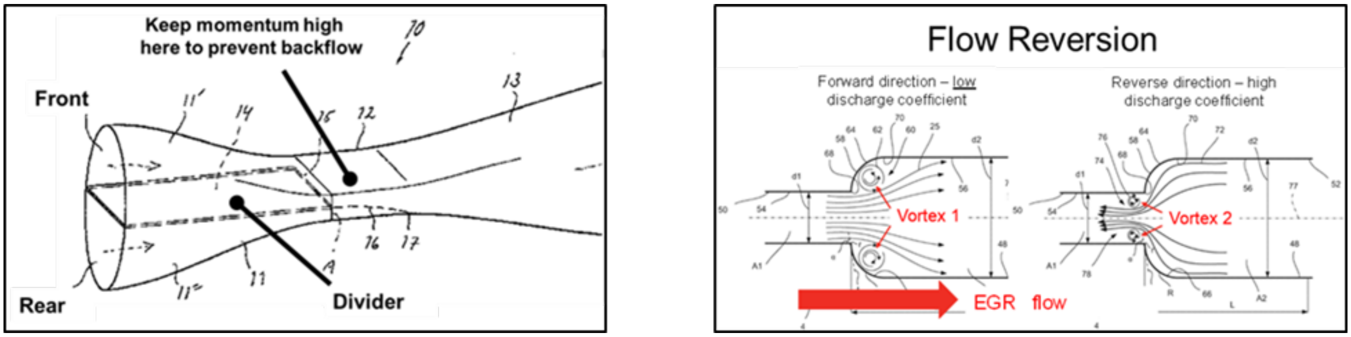
A significant challenge was to achieve high levels of EGR flow up to 25 percent without raising exhaust manifold pressure too high. High exhaust manifold pressure results in high cylinder residuals, leading to greater knock tendency, which constrains the compression ratio and thus engine efficiency. The exhaust system was designed for minimum and evenly distributed flow losses.

A twin entry divided turbine housing was developed and then integrated with a split exhaust manifold to minimize the interaction between cylinders. This design allows for maximum exhaust pulse utilization without driving high levels of residuals that result in a combustion phasing penalty. The twin entry turbocharger was sized appropriately for this engine, including dual wastegate capability with moderate temperature capable components to allow the engine to breathe evenly across the cylinders.

A high-pressure loop EGR approach utilizes exhaust pressure pulse energy to drive EGR. Current estimates are that a 15–20 percent EGR rate can be achieved while average exhaust and intake manifold pressures are equal. The EGR system draws from the split exhaust manifold to avoid crosstalk between cylinders. In addition, a pulse converter and flow anti-reversion device was designed and located in the EGR loop (Figure 10). The pulse converter's primary function is to combine the pulse flow from the two sides of the exhaust manifold, minimizing interference and always maintaining forward momentum. To avoid flow reversion and flow momentum loss, a reed valve or similar device can be installed in the EGR system.

The flow reversion device shown in Figure 10 is the preferred solution, as it is an entirely passive device. By reducing the reverse flow, the required net flow can be achieved at a lower forward flow. This reduces pumping losses and lowers residuals, which benefit engine efficiency.

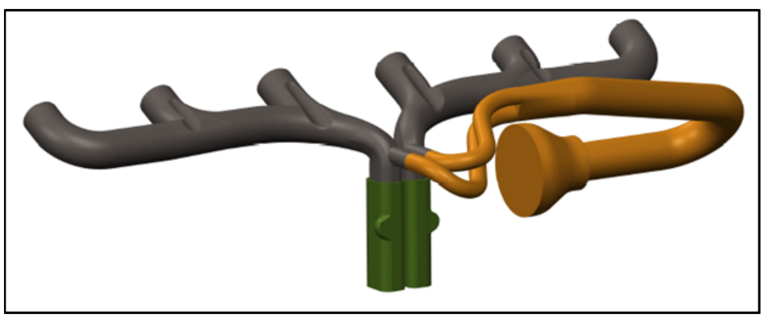
Figure 10: EGR Flow Pulse Converter (Left) and Anti-reversion Device (Right)



Source: Cummins, Inc.

Figure 11 shows a conceptual layout of a reduced-pressure EGR system. The individual elements of this concept are not new, but their combined application in an EGR system is novel, with significant expected benefits for a heavy-duty NG engine. A novel high-pressure loop EGR system is developed under this project.

Figure 11: Conceptual Reduced Pressure, High Pressure Loop EGR System



Source: Cummins, Inc.

Valvetrain Development

Variable valve actuation (VVA) has long been applied in light-duty SI applications to improve performance, emissions, and fuel economy. Engine efficiency can be increased through several means. VVA enables Miller cycling (late intake valve closing), which allows an increase in geometric compression ratio, resulting in higher efficiency. A high compression ratio at high engine load normally results in engine knock, but under these conditions the intake valve can be closed later, resulting in a reduced effective compression ratio. The primary efficiency improvement therefore comes under part-load conditions.

VVA also enables varying volumetric efficiency up to and including cylinder deactivation. This can be applied under cruise and other part-load conditions, resulting in less throttling of the engine, which reduces pumping losses and thus improves efficiency. Additionally, VVA allows for early exhaust valve opening, which can improve low-speed torque and transient response and help drive EGR. Thermal optimization opportunities could be exploited via management of heat transfer to the power cylinder during warmup and variable exhaust expansion ratios.

Aftertreatment

Cummins has invested substantially in developing aftertreatment systems for stoichiometric NG engines utilizing TWC models to create cutting-edge formulations. These models not only capture the key dynamic performance of TWCs but also predict the by-products, such as ammonia and nitrous oxide. Platinum group metal loading has also been incorporated into the models. The knowledge from the extensively researched and developed computational models of Cummins led to the release of the first near-zero (0.02 g/bhp-hr NO_X) NG engine product in 2016. This capability was applied to the operating conditions of a large-bore HD engine to optimize performance. The first 2024 X15N engines will utilize aftertreatment systems like the current production ISX12N. The same systems were utilized for the engine demonstration in this project. Engine out emissions data is being used by other low-cost aftertreatment projects within Cummins, with plans for launch in the next 2-3 years.

Ignition System

The ignition system serves as a critical subsystem for high efficiency NG engines. The ignition systems in widespread use today include conventional inductive systems, capacitive discharge systems, and hybrids of each. Inductive systems are well developed, inexpensive, and reliable. Capacitive discharge systems offer slightly more ignition energy at a substantial cost premium. The research literature indicates that charge dilution is a path toward higher efficiency. However, engines are fundamentally limited by the ignition energy available to tolerate air or EGR dilution. A high energy inductive ignition coil can provide the needed ignition energy for combustion stability at a lower cost when in high production. Improved thermal management of the spark plug temperature via a cylinder head cooling design and spark plugs with larger electrodes also has the capability of extending the service interval at a slight incremental cost.

Prototype ignition systems like multi-strike and dual coil offset high energy inductive ignition systems, which offer increased ignition energy levels in support of higher EGR fractions and lower production cost, are potential options for the project, along with alternative spark plug designs.

Control System

Engine controls are critical to use the sensory feedback in an optimal manner, ensuring robust efficiency, performance, and emissions over the life of the engine system and components. Key subsystems that the control system needs to manage include the air handling system (including turbocharging systems) to ensure the proper amount of charge and EGR composition, the fuel injection system, and the spark systems to ensure efficient combustion as well as the proper exhaust properties that enable the highest performance from the TWC.

The optimized design and development of the control system will ensure that:

- System level performance requirements are robustly met over a wide range of operating conditions and environmental conditions.

- Proper trade-offs are made with respect to the sensing and actuation systems to ensure that the system can meet the market cost and reliability requirements.

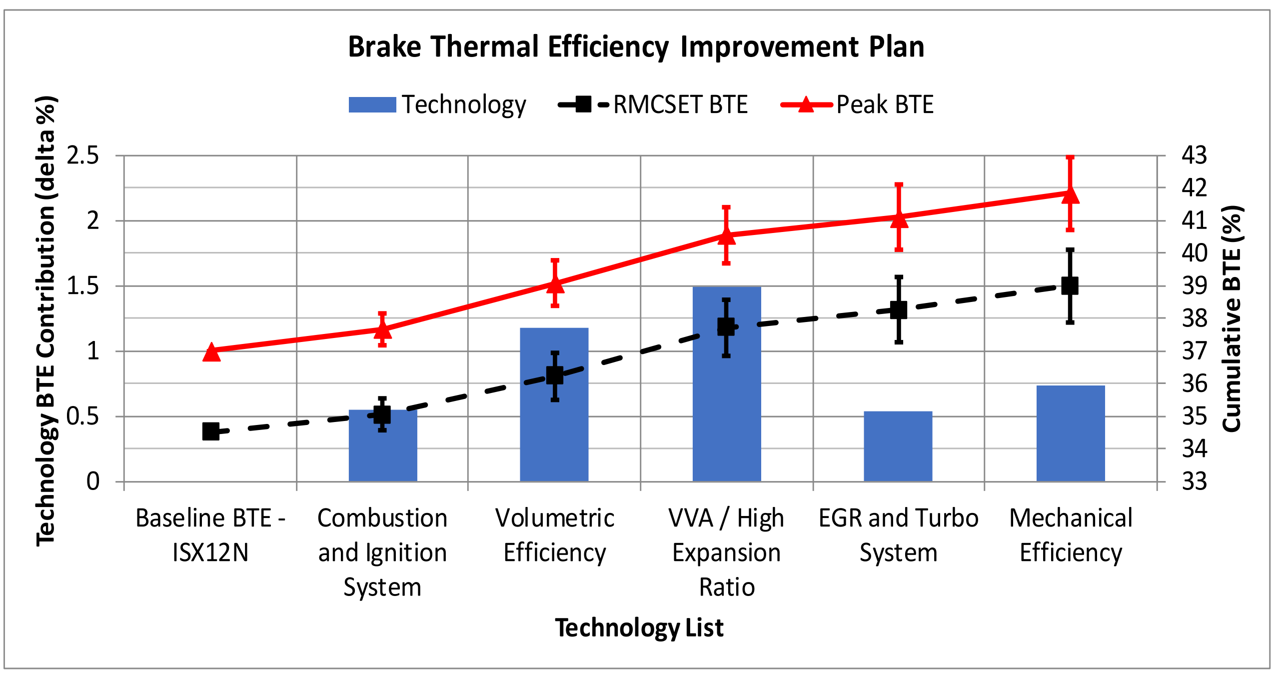
Robust emissions control requires both accuracy and precision in the lambda, or air-fuel ratio, control. To date, all CWI production NG engines are taking advantage of single a point fuel injection strategy and fixed geometry power cylinder boundaries. This project investigated the practical control system limitations of per-cylinder fueling via port fuel injection with respect to emissions control under all engine operating conditions. This approach will also reduce the overall cost of the fuel system.

Advanced air handling techniques, such as variable valve actuation drives, continued the emphasis on air flow estimation routines and/or measurements on a per-cylinder basis. The control system and sensors to support a VVA system were developed. Updates to the knock control algorithms were developed in this program to further improve system efficiency and maintain compatibility for a wide range of fuel compositions. The benefits of the control system changes through use of a multi-cylinder engine with fitted aftertreatment were quantified.

Planned Path to Target

Figure 12 provides a summary of the combined 10–16-percent improvement in efficiency expected compared to a commercially available, state-of-the-art, heavy-duty NG engine, the Cummins Westport ISX12N. The blue bars indicate the individual benefit expected from the aggregate technology improvement in percent BTE contribution on the left-hand axis. The black and red lines indicate the cumulative RMCSET cycle average and peak BTE projections, respectively, on the right-hand axis.

Figure 12: Cycle Average and Peak BTE Improvement Plan



Source: Cummins, Inc.

CHAPTER 3: Results

Single Cylinder Engine Design and Testing

The workflow for the design and analysis of the combustion chamber for developing and demonstrating closed cycle efficiency on an experimental SCE consists of hardware design, three-dimensional (3D) flow analysis, and 3D combustion CFD analysis for evaluating port flow quality, as well as combustion characteristics. The following unique components were designed for the combustion system that utilizes tumble charge motion:

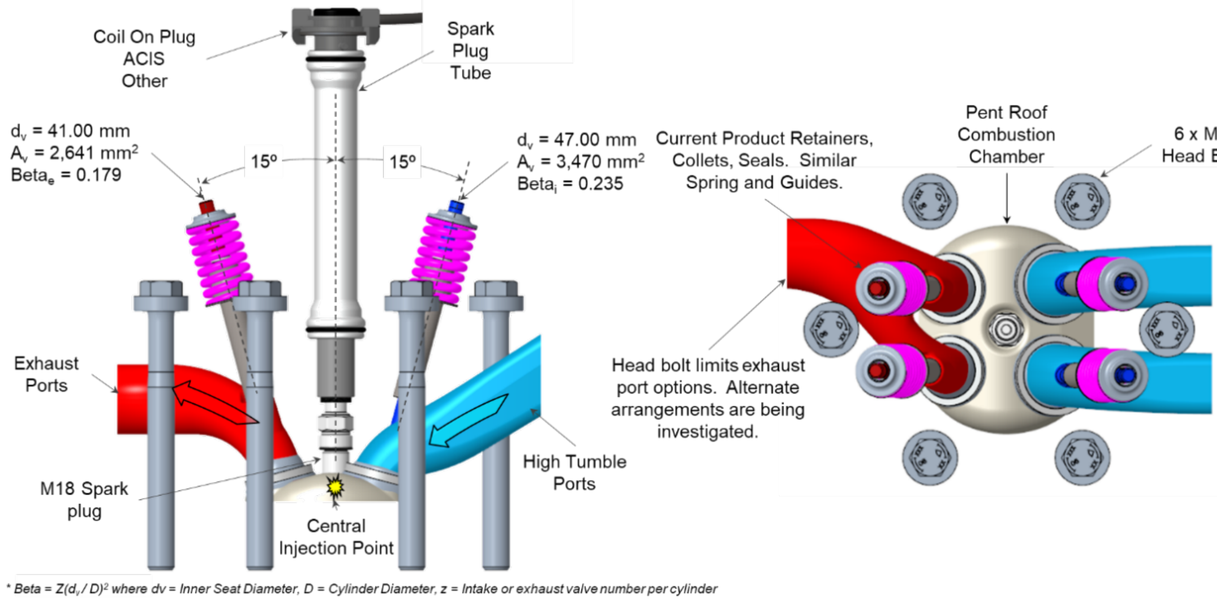
- Tumble charge motion intake ports
- Piston bowl shape and compression ratio
- Valve size and shape
- Cylinder head that incorporates the designed ports, valves within the constraints of coolant flow passages, and a head bolt pattern from the purpose-built SCE engine block

The combustion chamber design for the experimental SCE was built upon the bottom end of a Cummins X15 Diesel engine – Cummins' largest heavy-duty engine – with a bore size of 137 millimeters (mm). The lower gear train is maintained the same as the base diesel engine, while the upper gear train is redesigned to accommodate dual overhead camshafts. The block design features are largely the same as the base X15 engine. The crankshaft, connecting rod, bearings, piston cooling nozzles, piston pin, liner, and piston rings of the base X15 diesel engine are utilized for the experimental SCE. Pistons have a similar skirt and pin bore as the diesel engine piston with a bowl shape that is suited for NG combustion.

The combustion chamber uses a four-valve, pent roof design with the intake and exhaust valves inclined 15° from the center to achieve a reasonable flow area while keeping the cylinder head volume low for the compression ratio (CR) chosen. The cylinder head with the combustion chamber, intake and exhaust ports, and a centrally mounted spark plug is shown in Figure 13.

A flat top piston results in a CR of 15.3:1, and a target CR of 12:1 (similar to that of other NG engines) is achieved by incorporating an elliptical bowl. Flat top pistons, which allow for flexibility in testing multiple CRs by machining different bowl depths and shapes, was procured to support modeling to determine the ideal CR and piston shape.

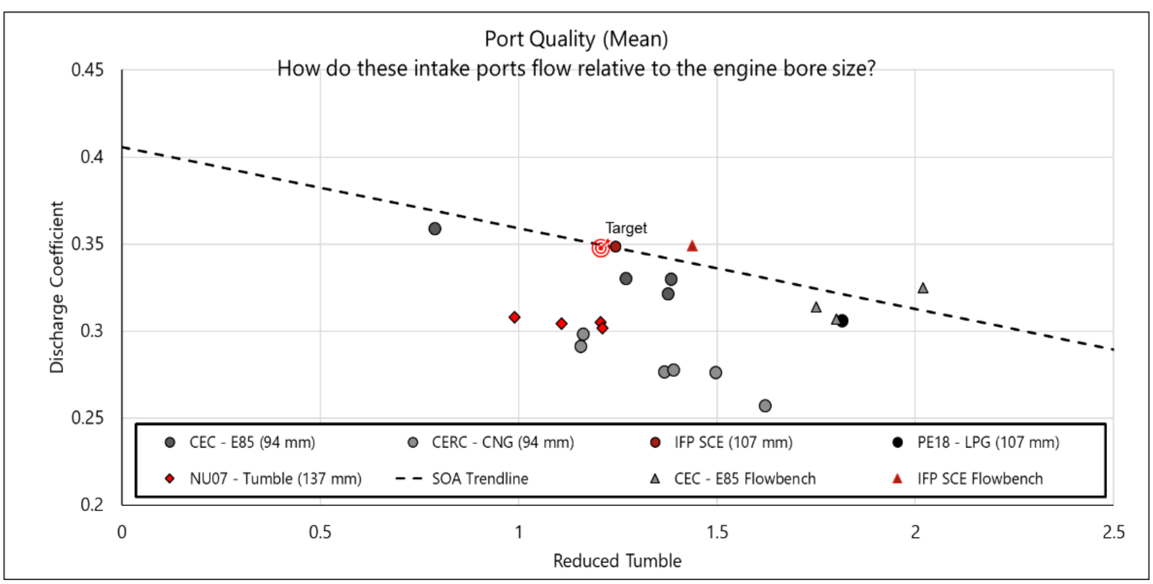
Figure 13: Single Cylinder Engine Combustion Chamber and Cylinder Head Layout



Source: Cummins, Inc.

Four different tumble ports were analyzed. The flow quality of tumble ports is evaluated by examining the trade-off between reduced tumble (tumble normalized by bore size) and discharge coefficient. While the initial tumble port designs are in the desired neighborhood of a tumble ratio as shown in Figure 14, the flow losses could be further reduced to achieve the target discharge coefficient at similar tumble ratios. The state of the art line shown in Figure 14 is based on ports that were designed for smaller displacement engines for past programs. A cylinder head with the rev03 NU07 tumble ports was procured for initial combustion chamber development testing on the SCE while further optimization work is done to improve port flow quality.

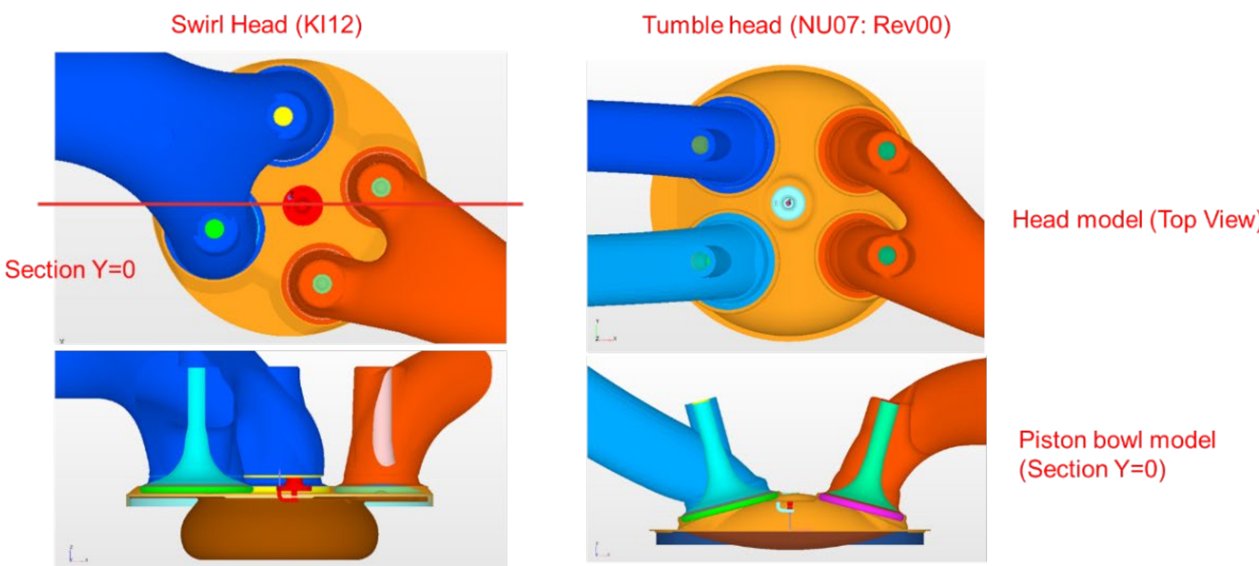
Figure 14: Port Flow Quality for Different Tumble Ports



Source: Cummins, Inc.

Converge CFD modeling is used for predicting combustion characteristics like cylinder pressure, ignition delay, burn duration, and heat release. Combustion CFD modeling is performed at the rated power condition (1800 rpm, 15 bar BMEP) to compare different combustion chamber designs. An experimentally validated combustion model for a combustion system with swirl charge motion and the same bore size (KI12) is used as the baseline for defining boundary conditions for this model. The combustion chamber designs for the baseline swirl and the rev00 tumble systems are shown in Figure 15.

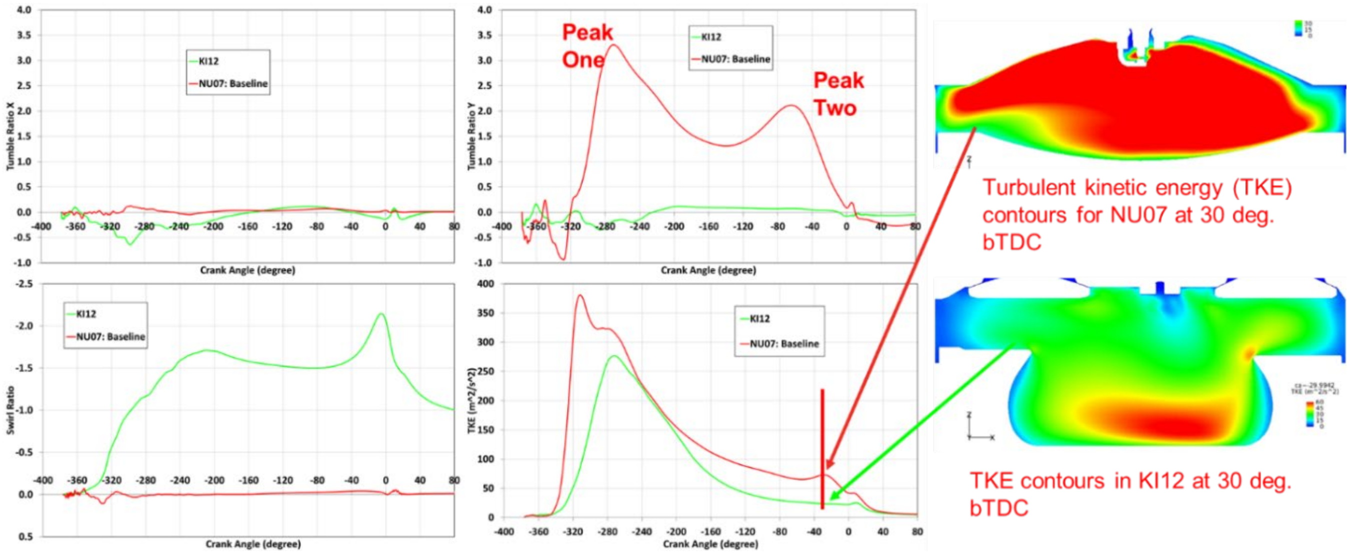
Figure 15: Swirl Versus Tumble Charge Motion-based Combustion Systems



Source: Cummins, Inc.

Turbulent kinetic energy (TKE) at different crank angle (CA) degrees before top dead center (bTDC) is a useful initial measure that strongly influences and helps predict the combustion characteristics of different combustion systems. Comparison of the swirl charge motion and the rev00 tumble charge motion-based combustion systems indicates a significantly higher TKE at 30 degrees CA bTDC for the tumble-based combustion system, as shown in Figure 16. The second tumble peak, which is closer to TDC and spark timing, is the more critical parameter for determining combustion behavior. It is critical to preserve the tumble after the second peak as much as possible to ensure a short ignition delay and fast combustion.

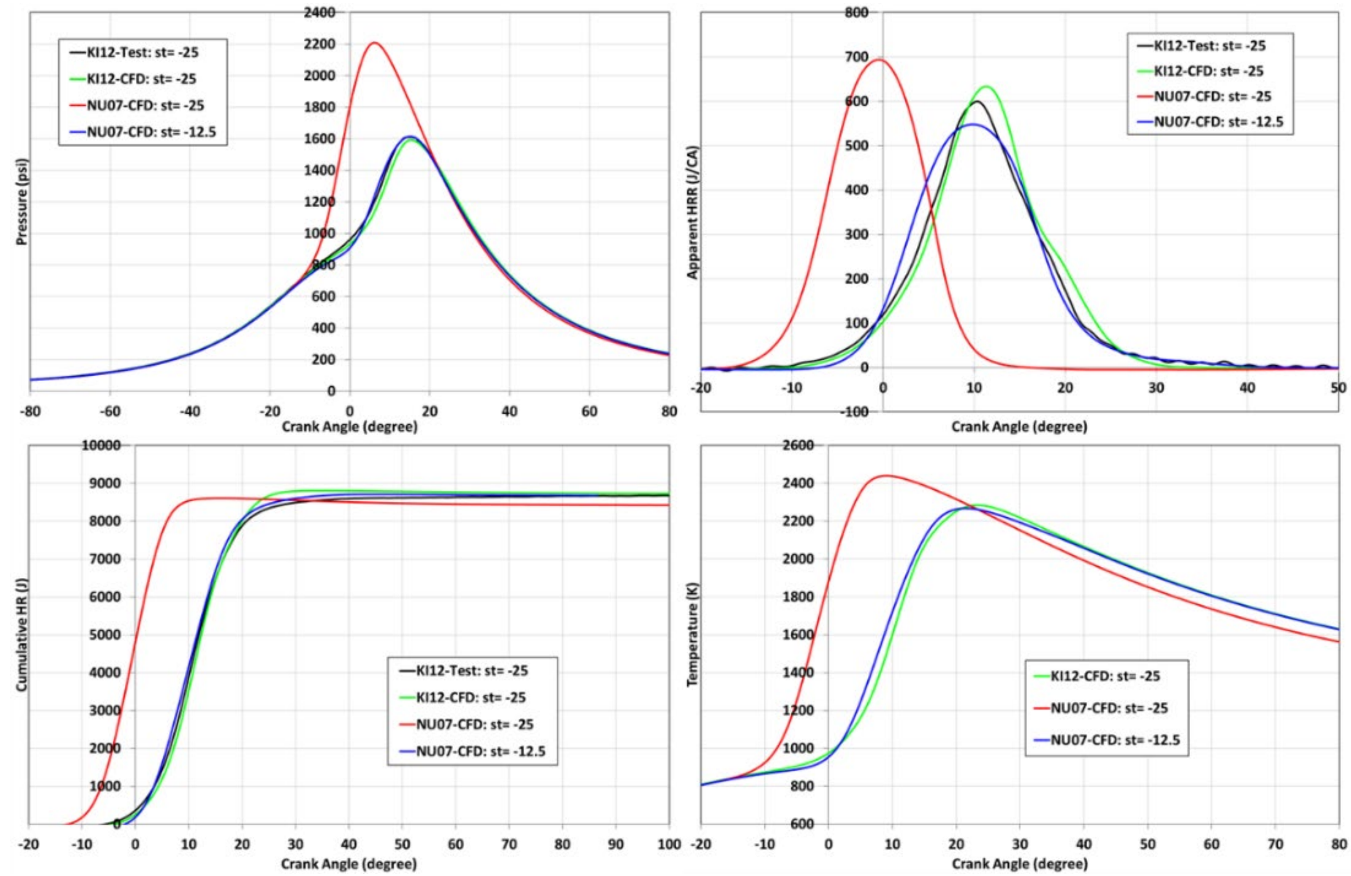
Figure 16: Tumble and Swirl Ratios, TKE for Swirl and Tumble-based Combustion Systems



Source: Cummins, Inc.

The higher TKE of the tumble-based combustion system results in a significantly shorter ignition delay, higher PCP, and apparent heat release rate than the swirl-based combustion system for the same spark timing, as shown in Figure 17. Spark timing had to be retarded by approximately (~)12.5 degrees to maintain the same crank angle, corresponding to 50 percent mass fraction burned (CA50), between the two combustion systems. The initial combustion CFD analysis with the rev00 ports was used for arriving at the tumble ratio targets for further iterations of the tumble ports. The port designs were iterated to improve the port flow quality while preserving the tumble ratio.

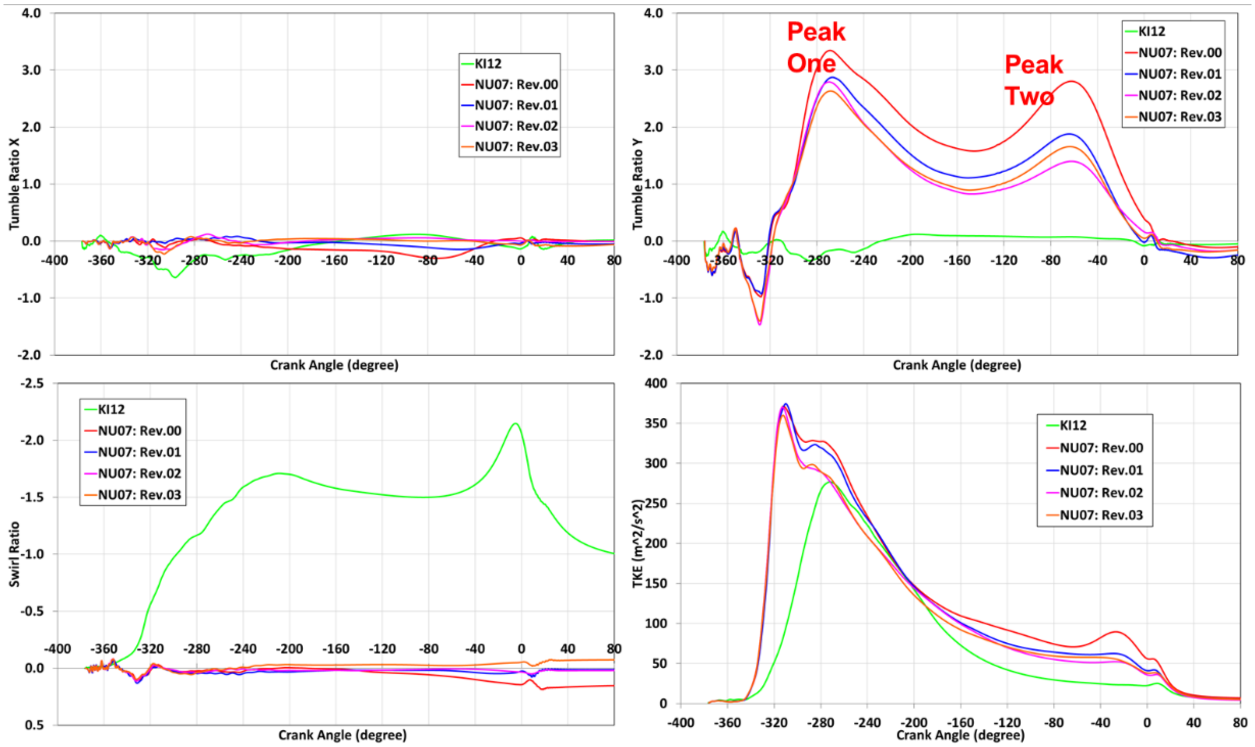
Figure 17: Combustion Characteristics of Initial Swirl and Tumble-based Combustion Systems



Source: Cummins, Inc.

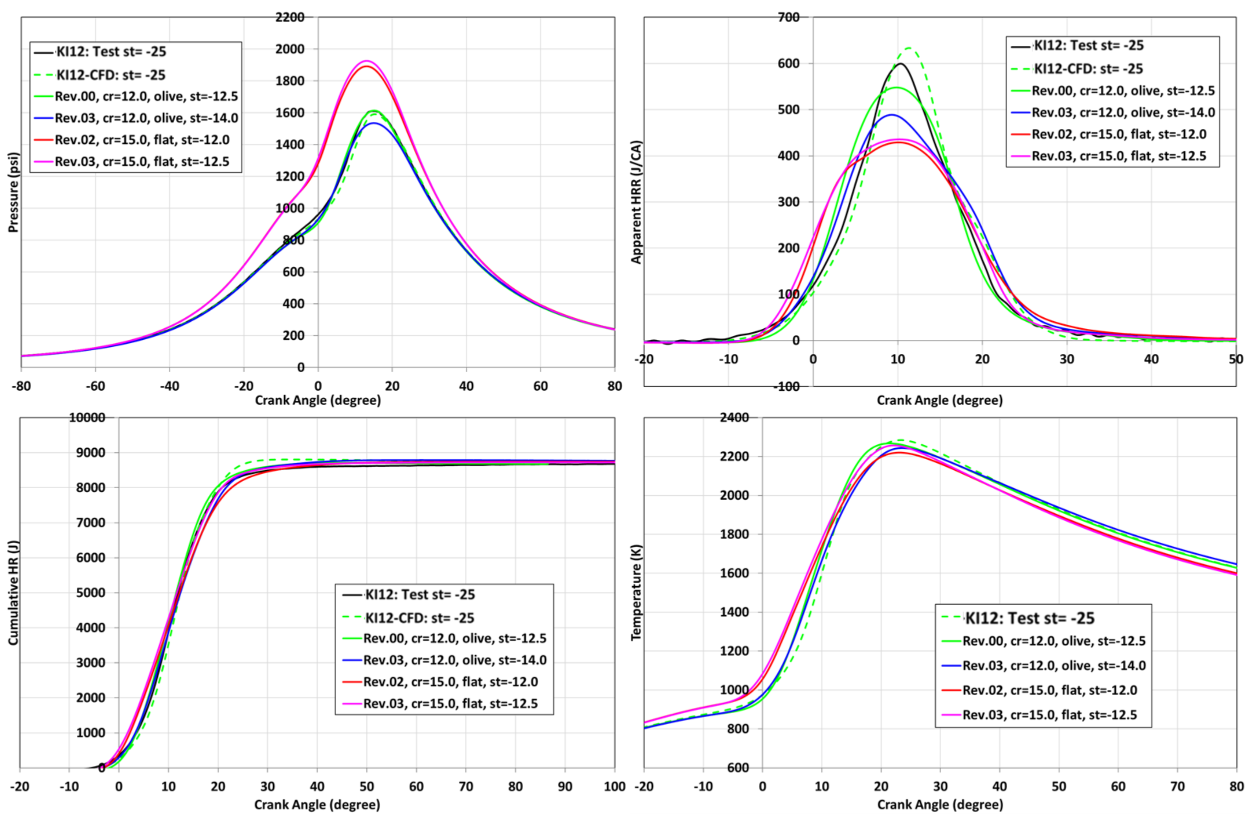
The successive tumble port iterations aimed at improving port flow quality were also analyzed to ensure that the combustion characteristics were maintained within desired limits. Figure 18 shows the tumble ratios and the TKE for the different port iterations. The rev03 ports currently offer the best trade-off between the discharge coefficient and the tumble with higher TKE close to TDC. Combustion CFD studies were completed for the different tumble port designs with a flat top piston at 15:1 CR and an elliptical bowl at 12:1 CR, as shown in Figure 19. Spark timing was adjusted to compare the different combustion chambers at a constant CA50 of 11 degrees after top dead center.

Figure 18: Tumble Ratio and TKE for Different Tumble Port Iterations



Source: Cummins, Inc.

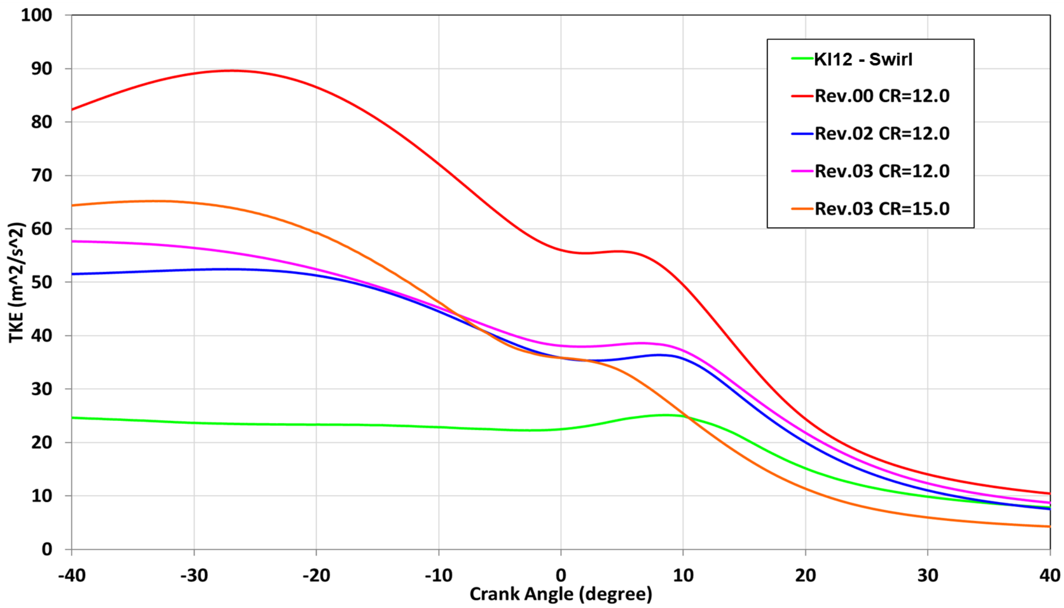
Figure 19: Combustion Performance of Different Combustion Chamber Options



Source: Cummins, Inc.

While the CR 15 piston results in a higher PCP for the same CA50, as expected, it also results in a longer combustion duration due to the flat top pistons decaying TKE faster because of destruction of the tumble motion, as shown in Figure 20.

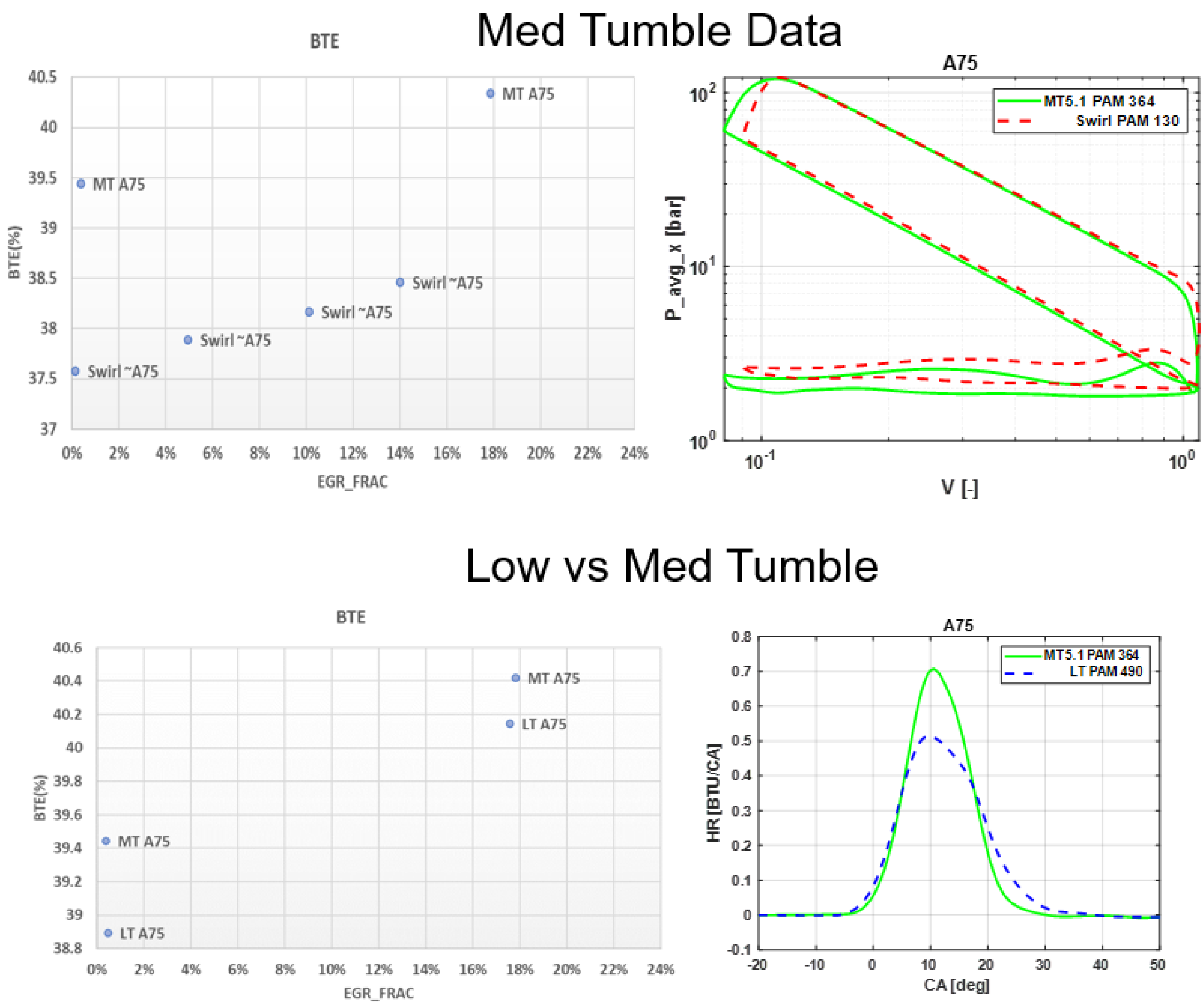
Figure 20: TKE for Different Combustion Chamber Designs



Source: Cummins, Inc.

The main objective of the single cylinder engine testing is the experimental testing of these combustion system hardware designs shortlisted from flow modeling and analysis. Data was collected at the A75 point on the baseline swirl and tumble cylinder head designs. The results are shown in Figure 21. Overall, very high EGR tolerance is being observed, with no gross misfire through 30 percent EGR. No evidence of ignitability challenges is observed. Overall data indicates efficiency improvements with the higher tumble cylinder head design due to shorter combustion duration and more favorable closed cycle efficiency, as seen in the BTE improvement of up to 2 percentage points compared to the swirl cylinder head. Shorter combustion duration is evident on the medium tumble design compared to the low tumble design, with slight directional improvement in BTE.

Figure 21: Single Cylinder Engine Key Test Results



Source: Cummins, Inc.

Mule Multi-cylinder Engine Design and Testing

The workflow for the air handling design and analysis for demonstrating OCE improvements on the ISX12N mule engine consisted of hardware design, GT power simulations for assessment of efficiency improvements, and Fluent/GT Power coupled CFD analysis for accurately estimating EGR flow at different operating conditions with the pulse capture EGR system.

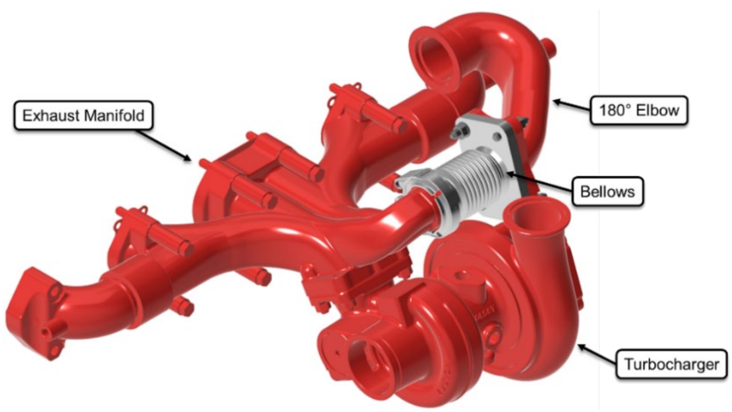
The following components were designed to demonstrate OCE improvements on the ISX12N mule engine:

- A divided exhaust manifold with EGR pull off: A divided exhaust manifold isolates the front and rear cylinders of the engine to minimize the interaction between cylinders. This design allows for maximum exhaust pulse utilization without driving high levels of residuals, which result in a combustion phasing penalty.

- An EGR pulse converter integrated in the bellows: The pulse converter's primary function is to combine the pulse flow from the two sides of the exhaust manifold, minimizing interference and always maintaining forward momentum while minimizing flow reversion and momentum loss. The pulse converter is an entirely passive device that minimizes reverse flow to meet the required net flow at a lower forward flow.
- A 180-degree elbow between the bellows and the EGR cooler.
- A twin entry turbocharger: Several different compressor and turbine options were considered. The turbocharger combination that offered the best trade-off between EGR flow capability and efficiency was chosen.

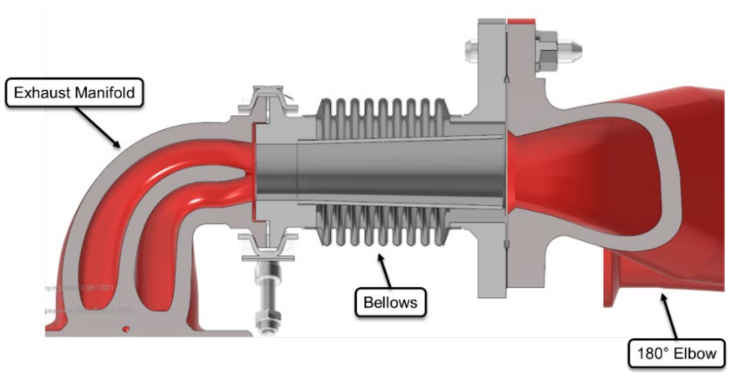
The final air handling components designed are shown in Figure 22 and Figure 23.

Figure 22: Air Handling System



Source: Cummins, Inc.

Figure 23: Pulse Capture EGR System



Source: Cummins, Inc.

The Fluent GT Power Coupled CFD analysis approach accurately captures the flow dynamics and interactions that cannot be predicted by a standalone 1D model without calibration data. The exhaust manifold and pulse capture EGR systems are initially evaluated at the peak torque operating point to evaluate the worst-case EGR flow capability of the system at the minimum pressure differential between the intake and exhaust. Once the EGR flow requirements at the lowest possible engine delta pressure at peak torque are met, the hardware would be evaluated at other operating conditions. The EGR flow data from 3D CFD simulation would also be used to calibrate the 1D model for further simulations at other operating conditions.

The compressor and turbine options considered for simulation are captured in Table 2.

Table 2: Compressor and Turbine Options

Iteration Name	Notes
ISX12N	Single entry turbine from current product ISX12N
NU-ISX12N-01	Initial recommendation for twin entry turbocharger
NU-ISX12N-02	Same compressor as NU-ISX12N-01, reduced turbine critical area
NU-ISX12N-03	Same compressor as NU-ISX12N-01, with 65-mm turbine
NU-ISX12N-04	Same compressor as NU-ISX12N-01, with less efficient 70-mm turbine

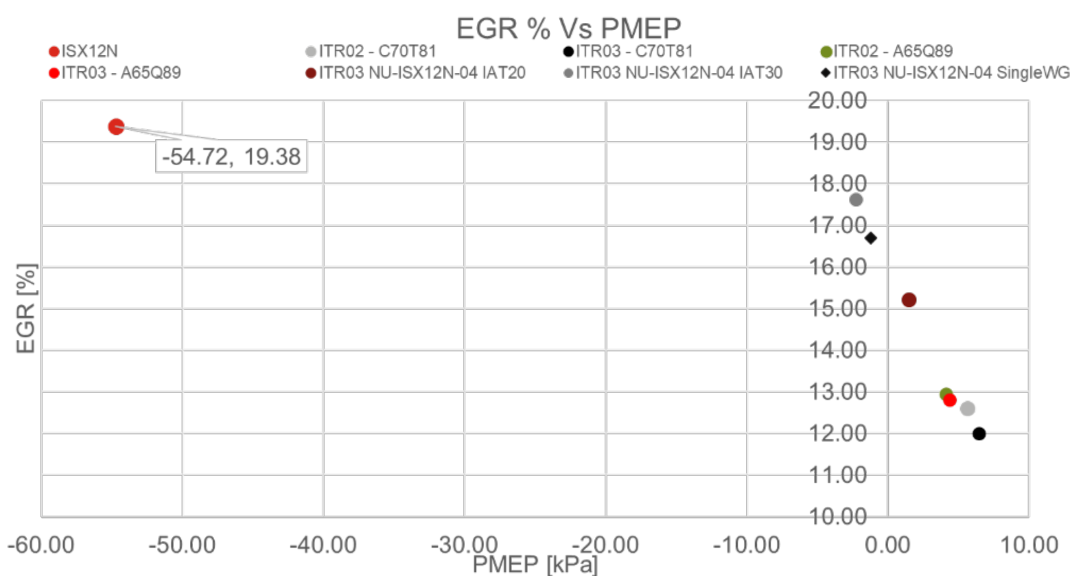
Source: Cummins, Inc.

Coupled CFD analysis was run for the following hardware combinations at peak torque:

- ITR01 exhaust manifold with NU-ISX12N-01 and NU-ISX12N-02 twin entry turbochargers
- ITR02 and ITR03 exhaust manifolds with NU-ISX12N-03 twin entry turbocharger
- ITR02 and ITR03 exhaust manifolds with NU-ISX12N-04 twin entry turbocharger
- ITR03 exhaust manifold with single wastegate NU-ISX12N-04 twin entry turbocharger
- ITR03 exhaust manifold with NU-ISX12N-04 twin entry turbocharger with a 20- and 30-kilopascal (kPa) pressure drop across the throttle to increase EGR flow capability

The results are summarized in Figure 24.

Figure 24: EGR Versus PMEP for Different Hardware Combinations at Peak Torque

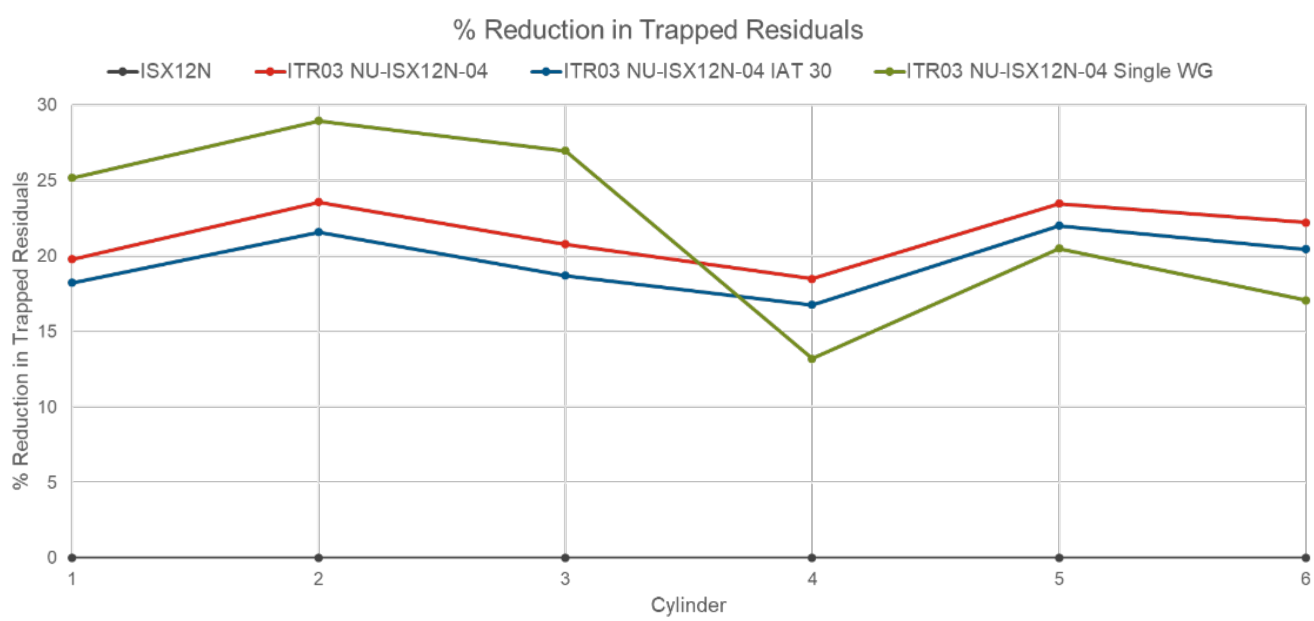


Source: Cummins, Inc.

Both ITR02 and ITR03 manifolds, along with more efficient turbochargers, result in a slightly positive pumping loop — as indicated by a positive pumping mean effective pressure (PMEP) — but at a significantly lower capacity to flow EGR due to the reduced engine pressure differential. Using a more restrictive and less efficient turbocharger, throttling the engine to raise exhaust and boost pressures, or using a single wastegate instead of a twin wastegate on the twin entry turbocharger are some of the ways to increase the pressure differential to reclaim EGR flow capacity. Increasing the pressure drop across the throttle to 30 kPa results in EGR flow capacity that is close to the baseline case while maintaining an almost net zero pumping loop.

The positive pumping loop results in improved scavenging and an approximately 21-percent reduction in trapped hot residuals at peak torque conditions, thereby reducing the tendency of the engine to knock and the required amount of cooled EGR to prevent engine knock. Figure 25 shows the percentage reduction in trapped residuals compared to the baseline ISX12N engine.

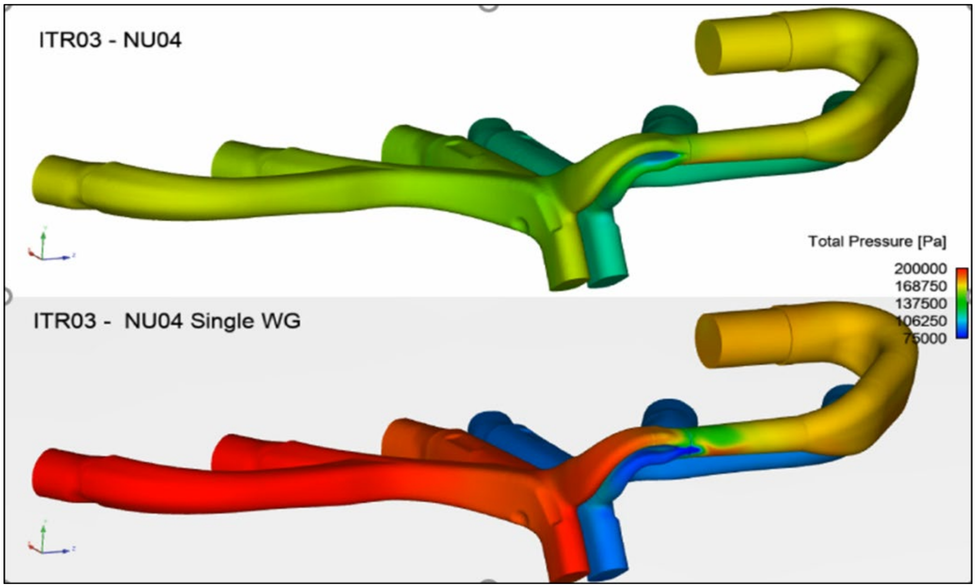
Figure 25: Percent Reduction in Trapped Hot Residuals



Source: Cummins, Inc.

A single wastegate that bypasses one of the two volutes alone of the twin entry turbocharger results in increased EGR flow capability, but it also results in a worse pressure balance between the front and the rear banks of the exhaust manifold, as shown in Figure 26, and in variation in trapped residuals between the front and the rear cylinders.

Figure 26: Exhaust Pressure in Front and Rear Banks for Twin and Single Wastegate Turbocharger

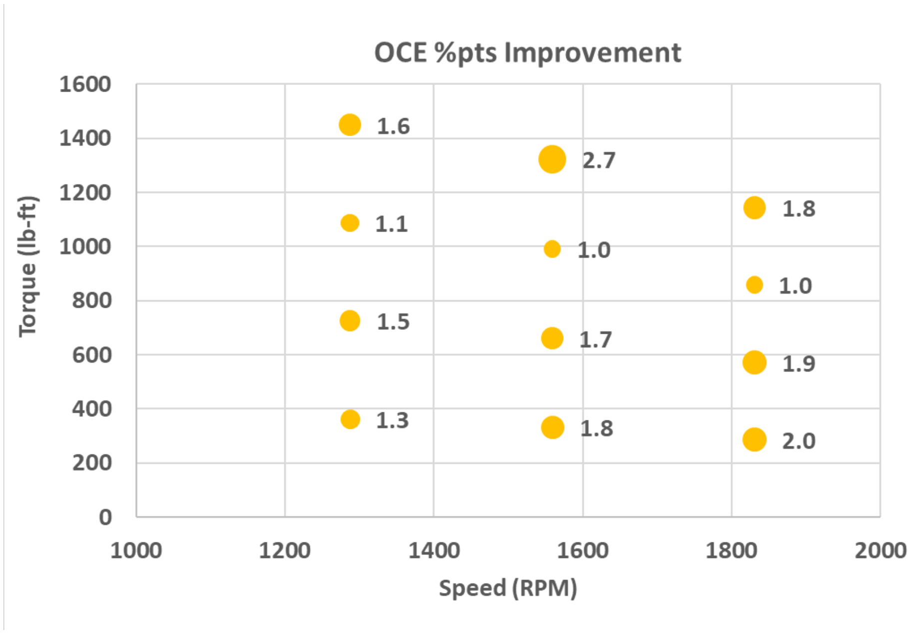


Source: Cummins, Inc.

Next, the exhaust manifold was analyzed for thermo-mechanical fatigue (TMF) requirements per Cummins engineering practice. This analysis evaluates the ability of exhaust system components to withstand repeated thermal cycling. Each cycle for the TMF analysis consists of operation at rated conditions to maximize the temperature of exhaust gases, followed by cooling of exhaust system components at motoring (fuel cut) condition. The TMF life target for exhaust system components depends on engine family size. A target of $10^{3.2}$ cycles has been established for heavy-duty engines, based on experience. A few features of the exhaust manifold showed a significantly lower TMF life. Those features of the exhaust manifold highlighted the areas requiring additional design changes for improvement. The manifold design was further iterated by making subtle changes to specific features based on feedback from TMF analysis, and an ITR06 version was designed.

The ITR06 exhaust manifold was procured, along with the pulse capture EGR system components and the ISX12N-04 twin entry turbocharger for evaluating the OCE improvements on the ISX12N mule engine. The efficiency improvements demonstrated from testing on the ISX12N mule engine with advanced exhaust systems are shown in Figure 27.

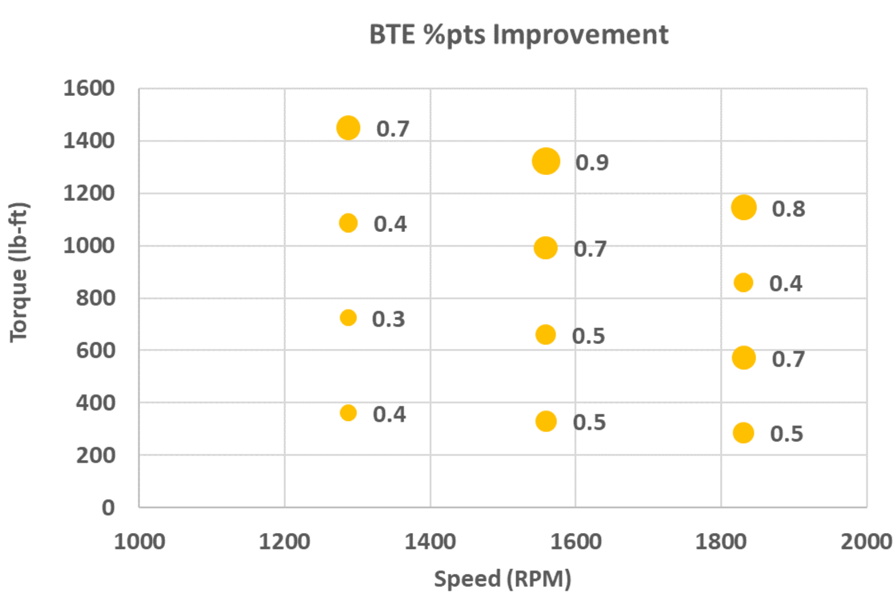
Figure 27: Open Cycle Efficiency Improvements on ISX12N Mule Engine



Source: Cummins, Inc.

Figure 28 shows the resulting BTE improvements compared to the baseline ISX12N engine. Similar or better improvements are expected with the final demonstration engine, with further optimization of the intake manifold and camshaft profile optimization.

Figure 28: Brake Thermal Efficiency Improvements on ISX12N Mule Engine



Source: Cummins, Inc.

Final Engine Design and Demonstration

Cylinder Head Design

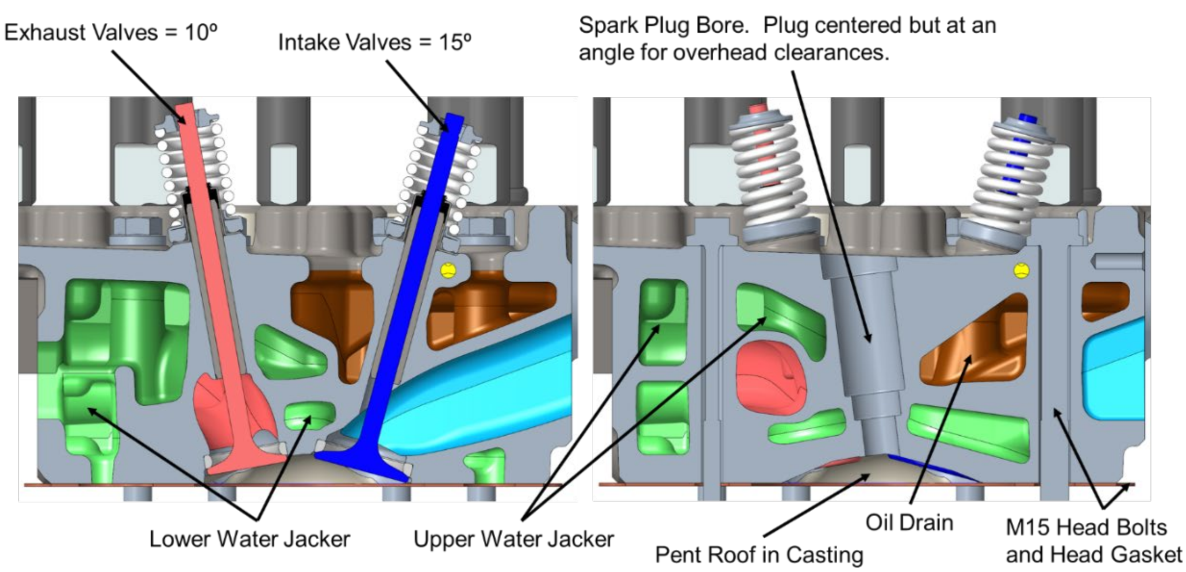
There are many design characteristics and interfaces required for a high efficiency NG engine, and the cylinder head is at the center of nearly all of them. Following is a summary of the most significant design requirements that influenced the final cylinder head design.

First, the head must mount to the existing cylinder block and use the same head gasket to reduce product cost and reduce the time to bring the product to market. This requirement dictated the bore diameter and spacing, oil and coolant feed locations, and head bolt size and locations.

Second, the need for high tumble motion in the combustion chamber dictated angled valves. The high tumble motion requirement is based on a mix of multi-dimensional combustion analysis and the single cylinder engine data reported earlier. Extensive experience from prior Cummins programs suggested that valve angles of approximately 15° from vertical was optimal. The base engine had an offset head bolt on the exhaust side, so the exhaust valve angle needed to be reduced to 10° for head bolt accessibility.

Third, the spark plug was kept in the center of the bore to generate uniform combustion. The ignition point is at the very center of the bore, although the plug bore was angled by 7° to make room for the intake-side components. Figure 29 shows the layout of valves and spark plug.

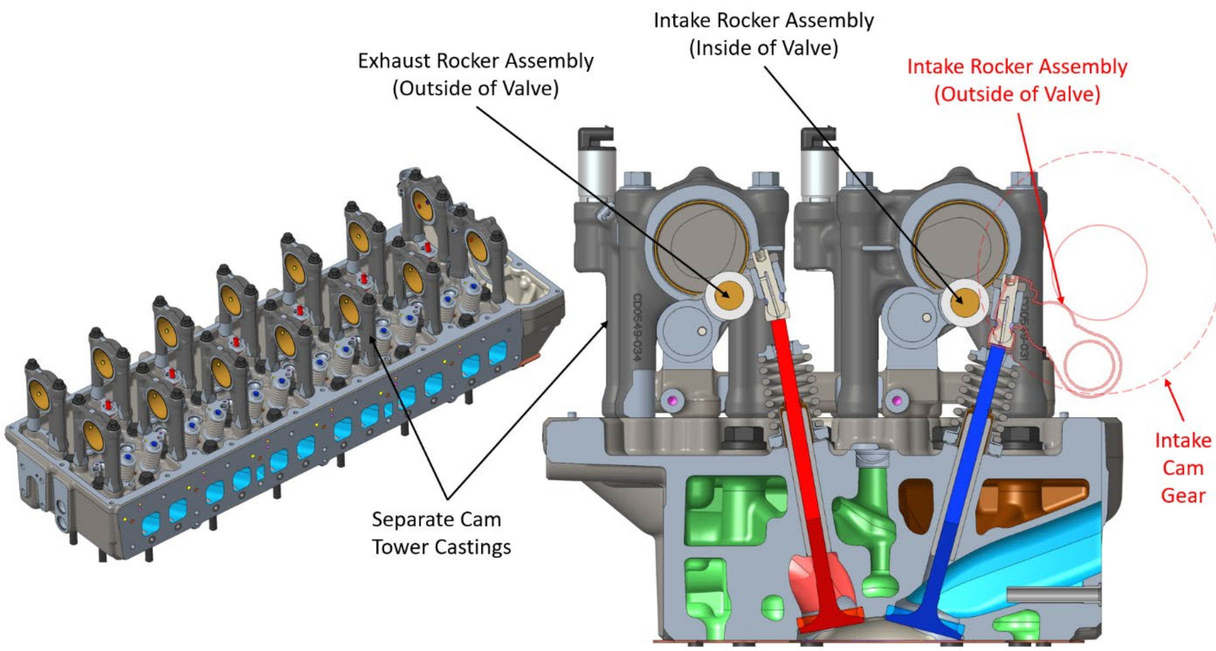
Figure 29: Valve and Spark Plug Layout



Source: Cummins, Inc.

Fourth, angled valves necessitate the need for a dual overhead cam architecture. Because the North America chassis are narrow at the rear of the engine, the intake side of the engine needed to remain narrow as well. This was achieved by positioning the rocker assembly and intake-side camshaft inboard of the valves. A type II valvetrain was selected, primarily due to its compact packaging, but it also benefits from high stiffness and low reciprocating mass. Type II systems have the camshaft mounted above the rocker assembly. If the cam bore machining was part of the head casting, the overall size and weight of the head casting would be high. Therefore, a new architecture was designed that consists of separate cam towers that could be cast individually, then assembled and finish machined on the head. Figure 30 shows the layout of the cam towers and how the head width would be increased if the rockers were outboard. The exhaust side rocker assembly is located outboard of the valve because the exhaust side is less sensitive to vehicle packaging constraints, and the valve angle is reduced to 10°. The intake side has more restrictive OEM constraints at the rear of the engine and the valves are angled at 15°. Consequently, the intake rockers are inboard of the valves.

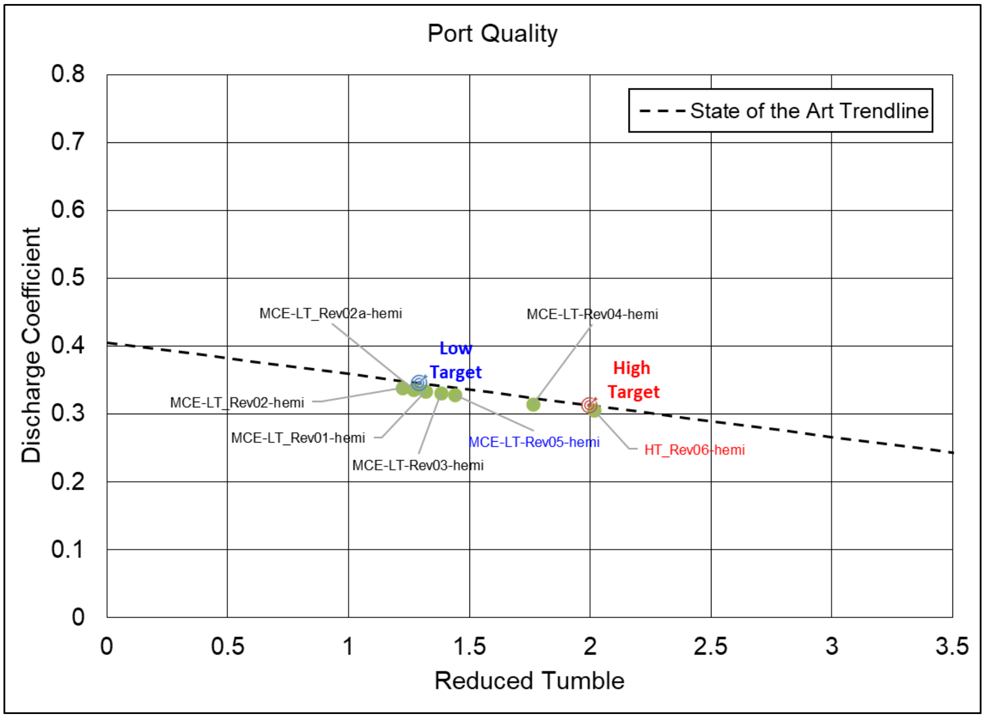
Figure 30: Cam Tower Layout



Source: Cummins, Inc.

Next was the tumble intake port design. The goal was to design intake ports that were efficient at delivering high tumble (tumble target = 2.0) with a high discharge coefficient. Figure 31 summarizes the results, where HT_Rev06 is the final geometry ordered for the multi-cylinder engine. The low tumble cylinder head LT_Rev05 was also designed for potential better efficiency and to mitigate likely ignitability issues due to high velocity at the spark plug gap; however, it was not needed later, after testing of the HT_Rev06 was successful.

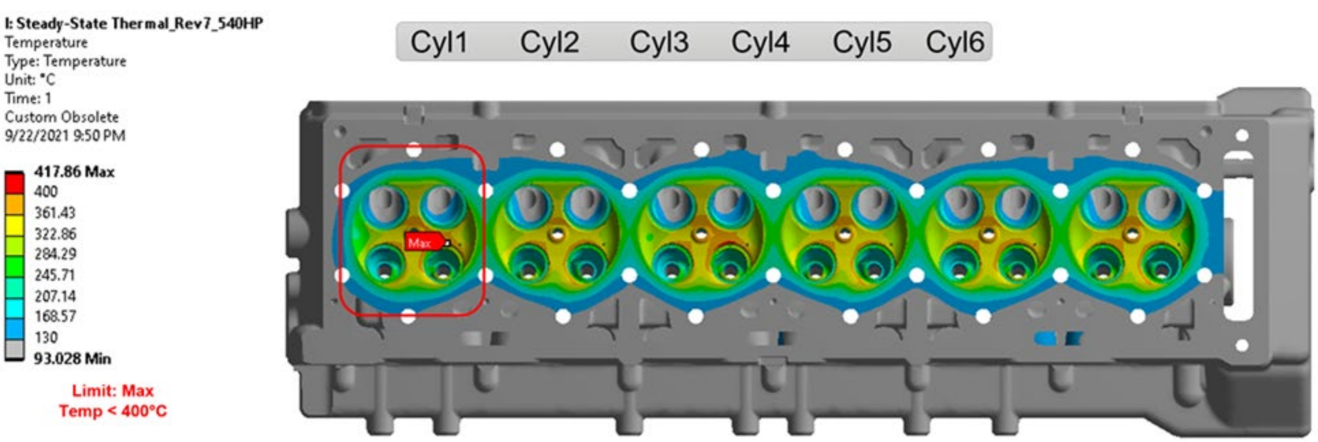
Figure 31: Port Quality Summary



Source: Cummins, Inc.

Figure 32 shows the initial predicted cylinder head temperature results, based on best known boundary conditions. There are two areas where the temperature is predicted to be slightly higher than the material limit of the valve material, but it was determined to not iterate the design more for the initial hardware because of the uncertainty in the initial boundary conditions. A higher fidelity prediction would require an updated calibration of the combustion model and several iterations between the combustion and the fatigue models to converge to a more realistic temperature. The low risk of cylinder head fatigue and temperature-related failures did not dictate the need for the significant time and resources required for further iterations in the design.

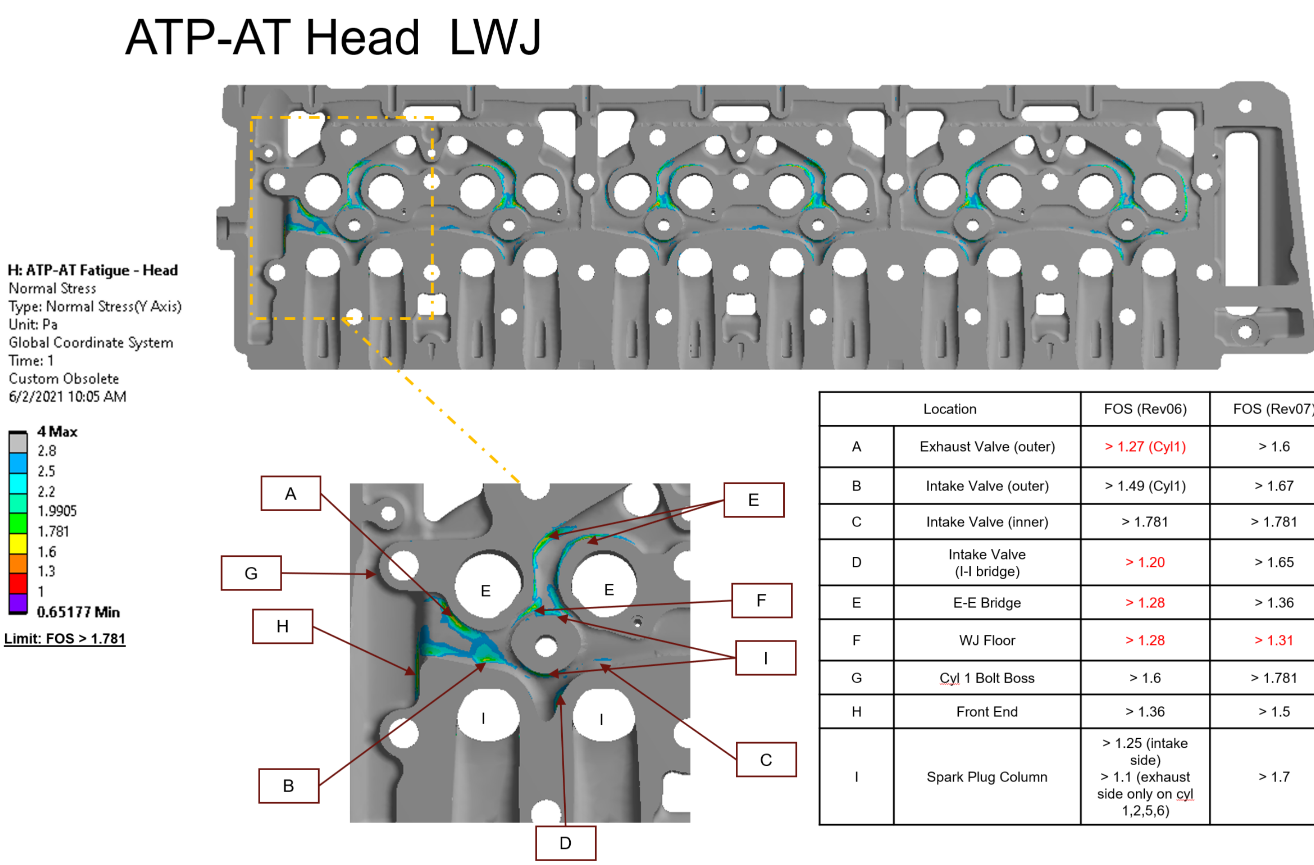
Figure 32: Cylinder Head Predicted Temperatures



Source: Cummins, Inc.

Fatigue life prediction is made by applying the mechanical and thermal boundary conditions and then cycling the loads to simulate a running engine. The PCP requirement for this engine was set to be 210 bar. The fatigue life parameter (FOS) is to be greater than 1.781 or greater than the base engine in the same area of the design. Figure 33 shows a section through the lower water jacket (LWJ). The LWJ is typically the most challenging due to the proximity to the combustion face and the small geometric cross-sections required to route coolant around the valves and spark plug. There are a few areas of the LWJ that are below the 1.781 limit and a few small regions that are below 1.4. These results are well within the range of Cummins' experience for new head designs and will be refined as the project team re-calibrates its models with data from the multi-cylinder engine.

Figure 33: Cylinder Head Fatigue Life Parameter (FOS) Predictions



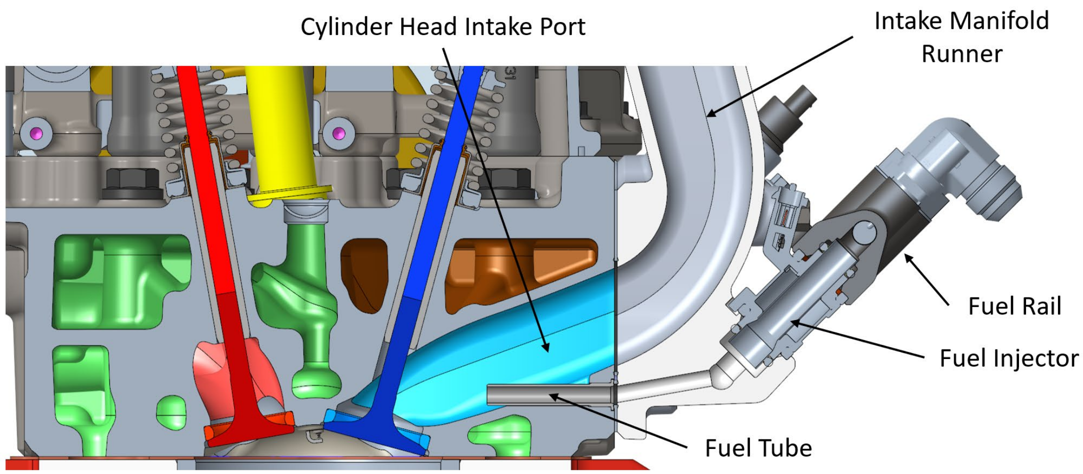
	Location	FOS (Rev06)	FOS (Rev07)
A	Exhaust Valve (outer)	> 1.27 (Cyl1)	> 1.6
B	Intake Valve (outer)	> 1.49 (Cyl1)	> 1.67
C	Intake Valve (inner)	> 1.781	> 1.781
D	Intake Valve (I-I bridge)	> 1.20	> 1.65
E	E-E Bridge	> 1.28	> 1.36
F	WJ Floor	> 1.28	> 1.31
G	Cyl 1 Bolt Boss	> 1.6	> 1.781
H	Front End	> 1.36	> 1.5
I	Spark Plug Column	> 1.25 (intake side) > 1.1 (exhaust side only on cyl 1,2,5,6)	> 1.7

Source: Cummins, Inc.

Fuel System

The intake manifold has two runners per cylinder, and each runner has a solenoid fuel injector. The injector is selected to meet flow demands at low supply pressure. Figure 34 shows the current design layout.

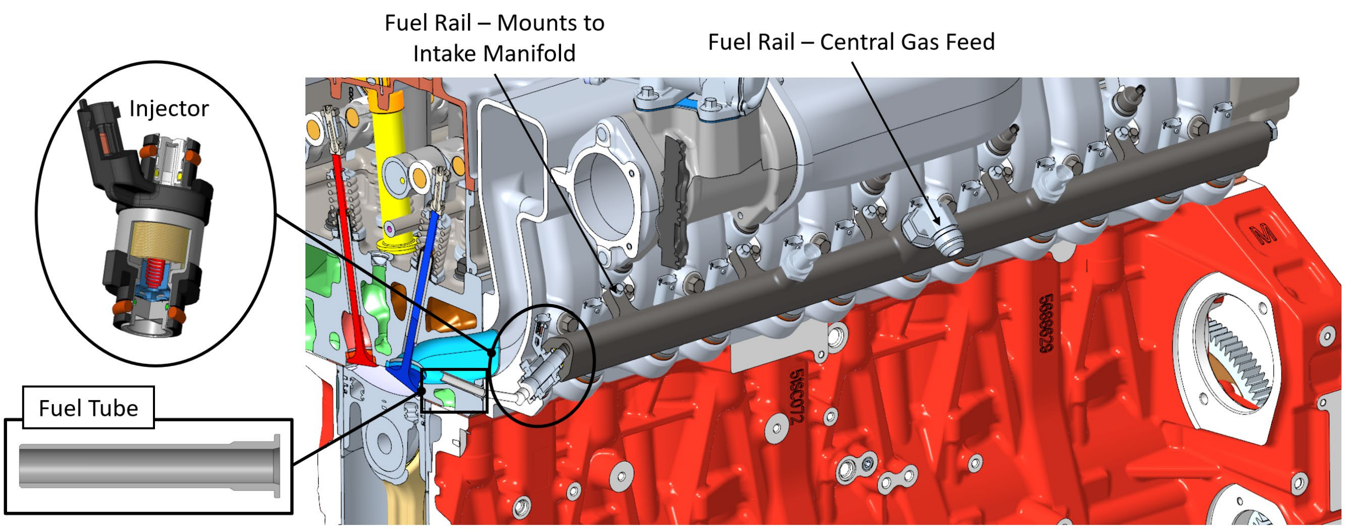
Figure 34: Fuel System



Source: Cummins, Inc.

Figure 35 shows the specific fuel system components. The injector has O-rings on top and bottom that seal to machined ports in the intake manifold and fuel rail. The rail supplies fuel to all the individual injectors through a longitudinal drilling in the rail that has a central gas feed. The rail has mounting tabs that get bolted to bosses on the intake manifold.

Figure 35: Fuel Injector, Fuel Rail, and Fuel Tube

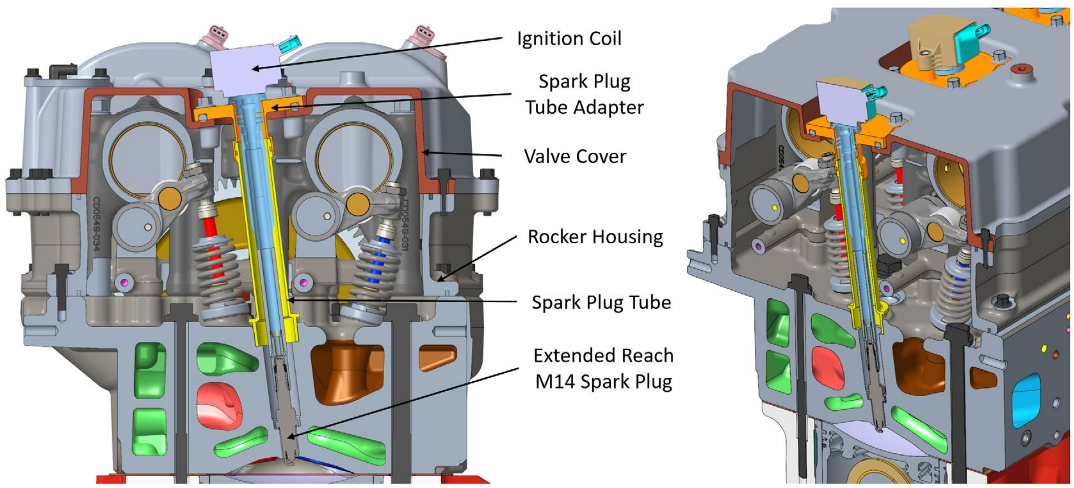


Source: Cummins, Inc.

Ignition System

The ignition system components are shown in Figure 36. As previously discussed, the spark plug is centered in the bore to have the flame propagate from the bore center and radiate radially outward.

Figure 36: Ignition System Components



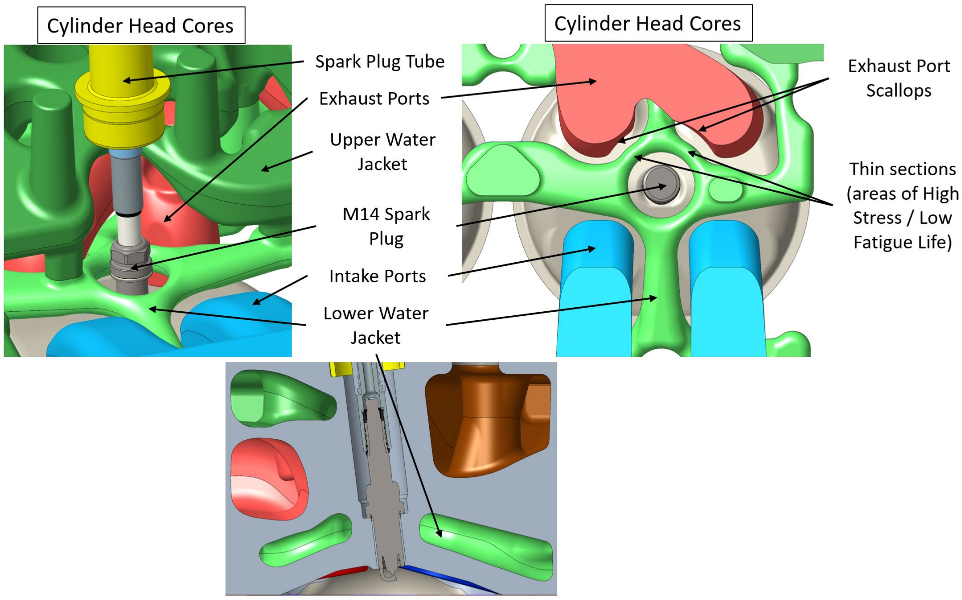
Source: Cummins, Inc.

The spark plug bore is isolated from oil in the overhead by pressing a spark plug tube into a bore in the cylinder head. Because the spark plug is at an angle to clear valve train components, an additional component (spark plug tube adapter) was designed to ease the installation and removal of the valve cover. The order of assembly is to first press the spark plug tube into the cylinder head, then assemble the valve cover to the rocker housing, and finally mount the spark plug tube adapter to the valve cover.

The original intent was to design the system to accommodate an M18 spark plug to allow for performance evaluation against predicted life benefits over the industry standard M14 spark plug. The areas of highest stress (low fatigue life) in the cylinder head occur in the LWJ region that surrounds the spark plug. Figure 37 shows how the spark plug is surrounded by water jackets for cooling. A larger diameter spark plug would require these high stress regions to be smaller in cross-section, which increases stress. Larger scallops could be cut into the exhaust ports or smaller valves, but this would have compromised engine breathability.

An extended reach M14 plug was selected because it allows the threaded region of the spark plug to be closer to the LWJ, which improves cooling and extends the life of the plug.

Figure 37: Spark Plug Cooling



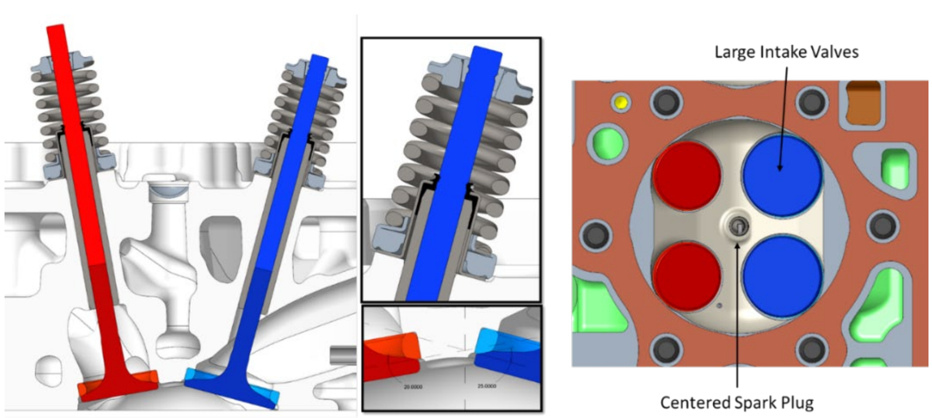
Source: Cummins, Inc.

Valve Train

The driving requirement for setting the valve and seat sizes was to get air most effectively into the cylinder, generate the tumble motion required for optimum combustion, and maintain robustness. The intake and exhaust valve sizing and spacing is optimized to enable enhanced breathing for better performance while keeping the spark plug in the center of the bore and to provide sufficient cooling. This helped maintain commonality across rocker shaft and rocker spacers for both intake and exhaust sides.

The valve and seat angles were set based on internal best practices and guidance from the valve supplier. The intake and exhaust valve seat angles are finalized at 25° and 20°, respectively, to maximize durability, even though larger seat angles were shown to help improve port flow efficiency. Figure 38 shows the overall layout of the valve hardware and Figure 39 shows some of the fatigue calculations for the valves.

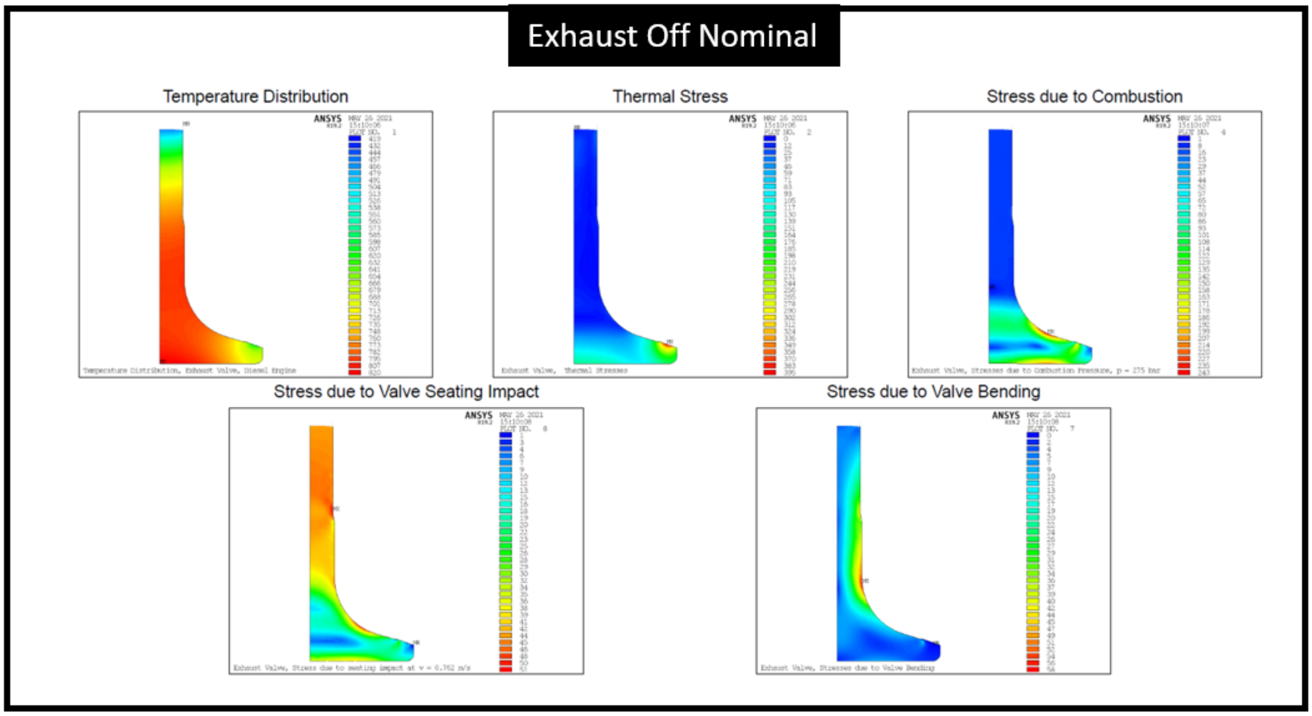
Figure 38: Valve Hardware



Source: Cummins, Inc.

Figure 39: Valve Wear and Fatigue Analysis

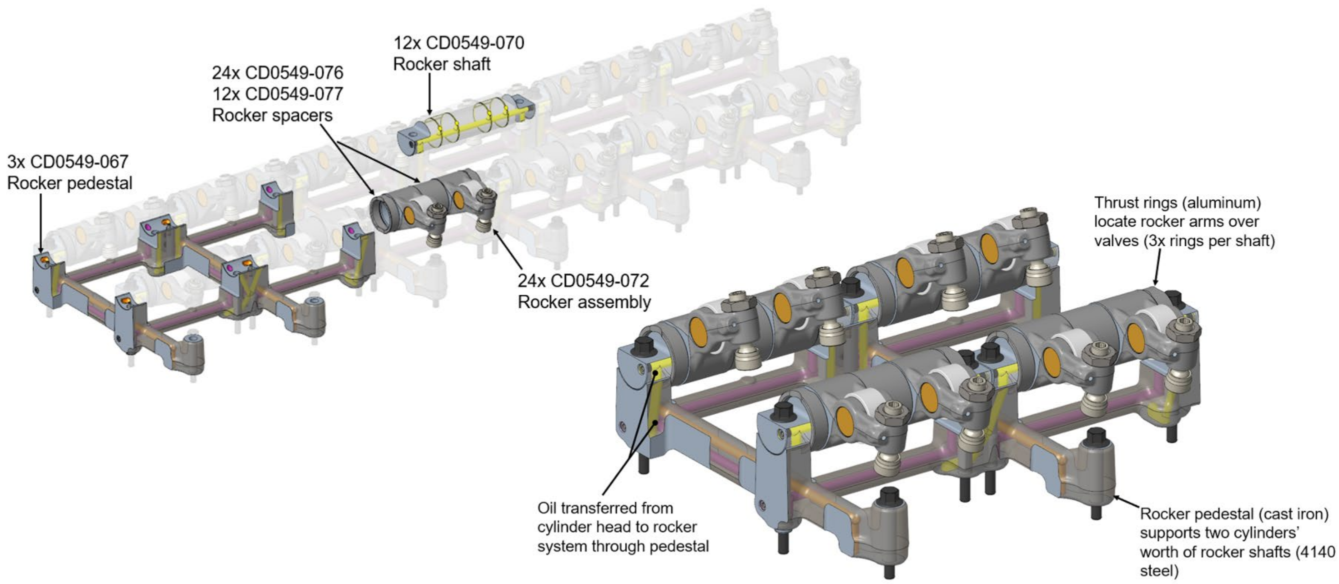
	Nominal Operation		Off-Nominal Operation	
	Intake VAT36-Sil1	Exhaust Ni31V-Sil	Intake VAT36-Sil1	Exhaust Ni31V-Sil
Maximum Temperature (°C)	605	799	630	820
Thermal Stress (MPa)	198	386	205	395
Stress due to Combustion Pressure (MPa)	257	188	333	243
Stress due to Valve Seat Impact (MPa)	53	51	53	51
Stress due to Valve Bending (MPa)	85	58	85	58
Consumed Fatigue Life (CFL)	< 0.01	< 0.01	< 0.01	< 0.01
Fatigue Factor	2.50	> 10	2.53	7.4



Source: Cummins, Inc.

The rocker system was designed as a type II. Figure 40 shows the basic rocker system components, with emphasis on part commonality. For example, the assembly consists of 3 identical rocker pedestals, 12 rocker shafts, and 24 rocker levers.

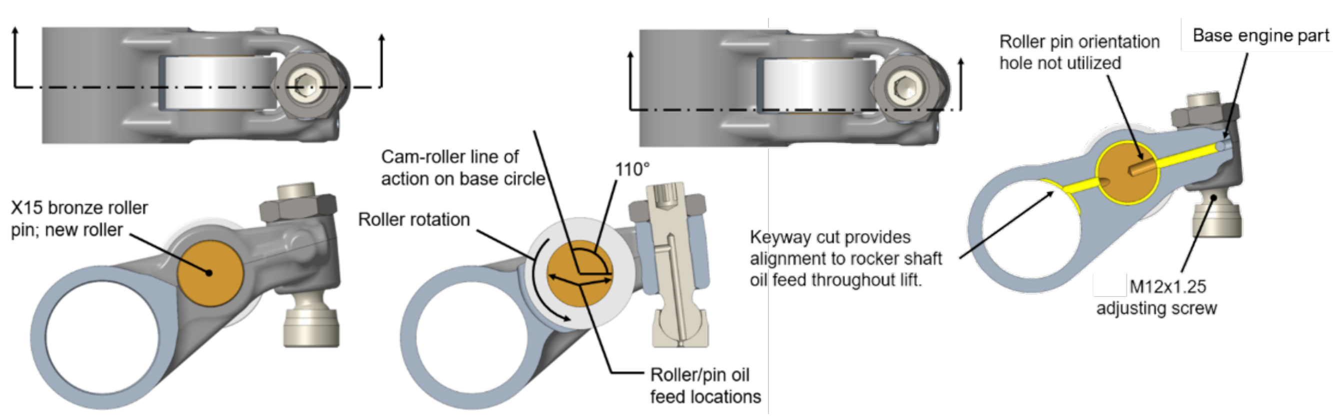
Figure 40: Rocker System



Source: Cummins, Inc.

Figure 41 shows the rocker levers in detail. All 24 valves are actuated by a common rocker lever designed with a rocker ratio of 1.67, which is well within the Cummins product experience for mid-range, heavy-duty, high horsepower, and NG engines. Valve lash adjustments are made using an adjusting screw and nut from a current product heavy-duty engine. Figure 42 shows intake and exhaust rocker lever fatigue results, which also fall within Cummins design limits.

Figure 41: Rocker Levers

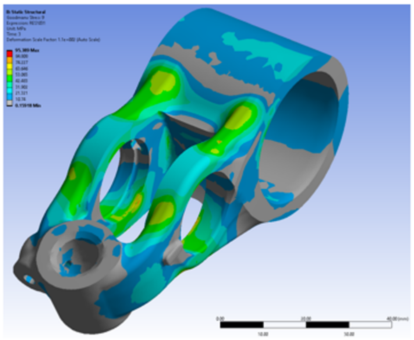


Source: Cummins, Inc.

Figure 42: Rocker Lever Fatigue

Exhaust Fatigue

EFR stress plot

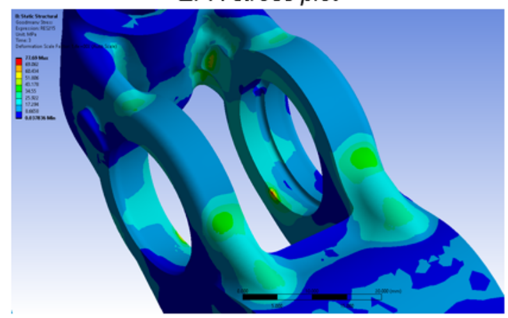


Peak loading on exhaust – 3658 N

Location/limit	FOS
Pin bore	>3.0
Ribbing	>3.2
ESW target	2.0
CES41020 per EP00006985	2.469

Intake Fatigue

EFR stress plot



Peak loading on intake – 2664 N

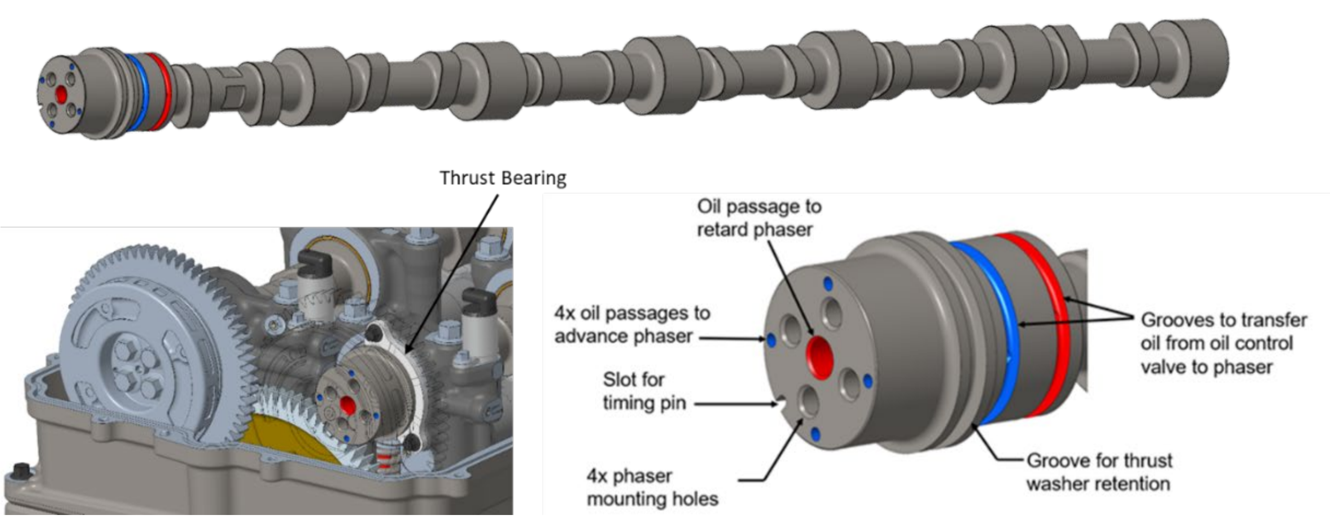
Location/limit	FOS
Pin bore	>3.03
Roller clearance cut	>3.92
Ribbing	>5
ESW target	2.0
CES41020 per EP00006985	2.469

Source: Cummins, Inc.

Camshafts and Cam Phaser

In addition to the valve actuation requirement, the intake and exhaust camshaft also mount the cam gear and cam phasers. Figure 43 shows detail at the rear of the camshaft, including oil transfer ports that move oil from the rear cam journals to the cam phaser through radial grooves and intersecting drillings. A thrust bearing groove is present to control the axial position of the camshafts.

Figure 43: Camshaft

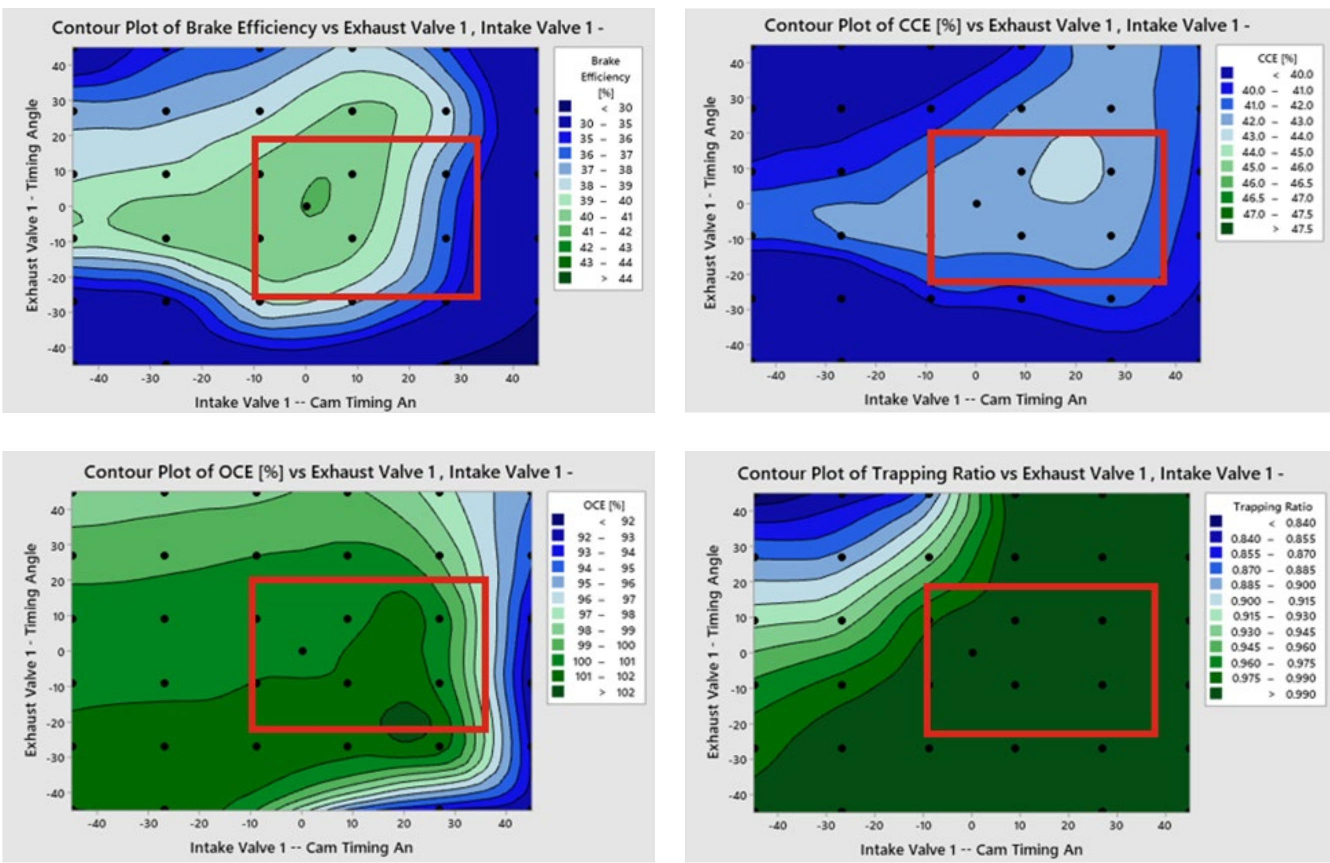


Source: Cummins, Inc.

A dual overhead camshaft with independent phasing control provides the flexibility to adjust the valve overlap to minimum at low loads to minimize residuals for stable low load performance. It also provides the flexibility to increase overlap where needed for optimum scavenging to maximize volumetric efficiency and minimize trapped residuals to increase the knock margin. Moderate Miller cycling is also being exploited to enable a higher compression ratio without a significant volumetric efficiency and turbocharging penalty. At lower loads, particularly near idle, pressure boundaries become critical for residual management. This multi-faceted optimization space is handled well by analytical methods, and graphical representation can be helpful to show the sensitivity.

At peak torque, the air handling efficiencies are favorable to allow for positive engine delta pressure in some conditions. The target operation space shown in Figure 44 is obtained by reviewing localized results and obtaining global compromise. It shows efficiency trade-offs versus the cam phasing adjustment angle with the target operating space (red box).

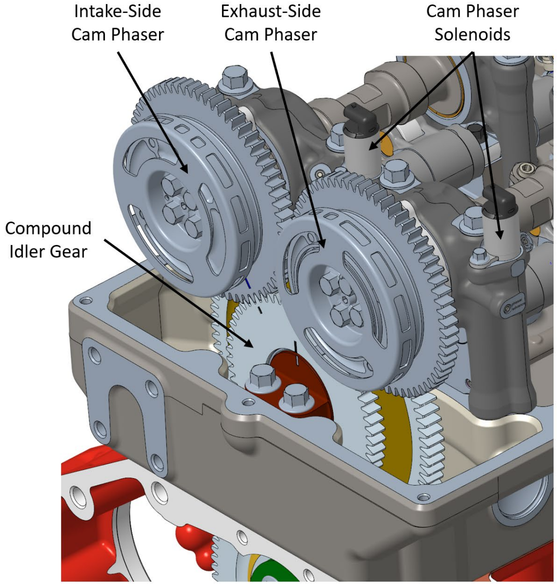
Figure 44: Cam Phaser Operation Space/Impact on Efficiency

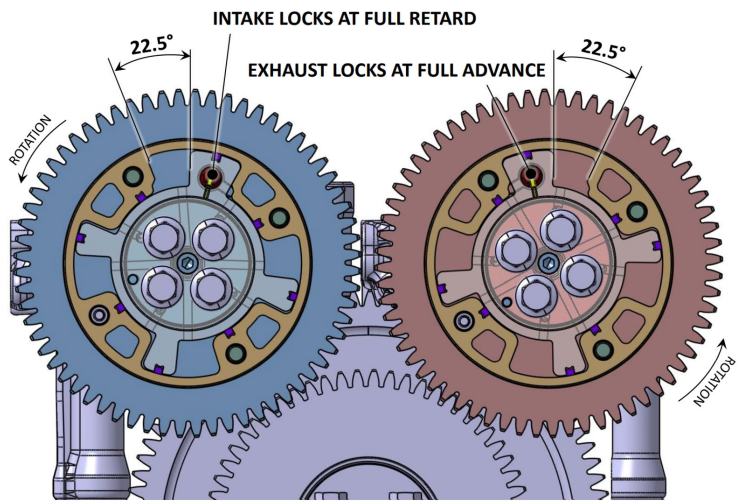


Source: Cummins, Inc.

The two cam phasers were designed to mount to the rear of the intake and exhaust cam gears. Figure 45 shows the design layout.

Figure 45: Cam Phaser Layout





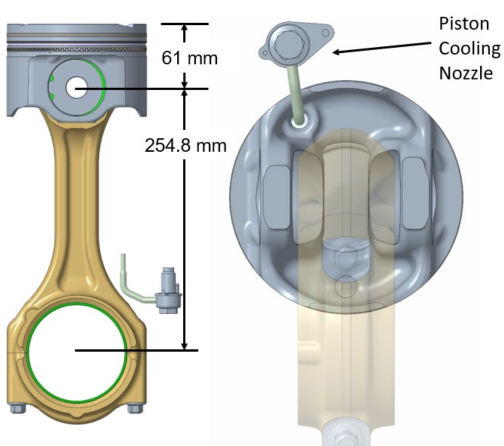
Source: Cummins, Inc.

Each phaser has a range of authority of 22.5 camshaft degrees (or 45 crankshaft degrees), with the intake phaser locking at full retard position and the exhaust phaser locking at the full advance position; this is achieved with a bias spring. These phaser positions are independently controlled with solenoid valves mounted on the rear cam towers.

Power Cylinder

Figure 46 shows the rod, piston, and piston cooling nozzle layout. With the piston pin joint, compression height, and oil feed location already defined, the unique portion of the piston design was the design of the piston bowl, under crown oil gallery, and ring grooves.

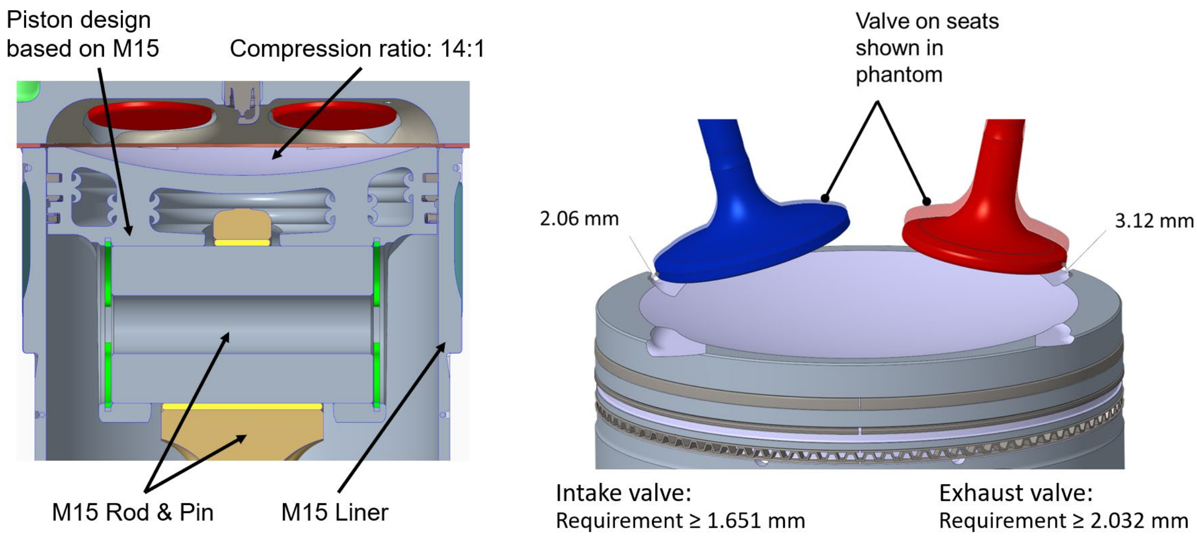
Figure 46: Compression Height, Rod Length, and Piston Cooling Nozzle



Source: Cummins, Inc.

The piston bowl design and cylinder head pent roof design were optimized together to achieve the desired compression ratio range capability. The size, location, and angle of the intake and exhaust valves largely dictate the volume in the head. Figure 47 shows piston architecture details. The requirement was to design the cooling gallery and under crown to enable a CR ranging from 13:1 to 16:1. The 13:1 CR has the deepest bowl and thus dictated the shape of the maximum shape of the oil gallery. A CR of 14:1 was ultimately selected for initial data collection to meet performance requirements while maintaining minimum wall thickness and clearance to the connecting rod. Shallower bowls can be cut into the piston, but the under-crown wall thickness will increase as a result and piston wall temperatures may be impacted. A higher CR also tends to increase the risk of knock and pre-ignition. Procurement plans include flat top piston hardware, which allows for testing at different compression ratios with minor machining changes as needed.

Figure 47: Piston Design



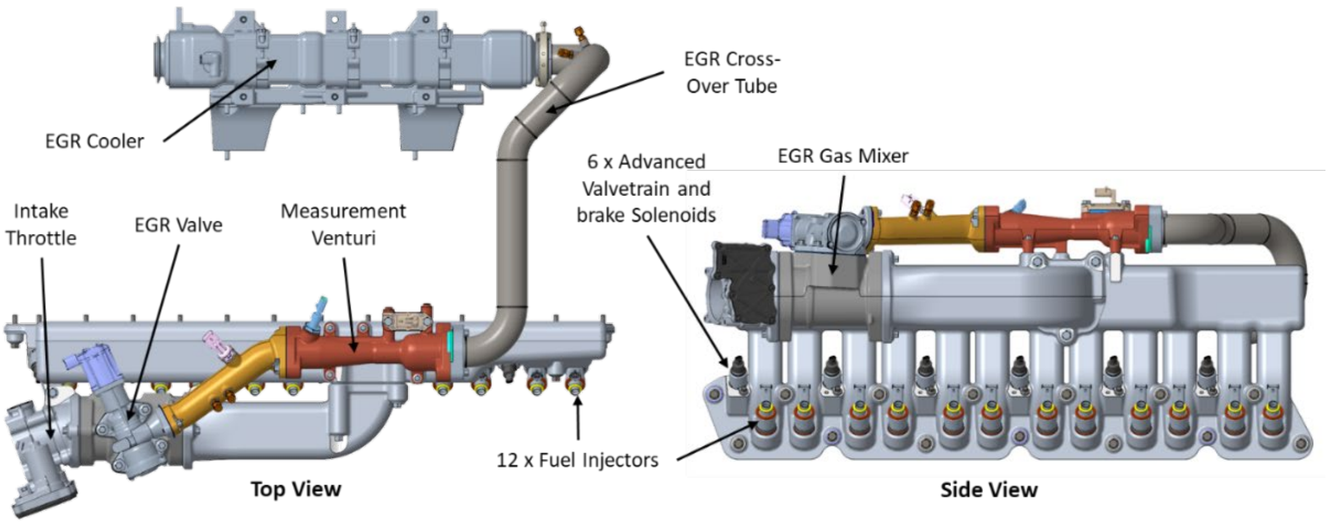
Source: Cummins, Inc.

The piston and piston pin analysis was satisfactorily completed by the supplier. It was conducted using predicted heat transfer coefficients from a combustion model and 210-bar PCP for inputs. The piston passed finite element analysis limits for all areas of the piston.

Air Handling

Figure 48 shows the layout of the intake manifold and EGR system. The intake manifold mounts directly to the side of the cylinder head and is responsible for mixing the EGR gas with the fresh air and delivering the fuel to the cylinders. The EGR circuit includes the EGR cooler, flow measurement venturi, control valve, and a static mixer. The EGR cooler is mounted to the cylinder head. The EGR cross-over tube mounts to the rear of the EGR cooler and routes the cooled exhaust gas across the valve cover to the measurement venturi. The gas then passes through the EGR valve and EGR gas mixer, where it is mixed with the fresh intake air. The intake manifold was designed with two runners per cylinder with one fuel injector mounted in each runner. The manifold provides mounting locations for six oil control solenoids needed to control the advanced valve train.

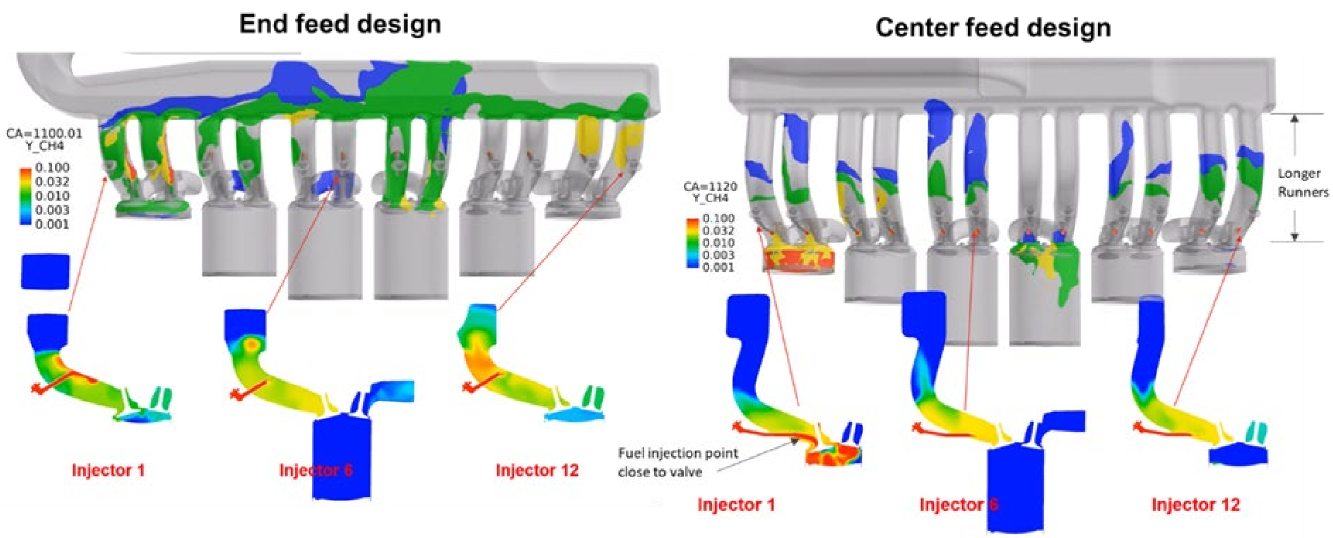
Figure 48: Intake Manifold and EGR Components



Source: Cummins, Inc.

One key design parameter that was used to optimize the performance of the intake manifold was the fuel injection performance. The intake manifold has a much longer intake runner length to eliminate charge and fuel being pulled into neighboring cylinders (crosstalk). In addition, the fuel injection point was moved to be closer to the intake valve and at the bottom of the port, which also effectively increased the runner length. Figure 49 shows CFD predictions for two concepts. In the optimized center feed design on the right, the residual fuel is shown to be held within the runner and hence not drawn into neighboring cylinders.

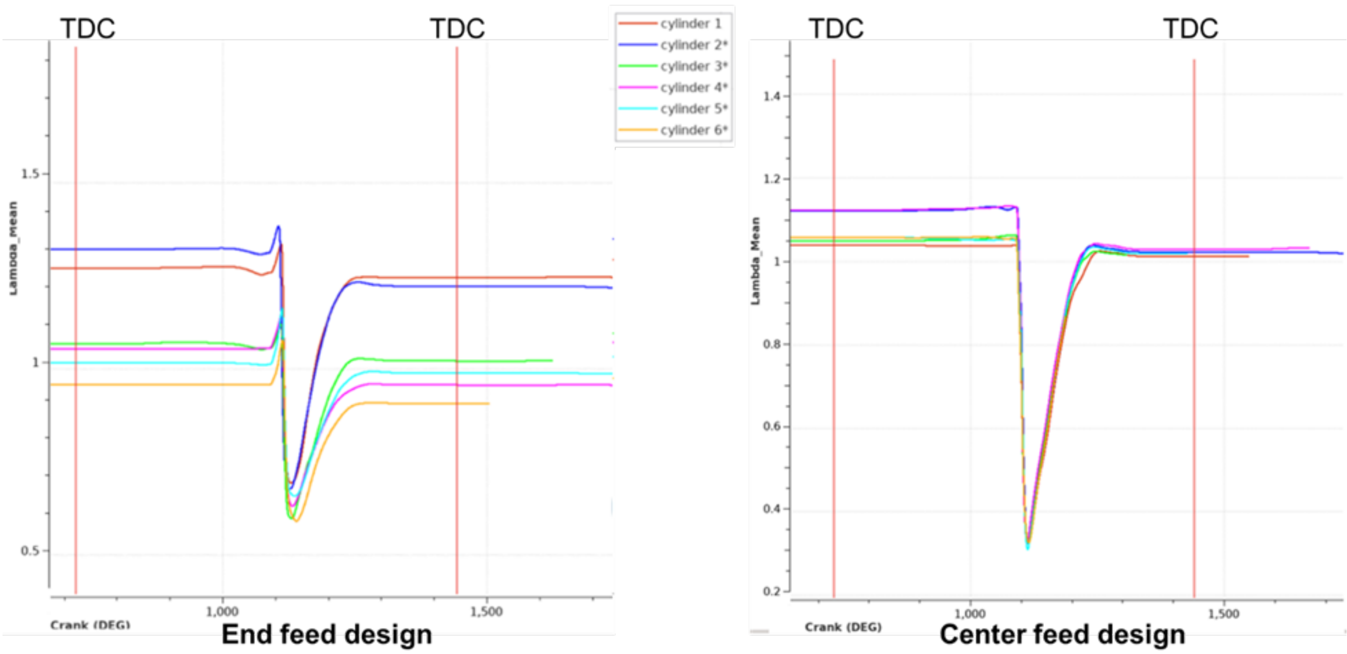
Figure 49: Intake Manifold Optimization for Fuel Mixing



Source: Cummins, Inc.

In addition to charge distribution between cylinders, another key performance metric in the iterative optimization is the mean lambda variation. Figure 50 shows a tight distribution of mean lambda values for center feed design, indicating excellent balance per cylinder. These figures also show the history of the model convergence from prior cycles and the air/fuel ingestion during the intake stroke.

Figure 50: Mean Lambda Distribution per Cylinder

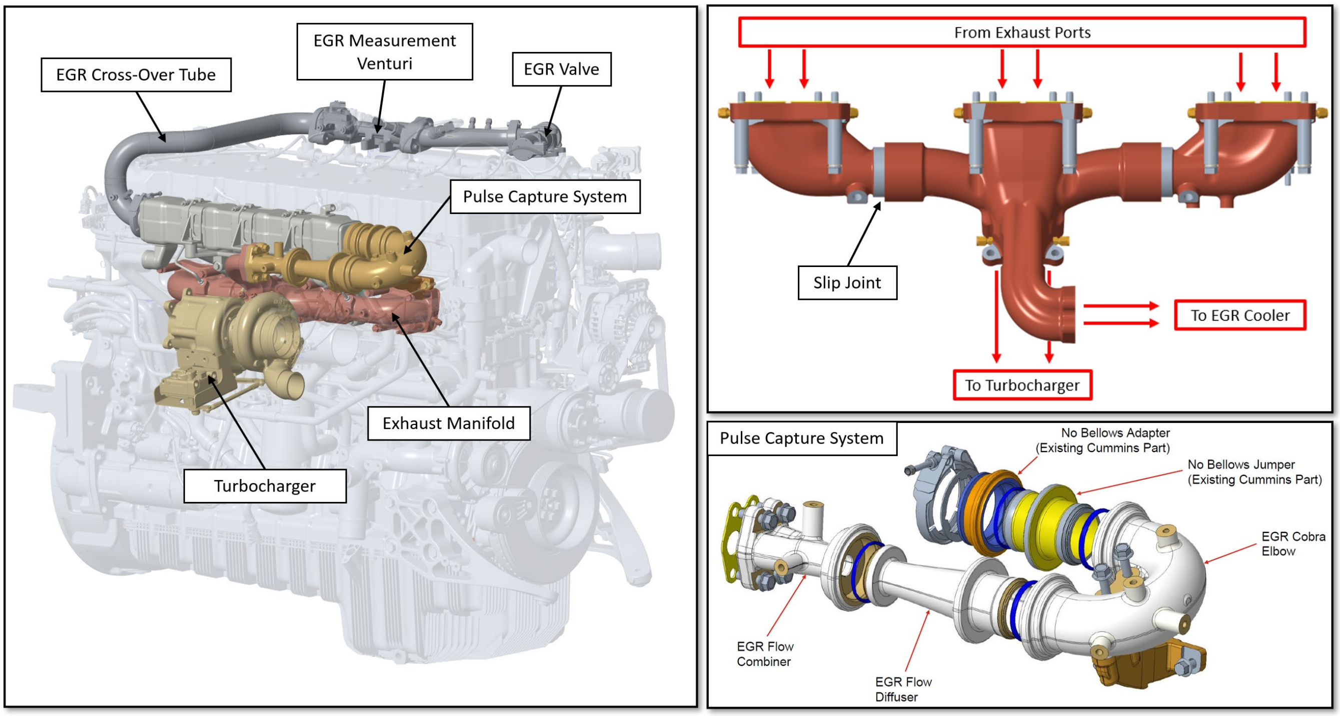


Source: Cummins, Inc.

Exhaust System

A divided exhaust manifold isolates the front and rear banks of the engine to minimize interaction between cylinders. This allows for the use of the pulse momentum while minimizing the crosstalk with anti-flow reversion features built into the pulse capture design. This gives increased EGR with reduced engine differential pressure, further contributing towards OCE improvement. This enables a higher efficiency turbo charger matching with better swallowing capacity, resulting in positive or near zero pumping. Building upon earlier ISX12N mule engine learnings, dual wastegate ports are included to maintain cylinder-to-cylinder balance via consistent manifold pressure boundaries. The exhaust system hardware is shown in Figure 51.

Figure 51: Exhaust Manifold and Pulse Capture System

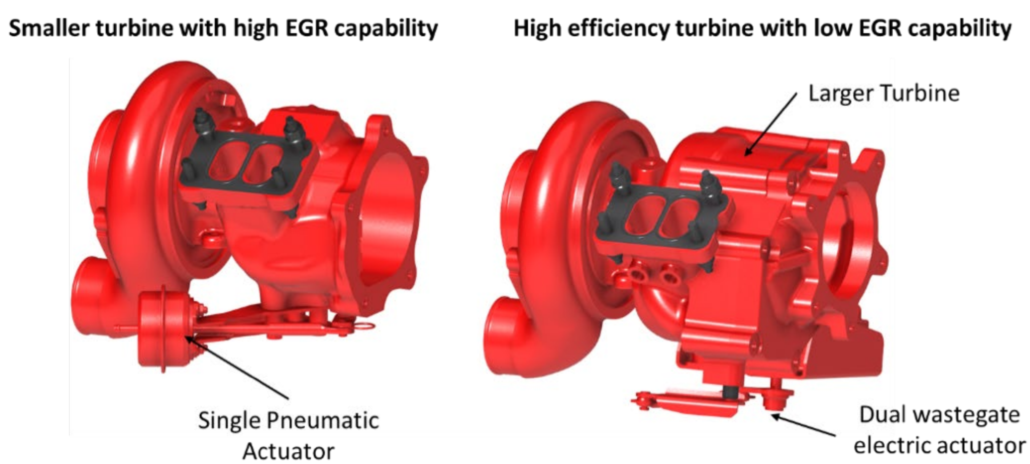


Source: Cummins, Inc.

The turbocharger sizing was completed using GT-Power analytical models. Design factors include turbine inlet temperature, altitude capability, turbocharger speed, surge margin, and EGR potential. A full factorial optimization was performed using map flow scaling to obtain initial targets and the selections were verified.

For a selected hardware configuration, the addition of EGR has clear knock mitigation, efficiency, and turbine inlet temperature benefits. Mule multi-cylinder engine test data is not available for a clear direction on a single design option; hence existing knock models were used to select multiple options with different EGR capability. These are shown in Figure 52.

Figure 52: Twin Entry Turbocharger Options



Source: Cummins, Inc.

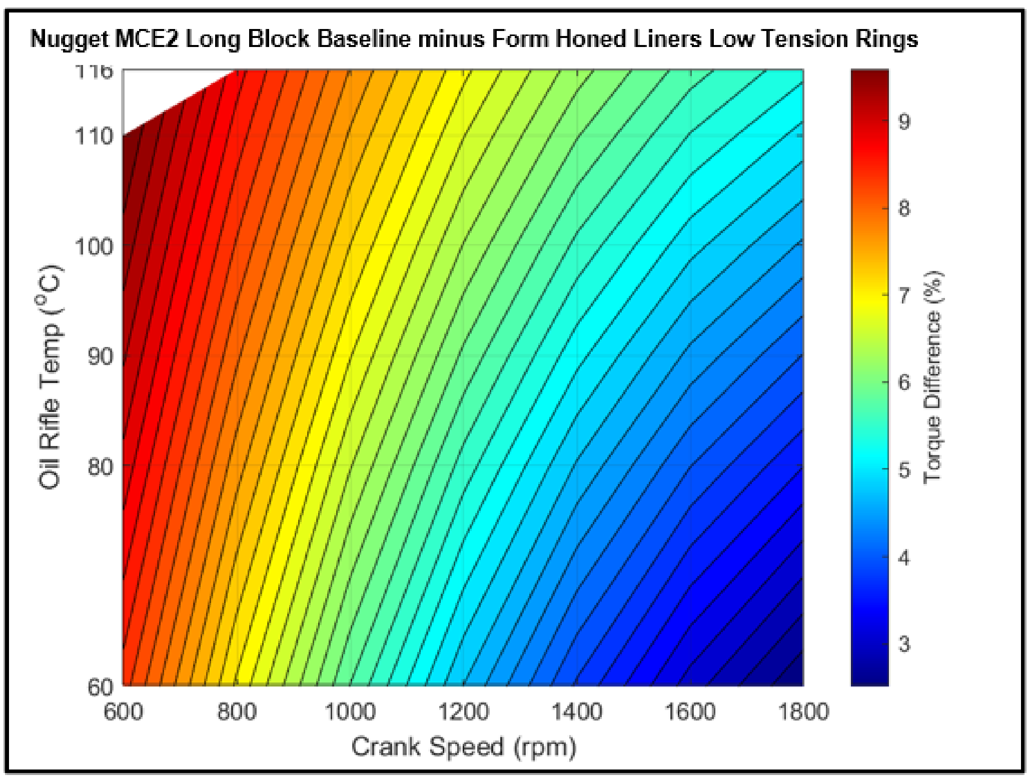
The first build uses a smaller turbine with a higher EGR capability to supplement the pulse capture EGR rates. Transient performance is achieved via target EGR rate adjustment and, as such, is kept with a traditional electro-pneumatic wastegate.

A high efficiency turbo charger with a lower EGR capability option was available later to upfit on the engine. It relies on an intricate balance of the turbine energy to achieve minimum necessary EGR targets while maintaining altitude capability. Two different turbine wheels are selected near the predicted target. With lesser excess turbine power, the transient spool up becomes even more critical and hence a full authority electronic wastegate control is fitted. The experimental results will be used to compare transient torque capability as well as part load efficiency trade-off versus that obtained with camshaft phasers alone.

Friction Reduction Technology Assessment

Key technologies contributing to friction reduction were addressed earlier in the report. Because of resource constraints, the project did not accommodate them in the final engine demonstration except the low compression height piston/long rod. The Cummins mechanical efficiency lab was used to quantify benefits of the form honed liner with torque plate and low-tension rings. The combination of the lower tension rings and the form honed liners improved the friction torque of the engine by between 4 percent and 9 percent across the engine speed range at nominal oil temperatures of 212°F (100°C), as shown in Figure 53.

Figure 53: Friction Reduction Technology Benefits



Torque (Nm)

Source: Cummins, Inc.

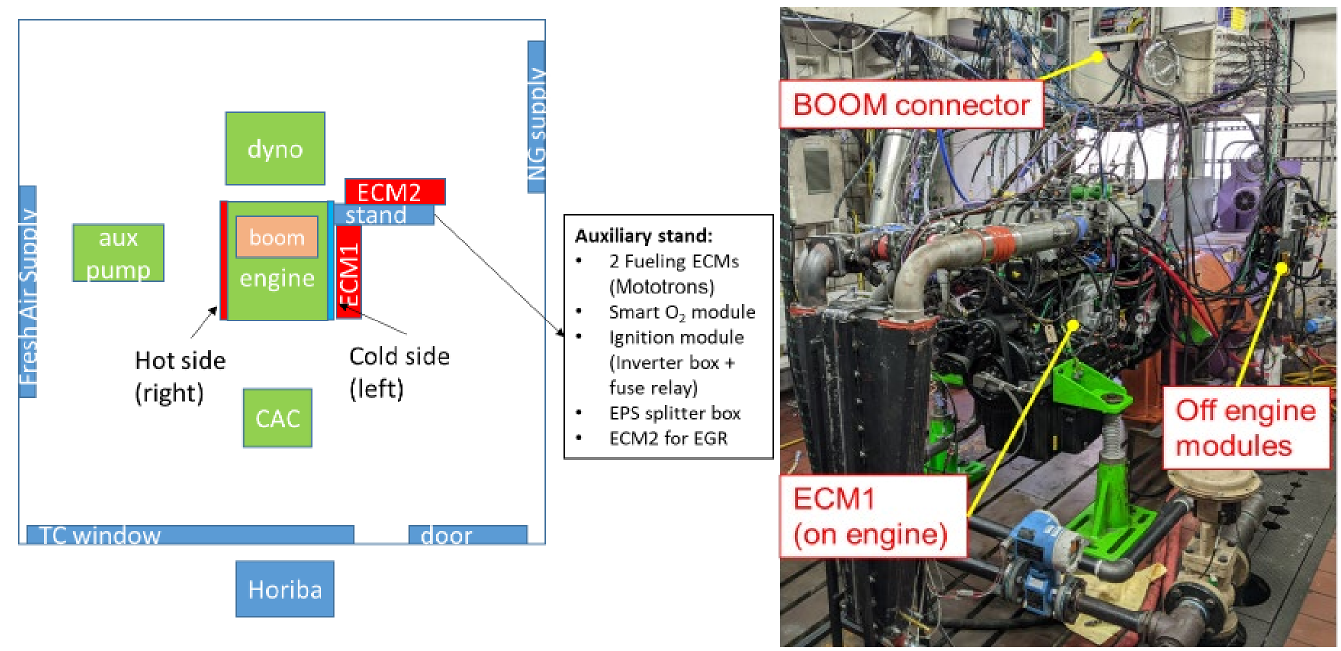
Aftertreatment

The aftertreatment architecture is based on learnings from prior NG engine products utilizing TWCs. The initial engine design included direct mounted close couple aftertreatment, although this continues to pose packaging constraints in foreseeable chassis; hence, an underfloor only TWC like the ISX12N was finalized for final engine demonstration.

Final Engine Demonstration

Figure 54 shows the test cell layout and the engine installed in test cell at the Cummins Technical Center in Columbus, Indiana. Because this is a prototype configuration, multiple control systems are integrated to produce the required control synergy.

Figure 54: Test Cell Layout and Engine Installation in Test Cell



Source: Cummins, Inc.

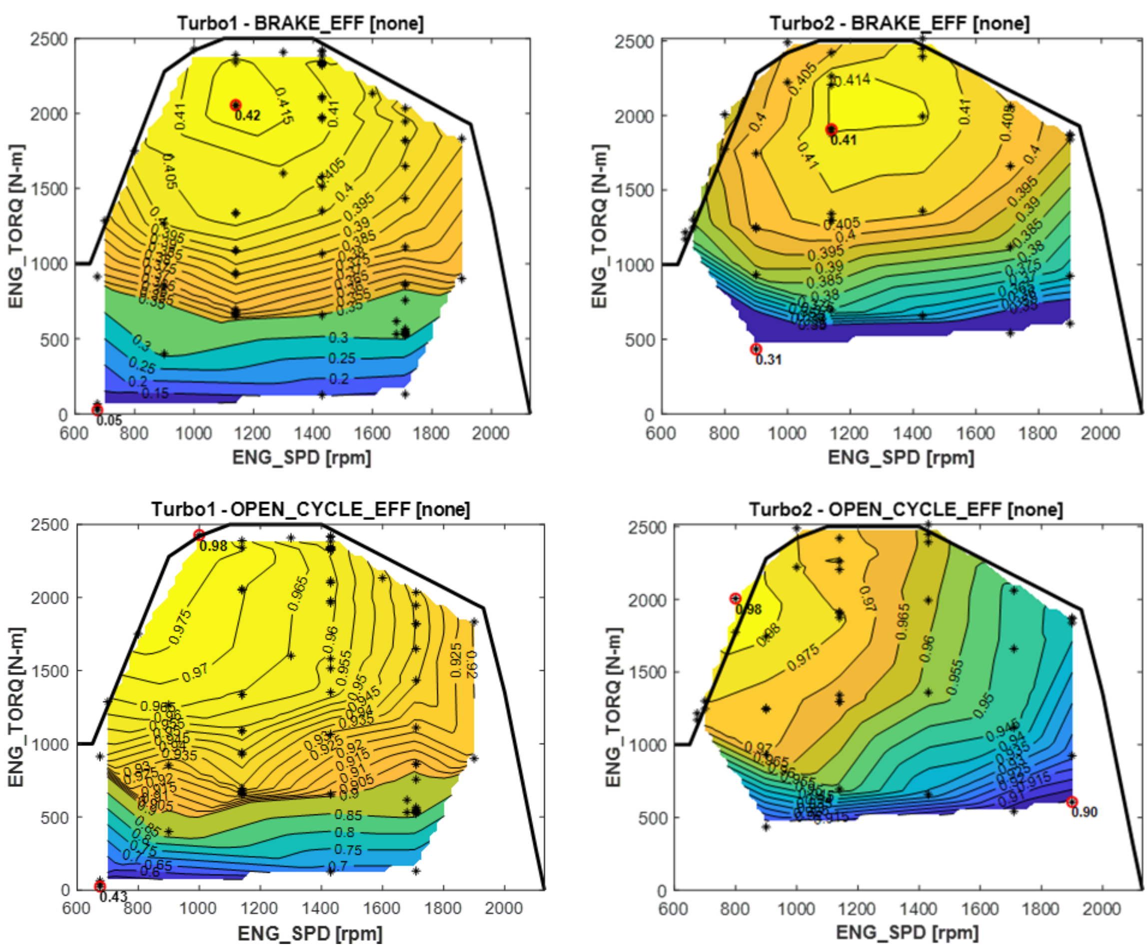
Leveraging the developed engine system model, localized optimizations were performed to establish the initial control system targets and virtual sensor tuning. Engine operation was limited to a sufficient, but less demanding, torque curve. Data points were collected throughout the speed and load operating space, with localized optimization taking place for combustion references (CA50 and EGR targets), intake phaser command (PMEP management, volumetric efficiency, and EGR potential), exhaust phaser command (scavenging, blowdown, and volumetric efficiency), intake air throttle (load control and EGR potential), and wastegate commands (PMEP control, load control, and EGR potential). The model-based virtual sensors proved sufficient, allowing for steady-state closed loop lambda control and initial combustion references that were quite helpful in narrowing the regions of interest. The smallest (Turbo1) and the largest (Turbo2) turbocharger were tested to understand extreme differences and a final recommendation was made.

The contours in Figure 55 were generated using the engine test results from Turbo1 and Turbo2 configurations to compare and draw conclusions on hardware recommendations for testing and final demonstration. Turbo1 represents a smaller turbine with high EGR driving capability. Turbo2 represents a higher swallowing capacity turbine with lower EGR rate capability.

BTE contour plots show evidence that the high efficiency regions were larger on Turbo2 at mid- to high-load regions compared to Turbo1. As a result, there is a slight advantage for RMCSET Cycle average efficiency with the Turbo2 option. While there is a point with slightly higher BTE (42.0 percent) with Turbo1, the project team was unable to explore the full potential of Turbo2 due to some phaser control issues related to position sensing and lube system-related challenges during the testing. With full phaser control, Turbo2 has the potential to be at a similar BTE range. OCE contours are very similar between the turbochargers.

The largest difference becomes apparent with Turbo2 at high engine speeds, where engine breathing is improved with the larger swallowing capacity turbine housing.

Figure 55: BTE and OCE Contour Comparisons With Turbocharger Options

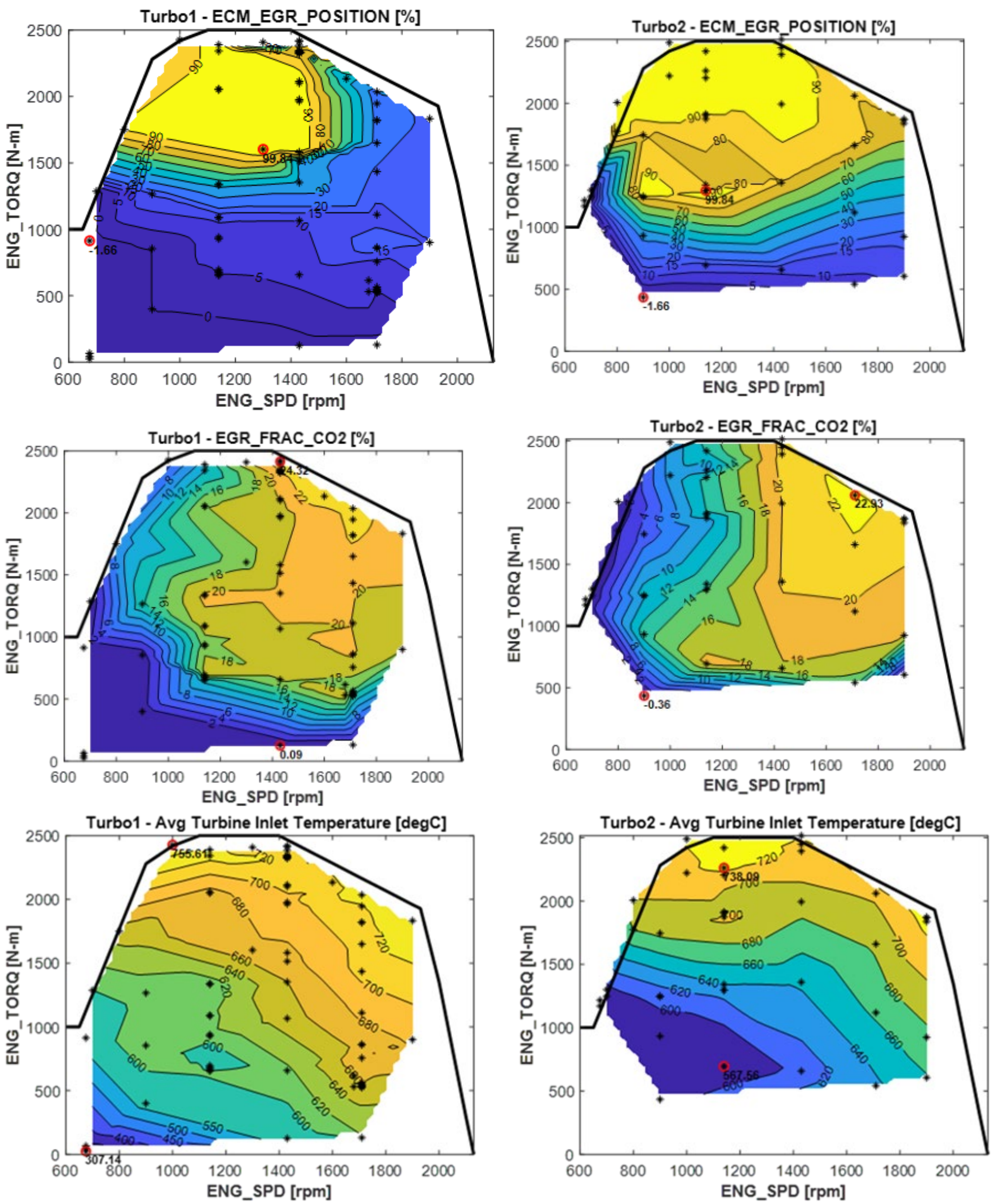


Source: Cummins, Inc.

Except for the highest engine speeds, where they are the same, test cell measured EGR rates are generally lower with the larger trim turbine housing. It is evident from the contour plots in

Figure 56 that Turbo2 is operated with a more open EGR valve due to the lower engine delta pressure. It is also evident that the engine can demonstrate peak torque of 2500Nm at 1000 rpm within acceptable limits. At peak torque condition, Turbo1 is already at its turbine inlet temperature limits; however, Turbo2 still has some margin. This would further allow for an exploration of higher peak torque at lower speeds with Turbo2; delta of ~72°F (40°C) improvement is observed at peak torque and an improvement of 18°F (10°C) to 36°F (20°C) is observed at the rated condition.

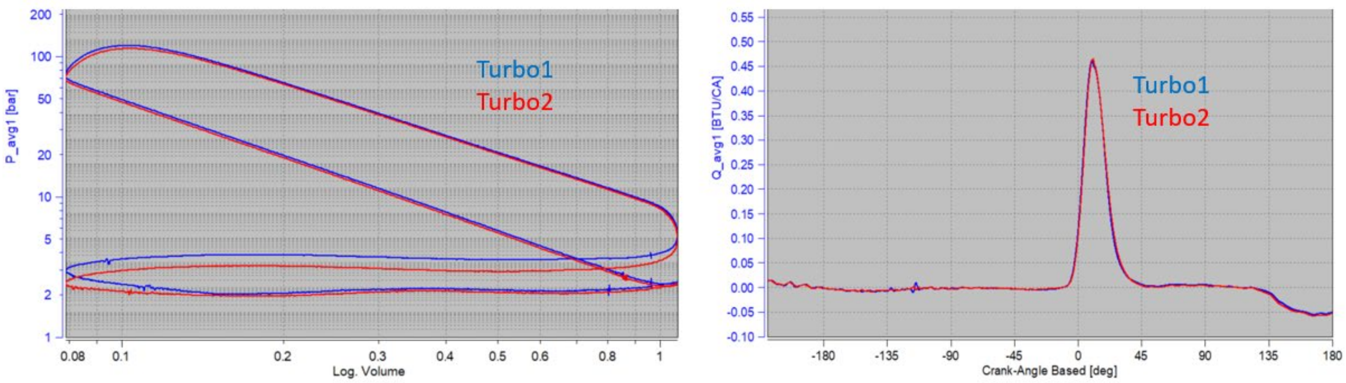
Figure 56: EGR Position, EGR Percent, and Turbine Inlet Temperature Contour Comparisons



Source: Cummins, Inc.

Comparing cylinder pressure and calculated heat release rate for an identical operating point shows a clear difference in the pumping loop portion of the map in Figure 57. This explains the improved PMEP, OCE, and brake thermal efficiency with the Turbo2 configuration.

Figure 57: Rated Power Turbo1 and Turbo2 Cylinder Pressure and Heat Release Comparisons



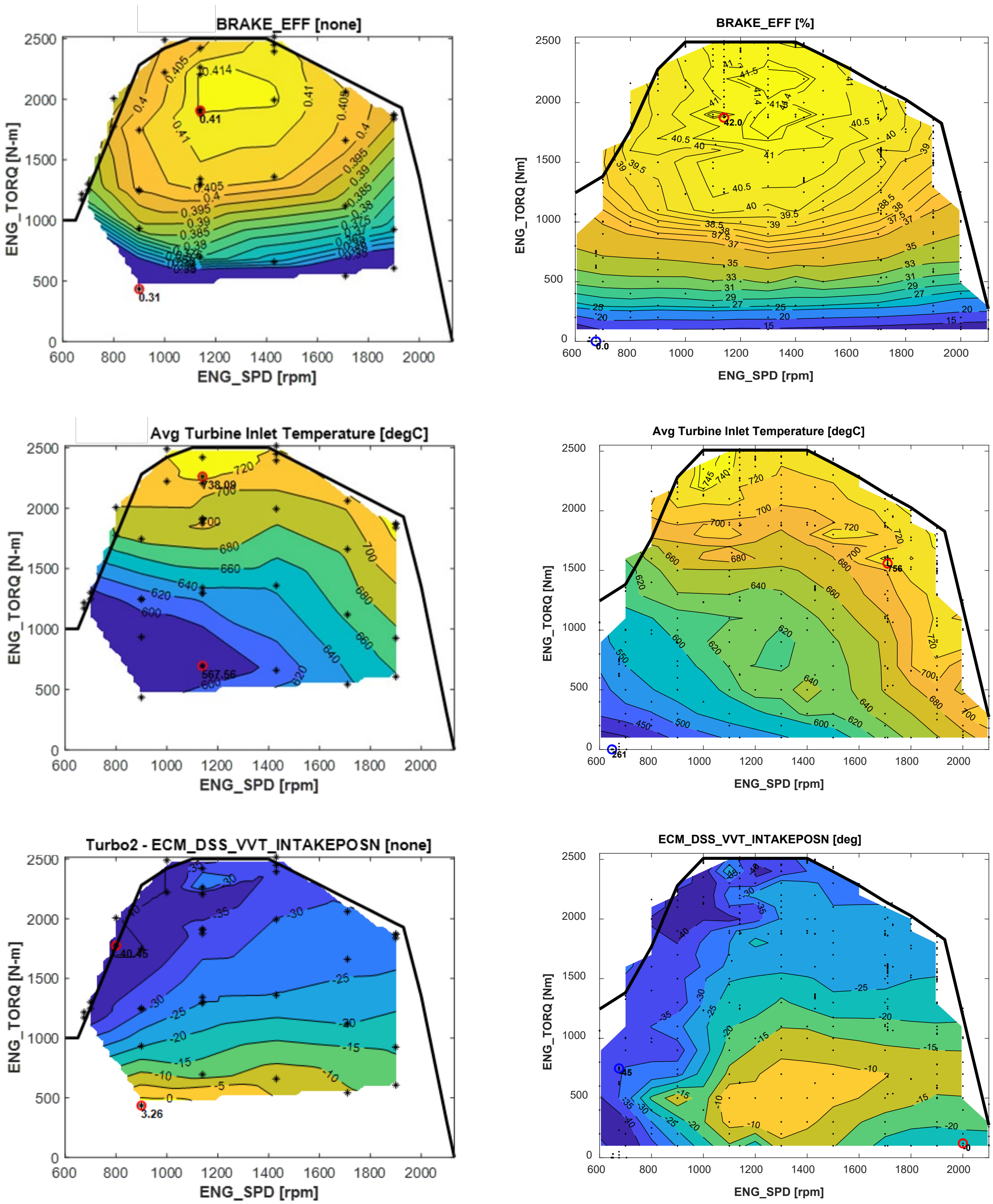
Source: Cummins, Inc.

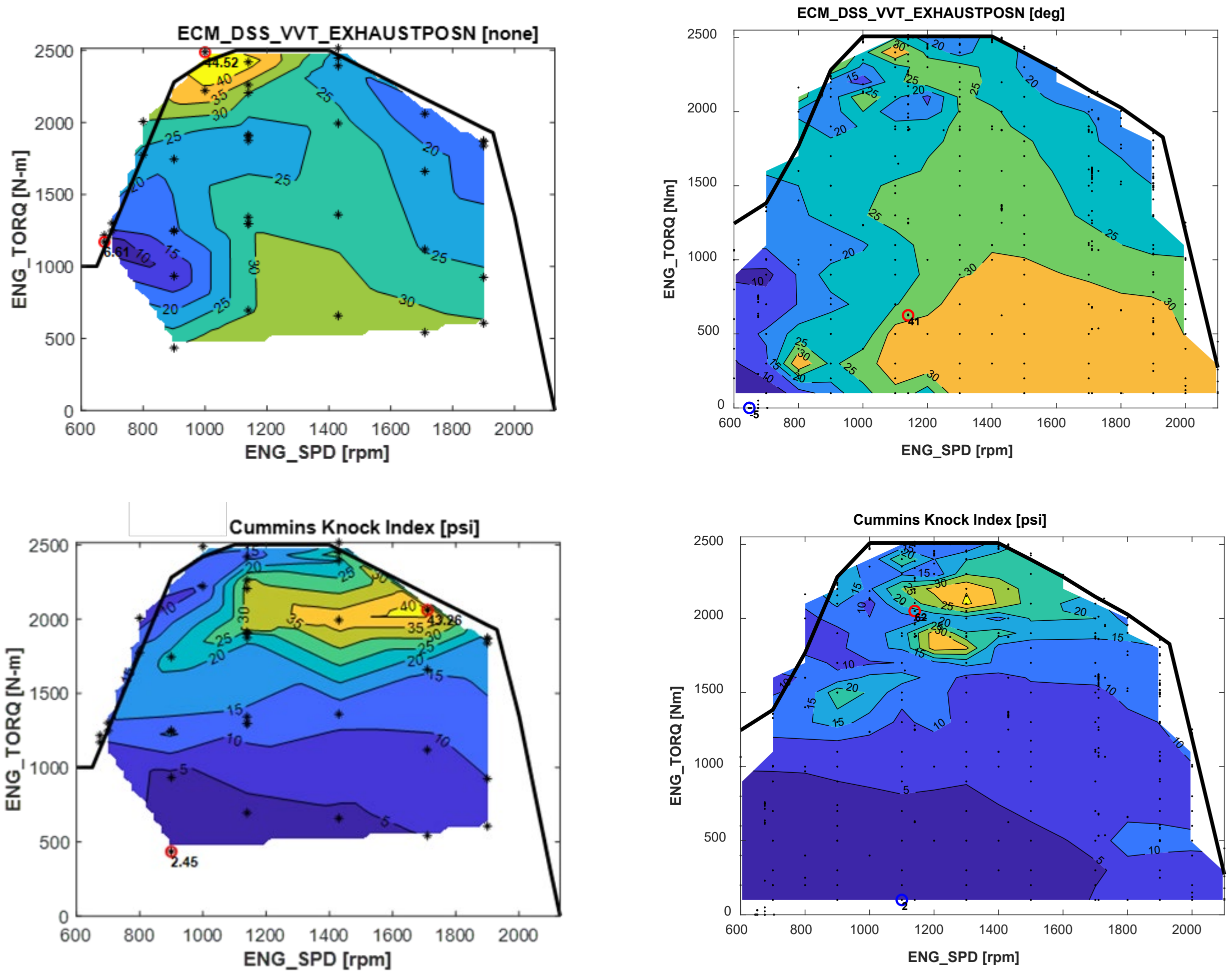
The additional control levers afforded on this architecture give additional optimization space at the expense of control system complexity. The air handling control system is managed via interactions between electronic waste gate position targeting, intake air throttle, intake phaser command, EGR valve command, and exhaust phaser command. Though it's possible to include it, the compressor recirculation valve is not activated unless surge conditions are predicted. Offline simulation has proven beneficial, although experimental refinement has been required. The project team refined the air handling targets and controls in conjunction with the fuel system, EGR rate, and combustion phasing targets (via active CA50 control) to obtain the best peak efficiency possible while maintaining satisfactory hardware constraints, including limitations on knock index and turbine inlet temperature.

One key trade-off that is present in this hardware is a balance between EGR potential and optimal intake phaser position (and turbine housing sizing); this is particularly true at low-speed high load conditions where available turbine energy is marginal. The exhaust cam phaser position is optimized to manage the balance between scavenging versus managing the amount of blow-through at high loads, and then residual management is used for part load and idle stability. While the selected cam phaser design allows for very low overlap, this is not favorable for idle combustion stability.

The contour plots in Figure 58 show a comparison of the initial data and the results of additional refinement and additional data points collected throughout the speed and load space. Some points include specific optimization investigation and off-nominal conditions, although the large bulk contours capture the capability of the system. The largest changes are near A100, where additional space is afforded once the cam phaser can be controlled to target, resulting in some efficiency improvement and demonstration of 2500 Nm at 1000 rpm without much blow-through. Also captured is a slight efficiency reduction and turbine inlet temperature at high engine speed and power, as EGR is reduced to gain margin against intermittent misfire.

Figure 58: Contour Plots Showing Efficiency Improvement

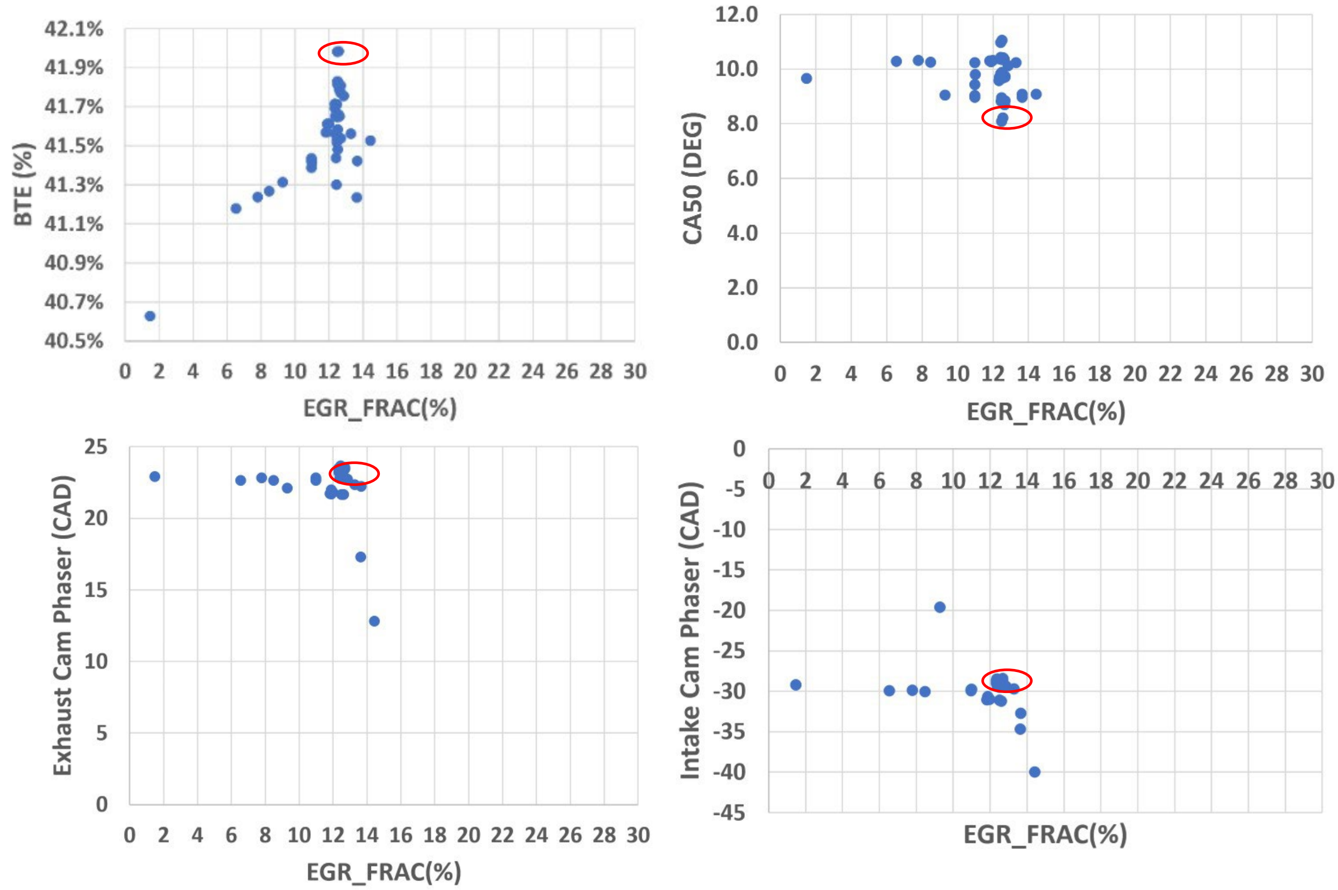




Source: Cummins, Inc.

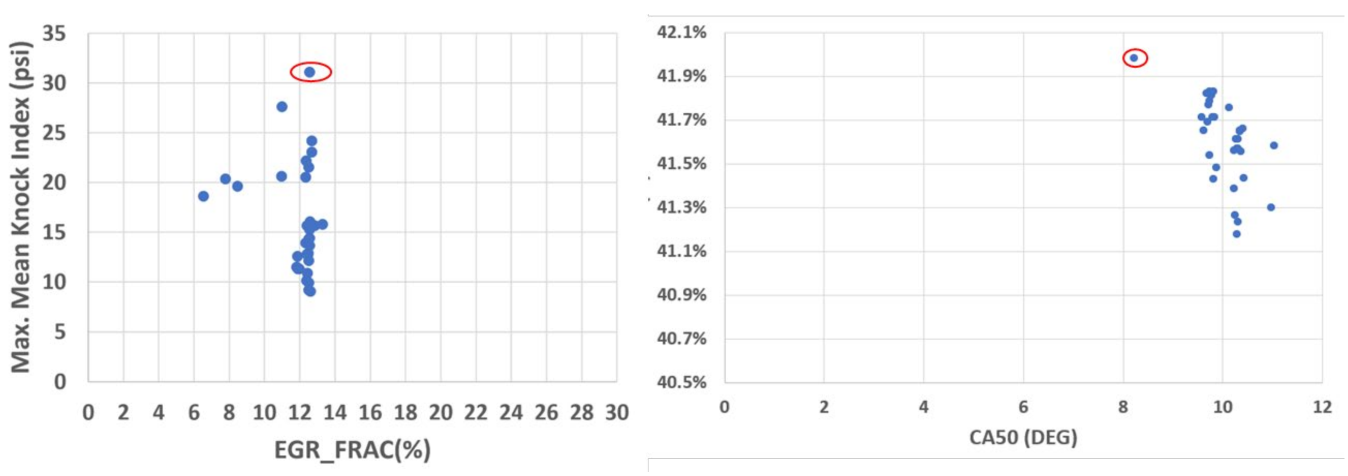
The multi-dimensional scatterplots in Figure 59 show EGR and cam phasing sweep data for peak efficiency optimization (speed 1140 rpm, load ~1875 Nm) with the final demonstration air handling hardware. It is evident that EGR is a strong lever for efficiency, having optimized both intake and exhaust cam phasing for maximum BTE. Once the elevated EGR rate could be achieved via the air handling configuration, the CA50 could be optimized to 8 degrees after TDC to achieve the best peak efficiency possible. The peak efficiency point with 42 percent BTE is highlighted in red in Figure 60.

Figure 59: BTE Impact With EGR, Cam Phasing, and CA50



Source: Cummins, Inc.

Figure 60: Mean Knock Index and CA50 Corresponding to Peak Efficiency Exploration



Source: Cummins, Inc.

Building upon stable steady-state calibrations, the emissions development started with transient controls development, including simulation studies and controls tuning in each of the control loops. The initial simulation evaluations provided base transient calibrations to allow the first transient maneuvers. Additional on-engine refinement of the air handling, fuel systems, and EGR control system delivered required updates to meet performance and regression requirements over the various cycles, including federal test procedure (FTP) and RMCSET. Once the engine was able to run these cycles, additional calibration development to maintain desired air-fuel ratio across speed/load conditions was conducted for optimum TWC conversion efficiencies.

The RMCSET and the cold and hot FTP cycles demonstrate ability to meet efficiency and emissions project requirements with the prototype components and control system. Fuel rate BTE exceeds minimum target requirements in line with cycle-based projections. Final emissions summaries are included in Table 3. Operating on the calibration, including active CA50 control via cylinder pressure feedback, with true 0 Nm idle load, the project team demonstrated excellent system out emissions performance. Where possible, the test system followed U.S. Environmental Protection Agency 1065 procedures, although this line cell does not have all of the equipment necessary that would be required for an official emissions certification. The aftertreatment system is of the intended formulation and construction but with relatively low hours from prior de-greening and engineering development. Raw emissions, including CO_2 specifically, are presented in the table below and do not include CO_2 equivalent calculations or production variation end of useful life assessment. Idle speed is targeting 650 rpm and all regression requirements are met.

Table 3: RMCSET and FTP Efficiency and Emissions Summaries

RMCSET(g/bhp-hr)	2023 ISX12N	FEL/Std	15L NG
CO	0.36	15.5	0.03
NMHC	0.001	0.14	0.015
CH4	0.04	0.43	0.01
NOx	0	0.02	0.004
PM	0.0003	0.01	NA
CO2	414	440	373
N2O	0	0.1	0
Avg NH3	N/A	N/A	1.5
Run Number	N/A	N/A	981096
Fuel Meter BTE [%]	35.2	N/A	40.2

FTP(g/bhp/hr)	2023 ISX12N		FEL/Std	15L NG	
	Cold FTP	Warm FTP	CHET	Cold FTP	Warm FTP
CO	2.21	0.73	15.5	0.66	0.48
NMHC	0.03	0	0.14	0.04	0.02
CH4	0.45	0.12	0.50	0.21	0.06
NOx	0.07	0	0.02	0.13	0.007
PM	0.0015	0.0044	0.01	NA	NA
CO2	544	495	531	396	386
N2O	0.0099	0	0.1	0.006	0
Avg NH3	N/A	N/A	N/A	22.4	28.6
Run Number	N/A	N/A	N/A	984146	984147
Fuel Meter BTE [%]	N/A	30.2	N/A	34.5	35.3

Source: Cummins, Inc.

CHAPTER 4: Conclusion

There is a direct interaction between engine efficiency, capability, and power density demands. Improvements in SI NG engine technology and hardware designs, such as those demonstrated in this project, allow for more aggressive tuning (higher compression ratio, combustion phasing, and brake thermal efficiency) and/or increased power output with similar boundary conditions and limits imposed. The tumble-based combustion system and balanced port designs allow for efficiency enhancement and higher power output, although knock limited behavior remains a challenge for higher BMEP. Additional modeling refinement based on the experimental results captured here may be considered to help uncover additional enablers; continued investigation and opportunities would require additional resources.

The following are the key outcomes of this research work:

- A 11.4-percent improvement in peak BTE over current product ISX12N engine.
- A 14-percent improvement in cycle avg RMCSET BTE over current product ISX12N engine.
- A 21-percent improvement in cycle avg FTP BTE over current product ISX12N engine.
- The above improvements while maintaining ultra-low emissions at similar levels to current product ISX12N, including near-zero NOx.
- With these efficiency improvements, the engine platform can meet the 2027 EPA/CARB vocational and tractor criteria CO2 regulations as well as the 2027 U.S. Environmental Protection Agency Phase II Vehicle Greenhouse Gas Emissions Model stringencies.
- Platform commonality would allow for reduced engine system costs of up to 23 percent compared to the current product ISX12N, with an additional 8-percent potential reduction expected from a lower cost aftertreatment system.
- Achieved TRL 6 with a path forward for commercial product development.

A 15-L NG engine with improved efficiency, ultra-low emissions, and performance similar to that of a diesel engine enables opportunities for broader NG adoption in the heavy-duty vehicle market, with improved total cost of ownership and payback periods for fleets. The 15-L NG engine can also enable adoption in heavy-duty line-haul applications, which, to date, has been a challenge for the smaller ISX12N engine. Cummins' internal analysis forecasts an increase in the North America NG heavy-duty market share from 4 percent to 5 percent in 2024, up to as high as 17 percent by 2030. Biomethane availability, usage, and infrastructure development are some of the critical factors contributing to this projected increase in adoption rates. Cummins launched a big bore 15-L NG engine (X15N) in 2024 with a technology similar to that of today's ISX12N. The technology demonstrated in this project will go through a development cycle, with tentative plans for a 2027 launch.

GLOSSARY AND LIST OF ACRONYMS

Term	Definition
3D	three dimensional
bar	metric unit of pressure, equal to 100,000 pascals
BMEP	brake mean effective pressure
bTDC	before top dead center
BTE	brake thermal efficiency
CA	crank angle
CA50	crank angle at which 50 percent mass fraction burned
CAPEX	capital expense
CFD	computational fluid dynamics
CO_2	carbon dioxide
CR	compression ratio
CWI	Cummins Westport, Inc.
EGR	exhaust gas recirculation
FMEP	friction mean effective pressure
FOS	fatigue life parameter
FTP	federal test procedure
G/BHP-HR	grams per brake horsepower per hour
GHG	greenhouse gas
kpa	kilopascal
L	liter
LWJ	lower water jacket
NG	natural gas
MM	millimeters
NGV	natural gas vehicle
NOX	oxides of nitrogen
NM	Newton meter
OCE	open cycle efficiency
OPEX	operating expense
PCP	peak cylinder pressure
PFI	port fuel injection
PMEP	pumping mean effective pressure

Term	Definition
RMCSET	ramped mode cycle supplemental emissions test
rpm	revolutions per minute
SCE	single cylinder engine
SI	spark-ignited
TCO	total cost of ownership
TKE	turbulent kinetic energy
TMF	thermo-mechanical fatigue
TRL	technology readiness level
TWC	three-way catalyst
VMT	vehicle miles traveled
VVA	variable valve actuation

Project Deliverables

The following project deliverables, including interim project reports, are available upon request by submitting an email to pubs@energy.ca.gov.

- Kickoff Meeting Presentation
- Quarter 1 Webinar and Progress Report
- Open and Closed Cycle Engine Test Hardware Design Report
- System Design Report
- Quarter 2 Webinar and Progress Report
- Quarter 3 Webinar and Progress Report
- Presentation on Closed Cycle Test Engine First Start
- Quarter 4 Webinar and Progress Report
- Report Regarding Engine Torque Curve, Displacement and Architecture
- Quarter 5 Webinar and Progress Report
- Quarter 6 Webinar and Progress Report
- Mid-Project Presentation
- Quarter 7 Webinar and Progress Report
- Report Summarizing Critical Parts Design
- Quarter 8 Webinar and Progress Report
- Multi-Cylinder Engine Design Report
- Quarter 9 Webinar and Progress Report
- Quarter 10 Webinar and Progress Report
- Multi-Cylinder Engine Build Complete and Documented to NREL
- Quarter 11 Webinar and Progress Report
- Quarter 12 Webinar and Progress Report
- Presentation/Report for Test Results and Hardware Recommendation
- Report Containing Steady State Engine Maps
- Quarter 13 Webinar and Progress Report and Presentation on Hardware Recommendation Test Results
- Quarter 14 Webinar and Progress Report
- Efficiency Target Report
- Emissions Target Report
- Final Project Presentation
- Draft Project Report
- Final Project Wrap-Up Meeting