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DEVELOPMENT AND DEMONSTRATION OF ADVANCED ENGINE AND VEHICLE TECHNOLOGIES FOR CLASS 8 HEAVY DUTY VEHICLE (SUPERTRUCK II)

DOE Program Award Number: DE-EE0008265
Award Type: Cooperative Research and Development Agreement

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1. Executive Summary

The PACCAR SuperTruck II program has completed the planned technical work on the project. This work includes the planned vehicle demonstrator showing a greater than 100% improvement in vehicle freight efficiency (on a ton-mile-per-gallon basis) relative to a 2009 baseline, and an engine system demonstration achieving a greater than 55% engine brake thermal efficiency at a 65-mph cruise point on a dynamometer. Both program targets were exceeded by PACCAR with robust margins. Finally, PACCAR used technologies that have a possible route to production, with no specialty or unrealistic solutions proposed.

PACCAR followed a structured approach, evaluating new technology concepts early in the program with state-of the art simulation tools supported by validation on fundamental test-rigs, in an iterative approach. As a result, the team was able to reach high maturity levels on new technologies early, and as a result minimal additional optimizations to the final engine and demonstrator vehicle were required at the end of the program to reach the program goals.

The engine system included a high efficiency diesel engine integrated with a state-of-the-art waste heat recovery system capable of meeting the upcoming stringent emissions standards. The project focused on developing technologies with high commercialization potential, keeping complexity, build constraints, weight, and return-on-investment in mind. As a result, many of the technologies developed are now included in PACCAR's production development programs. The efficiency achieved with the engine and waste heat recovery system combined is 55.7%, with a contribution of the engine alone of 51%.

The design approach of the Kenworth demonstrator vehicle was aggressive, implementing many new technologies which support not only the next generation diesel powered vehicles but also the next generation zero-emission vehicles. The demonstrator sets the new baseline for aerodynamic drag reduction with a revolutionary cab over engine design including a center driver position. The powertrain includes a 48V mild-hybrid diesel electric architecture which maximizes fuel efficiency improvement while minimizing additional cost and complexity at the same time. These technologies together with many other efficiency and weight saving features resulted in a freight efficiency improvement of 110% on our traditional fuel efficiency testing route up-to 136% under steady state 65 mph cruise point conditions.

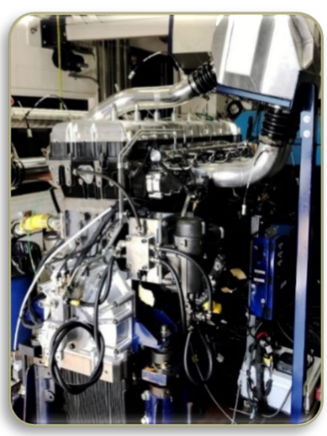

Figure 1: PACCAR MX SuperTruck II Engine (Left) and Kenworth SuperTruck II Vehicle (Right)

2. Acknowledgements

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The author of this report greatly acknowledges the support of the Department of Energy and the support of Siddiq Khan Ph.D. (Technology Manager) and Ralph Nine (Project Officer) throughout this program. Your continued support, especially during the COVID pandemic, greatly contributed to its success in meeting the goals of the Department of Energy and the internal goals for PACCAR.

PACCAR wants to thank the State of Washington Congressional representative Rick Larsen, along with Senators Patty Murray and Maria Cantwell, who supported us during the program application.

The author of this report wants to thank the (sub-recipient) partners under this program who contributed to its success: Eaton, AVL, Ohio State University, the National Renewable Energy Laboratory (NREL), and United Parcel Service (UPS).

3. List of Acronyms

AFR	Air-Fuel Ratio
AMT	Automated Manual Transmission
APU	Auxiliary Power Unit
BiW	Body in White
BMS	Battery Management System
BTE	Brake Thermal Efficiency
COE	Cab Over Engine
DEF	Diesel Exhaust Fluid
DOE	Department of Energy
DoE	Design of experiments
DOT	Department of Transportation
eRAD	Electrically Regenerative Accessory Drive
FEAD	Front End Accessory Drive
GCI	Gasoline Compression Ignition
GEM cycle	Greenhouse-Gas Emission Model
GITE	Gross Indicated Thermal Efficiency
HDCC	Heavy-Duty Composite Cycle
HVAC	Heating, Ventilation, and Air Conditioning
LIVC	Late Intake Valve Closing
MDC	Motor Drive Converter
NREL	National Renewable Energy Laboratory
ORC	Organic Rankine Cycle
PCP	Peak Cylinder Pressure
PTC	PACCAR Technical Center
PTO	Power Take-Off
SCE	Single Cylinder Engine
SFFFA	Set Far Forward Front Axle
TRL	Technology Readiness Level
TMY	Typical Meteorological Year
UPS	United Parcel Service
VMT	Vehicle Miles Traveled
WERC	Wisconsin Engine Research Consultants
WHR	Waste Heat Recovery

4. Introduction

In 2016, the U.S. Department of Energy (DOE) inspired American truck manufacturers through funding the SuperTruck II program to demonstrate a system of new technologies with the primary benefits of improving freight efficiency energy use, reducing vehicle operating costs and improving environmental sustainability. The opportunity to harness the creativity of OEMs and suppliers to push the envelope of vehicle design was fully embraced by PACCAR, who put together a team of leading industry and research partners.

The SuperTruck II program required that OEMs' cost share must be at least 50% of the total allowable costs for demonstration projects. The period of performance of the PACCAR SuperTruck II program was 60 months with an extension authorized later due to challenges from the COVID pandemic.

Where SuperTruck I's primary goal was to achieve more than a 50% freight ton mile per gallon improvement, SuperTruck II had more aggressive goals. As the DOE Notice of Intent states the objective of the planned FOA is to research, develop, and demonstrate a class 8 long-haul or regional haul truck that uses conventional fuels, can meet prevailing emission standards and applicable safety and regulatory requirements and can achieve the following goals:

- A greater than 100% improvement in vehicle freight efficiency (on a ton-mile-per-gallon basis) relative to a 2009 baseline.
- A greater than or equal to 55% engine brake thermal efficiency demonstrated in an operational engine at a 65-mph cruise point on a dynamometer.
- Develop technologies that are commercially cost effective in terms of simple payback.

Over the period 2017 through 2023, PACCAR succeeded in innovating an engine with a waste-heat recovery system that demonstrated > 55% brake thermal efficiency and a complete tractor-trailer system capable of exceeding freight efficiency performance improvements in excess of 100% versus each team's model year 2009 baseline diesel tractors with standard 53' dry van box trailers.

These accomplishments created a watershed of practical engineering and operational information to assist in developing production tractors and trailers to meet increasingly demanding performance needs of the North American freight industry. These successes were in no small part due to the ingenuity and persistence of each of the program partners: PACCAR (prime), Eaton, AVL, OSU, NREL, and UPS.

The fundamental basis of these SuperTruck II demonstrations is that a systems-engineering perspective encompassing the entire tractor-trailer as a single performing unit, can yield significantly better performance improvement than more piecemeal approaches looking at optimizing individual systems.

The intent of this report is to highlight the innovative technologies developed and demonstrated by PACCAR, and how they may benefit future production freight vehicles. Many of the concepts are universally applicable to new vehicles, irrespective of future powertrain choices, others are specific to internal combustion engines.

5. Program Schedule & Milestones

The PACCAR SuperTruck II program followed a cadence of analyze-design-fabricate-test activities. The design and engineering study considered a range of engine and powertrain concepts and solidified the performance and interactions of the proposed technologies. The study led to component and system design and fabrication. Upon build, the components were integrated into sub-systems, validated, and subsequently integrated into the full vehicle for optimization and final demonstration. The project was completed over five budget periods as outlined in the following.

Budget Period 1: Design and Engineering Study

Phase 1 will focus on in-depth analysis of two powertrain configurations: 1) the main technical path involves a displacement-on-demand engine with moderate hybridization, and 2) evaluate a concept involving aggressive engine downsizing combined with full hybridization. Advanced simulation tools will be employed to evaluate a range of configurations to minimize engine efficiency losses resulting from combustion, air systems, friction and parasitics. Various hybrid concepts will be evaluated and assessed along with advanced predictive features over relevant drive and duty cycles. A preliminary hybrid energy management strategy will be developed. The baseline vehicle will be tested to provide reference data. Extensive vehicle aerodynamic and rolling resistance improvements and lightweighting will be analyzed with state-of-the-art modeling tools. The key Phase 1 deliverable will be an engine, powertrain and vehicle concept that has the best potential of meeting project efficiency goals while at the same time representing a commercially viable solution.

Milestone	Type	Description
BTE Analysis Completed	Technical	Engineering Analysis of 55% BTE Engine Paths Completed
BTE Design Selected	Technical	55% BTE Engine Design Selected
Powertrain System Selected	Technical	Powertrain System for 100% Freight Efficiency Improvement Selected

Table 1: Budget Period 1 Milestones

Budget Period 2: Design and Start of Prototype Fabrication

Phase 2 will focus on materializing the concepts selected in Phase 1 with component design, build and Hardware-In-the-Loop (HIL) testing. Engine component and sub-systems designs, including Waste Heat Recovery (WHR) will be developed and prototype drawings released. The hybrid and electrification voltage architecture will be selected, and necessary components will be specified and sourced. Select powertrain electrical and mechanical components will be designed and fabricated. Aerodynamics components, such as hoods, mirrors and fairings will be designed and optimized along with select lightweight structures. The potential for efficiency improvements through reduced rolling resistance will also be quantified. The designs will be used to build prototypes to undergo rig, test cell and vehicle testing in future phases. The main deliverables of this phase are designs and test plans.

Milestone	Type	Description
Engine Component	Technical	Design of Internal and External Engine Components with Long Lead Times is Complete
Powertrain Components Selected	Technical	Design of Electrified Powertrain Components is Complete and Components are Selected for Fabrication
Mule Vehicle Tested	Technical	Mule Vehicle is Designed, Assembled and Tested

Table 2: Budget Period 2 Milestones

Budget Period 3: Completion of Component Build, Testing and Validation

Phase 3 will focus on build and test of the prototype engine, WHR and powertrain system designs developed on component rigs, engine- and powertrain test cells and on vehicle. Control strategies will be developed at the engine and powertrain and the hybrid energy management levels. Electrical motors, accessories and power electronics will be rig tested. The WHR system will be integrated to an engine in a test cell. Mule versions of the truck and trailer will be equipped with a suite of technologies aimed at reducing aerodynamic drag, rolling resistance and weight. The main deliverables of this phase are finalizing the prototype builds, developing the high-level controls and initial feasibility testing.

Milestone	Type	Description
Engine Components Fabrication Complete	Technical	Final Internal and External Engine Components are Fabricated
Powertrain Components Fabrication Complete	Technical	Final Electrified Powertrain Components are Fabricated
SuperTruck II Tractor Component Designs Frozen	Technical	Design is Frozen for Components of the SuperTruck II Tractor

Table 3: Budget Period 3 Milestones

Budget Period 4: Powertrain Testing, Build of SuperTruck II Vehicle, SuperTruck II Field Test:

Phase 4 will focus on engine and powertrain testing to finalize controls and calibration of the overall system. This involves final integration of engine, transmission, axles, energy storage system, hybrid energy management and advanced predictive features. Testing of the WHR integrated to the engine will also start in this phase. The freight efficiency contribution from the powertrain will be evaluated over all relevant drive/duty cycles and assessed against the project goals. Select aero- and rolling resistance improvements and lightweighting concepts will be carried over from the mule vehicle to the SuperTruck II vehicle. The engine, powertrain and vehicle sub-systems will be integrated into the SuperTruck II vehicle. The truck will undergo initial drivability and fuel economy testing. Key deliverables of this phase are preliminary demonstrations of freight efficiency improvements resulting from engine, powertrain and vehicle optimization.

Milestone	Type	Description
SuperTruck II Vehicle Build is Complete	Technical	SuperTruck II Vehicle is Built
Powertrain Demonstration Complete	Technical	Powertrain for SuperTruck II Demonstrates viability of greater than 100% Improvement in Freight Efficiency in Powertrain Test Cell
55% BTE Demonstration Complete	Technical	Demonstrated 55% BTE Engine at 65mph Road-Load Point in an Engine Test Cell and Hot FTP and SET Emissions
Simple Payback Demonstrated	Technical	ROI analyses show that 50% of the new technologies at TRL level 3 and above have a Simple 3-Year Payback

Table 4: Budget Period 4 Milestones

Budget Period 5: 55% BTE Engine and SuperTruck II Freight Efficiency Demonstration

Phase 5 will focus on demonstration of the SuperTruck II vehicle freight efficiency target. Engine and WHR hardware and controls concept will be finalized, and the system will be calibrated to demonstrate the 55% BTE goal in a 65-mph cruise point on a dynamometer. Truck freight efficiency and fuel economy tests will be carried out over all relevant drive/duty cycles and compared to the baseline measurements as well as analytical models. Several iterations will be performed to ensure adequate performance and commercial feasibility of the SuperTruck II vehicle. Key deliverables out of this phase are demonstrations of engine BTE of 55% and vehicle freight efficiency improvement of greater than 100% over the 2009 baseline as well as final documentation and project close-out.

Milestone	Type	Description
SuperTruck II Vehicle Field Test Complete	Technical	SuperTruck II Vehicle is Field Tested
100% Freight Efficiency Improvement Demonstration Complete	Technical	SuperTruck II Demonstrates > 100% Improvement in Freight Efficiency in Road Tests
SuperTruck II ROI Analysis Complete	Technical	Complete ROI analyses for all new technologies incorporated into the SuperTruck

Table 5: Budget Period 5 Milestones

6. Engine Definition and Validation Approach

Operating Point Selection Method for 55% BTE Engine Testbed Validation

PACCAR selected an engine operating point relevant to real-world class-8 heavy-duty vehicle engine operating conditions to demonstrate 55% BTE. Using the various resources available that provide insight into real-world engine operating conditions, there are two ways to calculate the road load for a given drive cycle: (1) time-based or (2) fuel-weighted.

Using method (1), the road load is calculated using a histogram of the time spent at a given engine power/torque and engine speed and therefore the road load point is represented by an average power versus time. However, this approach misrepresents the average operating point given the following observations: considering the engine fuel rate at idle, compared to operation at maximum engine load, which has a fuel rate more than an order of magnitude greater. A short amount of time spent at maximum load, with a much higher fuel rate, is far more significant in terms of overall fuel used (and freight efficiency impact) than is a longer time spent at idle. Averaging engine load by time only would erroneously give equal weighting to the time spent at idle, whereas time spent at maximum load has the greatest leverage on the overall freight efficiency. A more accurately approach (2) to define the SuperTruck-II engine demonstration operating point, is to define the powertrain load in terms of a fuel-weighted value. Rather than simply averaging power over time, the engine power at each timestep is multiplied by the corresponding fuel rate, and the resulting total is then divided by total fuel used. The average operating point is still time averaged, but is also weighted by amount of fuel used, to reflect which operating points have the greatest significance in terms of overall fuel used and the most leverage on the overall freight efficiency. For example, when a truck is equipped with the coasting-in-neutral feature, the engine may spend up to 15% of the cycle time coasting in neutral, but it will burn less than 10% of the cycle fuel while coasting. This is the result of the reduced fuel flow rate under engine idling conditions.

For the definition of the engine operating point both PACCAR drive cycle analysis and connected fleet data analysis has been used. The various drive-cycles are discussed in more detail in the freight efficiency improvement drive-cycle definition section.

Connected Truck Analysis

All modern PACCAR trucks come equipped with remote diagnostics that enable vehicle diagnostic codes to be transmitted to a remote database. Data is acquired at every engine start-up, fault-setting event, engine shutdown and once-hourly. From the PACCAR connected truck database, the data was filtered to use model year 2016 and 2017 trucks equipped with PACCAR MX engines only. Only prime data points (not start-up, shutdown, or fault set) were selected for this analysis, resulting in >5.6M data points to analyze.

The data was binned by engine speed and torque and the average power of each bin was multiplied by the frequency an engine operates in that bin. Assuming that engine power and fuel rate are analogous (in order to produce more power an engine has to burn more fuel), analysis of engine power can be used to identify fuel usage. This analysis resulted in the power weighted average engine power distribution shown in Figure 2. On each box, the central bar is the median with the edges of the box showing 25th and 75th percentile while the whiskers extend to cover 99.9% of the population. This figure shows that the power-weighted average power of PACCAR connected trucks is close to 200 kW.

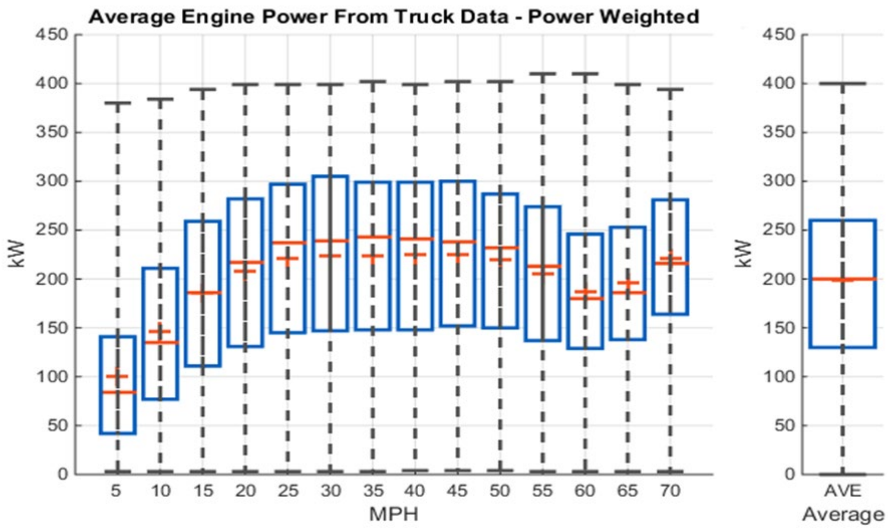


Figure 2: Fuel Averaged Engine Power for Connected Trucks

Figure 3 shows the iso-contour plot of the MY17 PACCAR MX-11 engine brake thermal efficiency, with fuel averaged road load points superimposed for a variety of drive cycles used by PACCAR. This figure shows that:

- The fuel weighted average power for the overall HDCC cycle is approximately 165 kW. The average power values for the I-5 routes at 55 and 60 mph (which are sub-cycles of the HDCC) are very close to that of the HDCC.
- The fuel weighted average power over the I-5 route at 65 mph is close to 200 kW. This aligns well with the average power calculated for customer trucks equipped with telemetry (connected trucks), as shown in Figure 2.

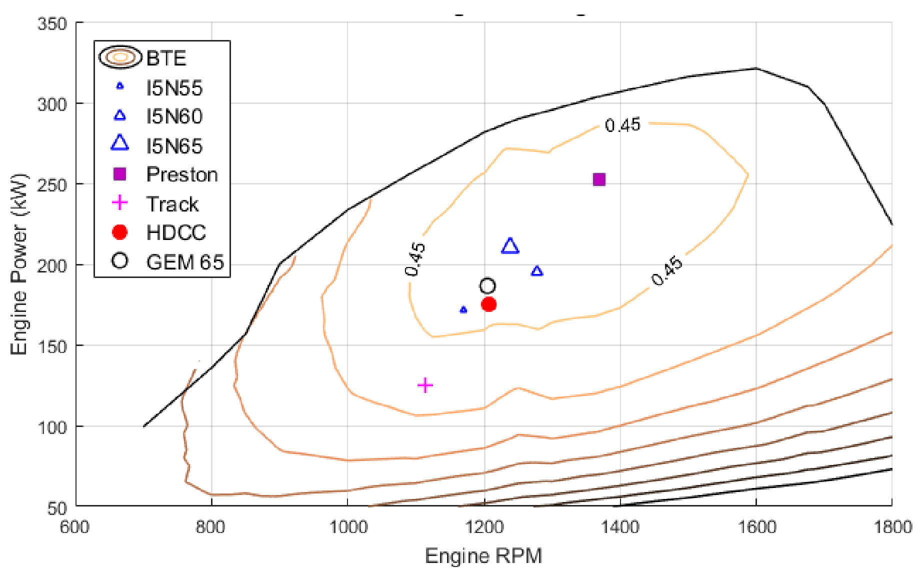


Figure 3: BTE Iso-Contour Map with Selected Drive Cycles

This analysis showed that, depending on vehicle speed, the fuel weighted average power lies between 160 to 200 kW. However, the fuel weighted power data is based on current production vehicles, not the SuperTruck II vehicle. As a consequence, the fuel weighted power requirement for SuperTruck II will change with the implementation of different technologies. For example, the implementation of engine off coasting will reduce the fuel burn at very low load, resulting in an increase in fuel weighted average power. Conversely, the reduction in road load associated with reduced aerodynamic drag and rolling resistance will result in reduced power at cruise conditions resulting in a reduction of fuel weighted average power. The final value of road load under cruise conditions for the SuperTruck II vehicle will differ from current production vehicles but cannot be established until the suite of technologies has been established. As a consequence, the selected power for the cruise condition is based on the available data from current production vehicles and reads 180 kW.

Emission Cycles

One of the contractual milestones for budget period 5 reads: "Demonstrate 55% BTE Engine at 65mph Road-Load Point in an Engine Test Cell and Hot FTP and SET Emissions". Therefore, the standard FTP and SET cycles must be defined. Since PACCAR wanted to develop the SuperTruck II engine not only being capable of demonstrating high efficiency in one operating point, but also demonstrate operating capabilities equal to its current production engines, the FTP and SET cycle point have been derived from the 320kW PACCAR MX-11 MY17 production engine which forms the baseline for the PACCAR MX SuperTruck II demonstration engine. Following the standard criteria for emission drive-cycles, the SET and FTP demonstration cycles for the SuperTruck II engine demonstration are shown in Figure 4, and Figure 5 respectively.

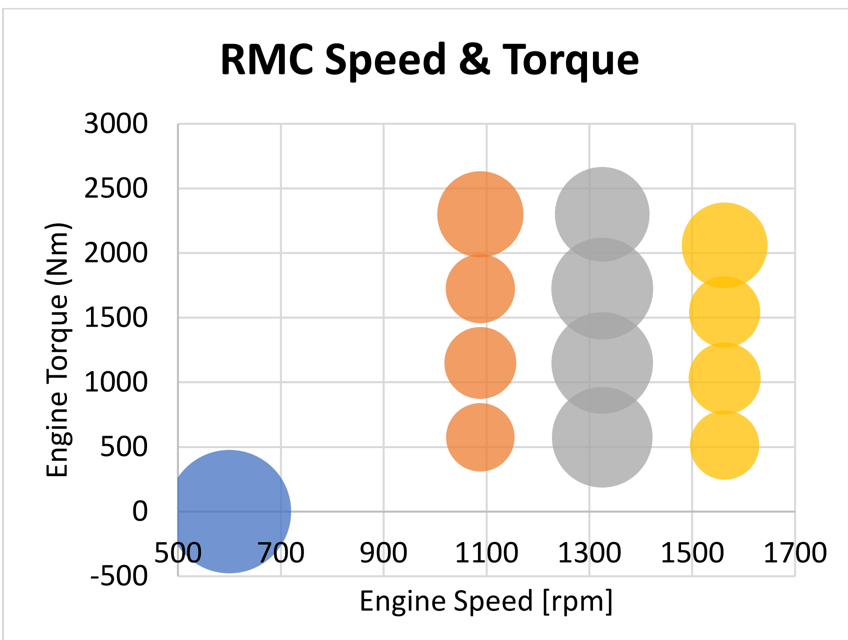


Figure 4: RMC 13 mode operating points for the PACCAR MX SuperTruck II engine

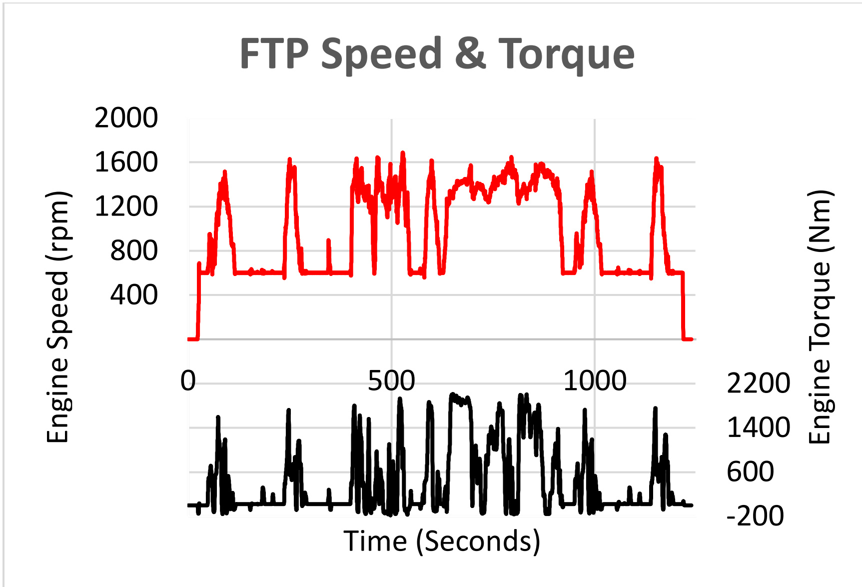


Figure 5: FTP cycle for the MX SuperTruck II engine

Future diesel engines not only require high efficiency, but also need to be emission compliant from which the upcoming NOx regulations are the most challenging. The PACCAR team decided to leverage the SuperTruck II program and challenge themselves to meet not just the current NOx emission regulation standards but to implement the most stringent upcoming regulation which is the CARB EPA 2027 Ultra Low NOx as the constraint. In practice this means a tailpipe-out NOx reduction of 90% compared to the existing regulatory landscape, demonstrating a 0.02 [g-bhp-hr] technology solution.

Engine Technologies Selection Approach

For the 55% BTE engine project the required activities have been split up in the following phases;

- Budget period 1: simulation (0D, 1D and 3D) and design for new components.
- Budget period 2: component testing on single cylinder engine, with focus on combustion.
- Budget period 3: full 6-cylinder engine testing with optimized components from SCE.
- Budget period 4: engine optimization / calibration.

Figure 6 shows the roadmap to achieve the project objective of 55% BTE for the engine. On the left side of the graph, we show the efficiency losses, relative to the ideal cycle efficiency, that result in today's state of the art MX engine peak of 47% BTE. On the right side of the graph is the roadmap of technologies under investigation to meet the 55% BTE goal.

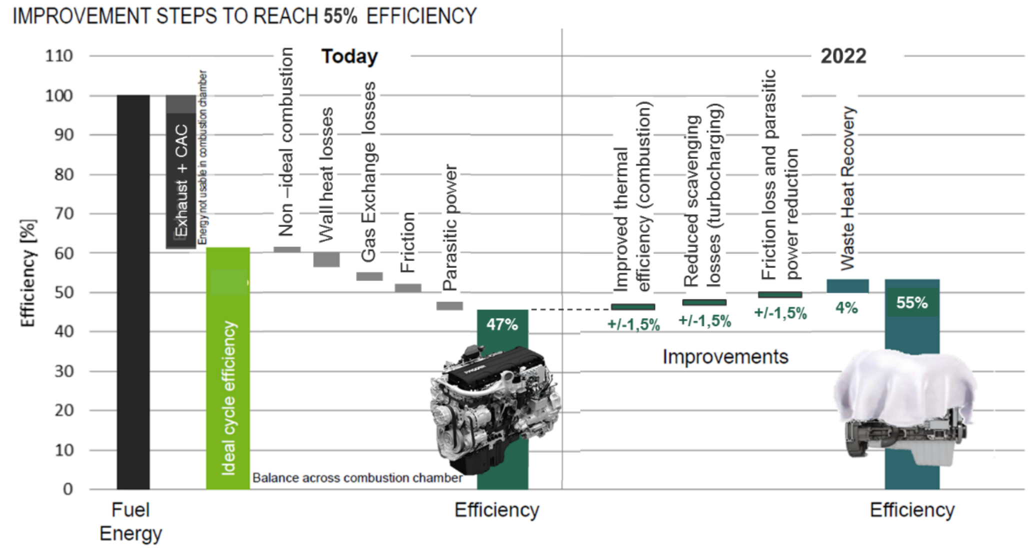


Figure 6: Roadmap to Meet Brake Thermal Efficiency Goal

To achieve the SuperTruck II program engine efficiency goal of 55% BTE, adoption of new engine technologies and the addition of a Waste Heat Recovery (WHR) system is required. To find the right balance between optimum BTE improvement and expedite development efforts of technologies with high future commercialization potential, the team categorized potential solutions into three engine concept bins at an early stage of the project, which are summarized in the overview shown in Figure 7.

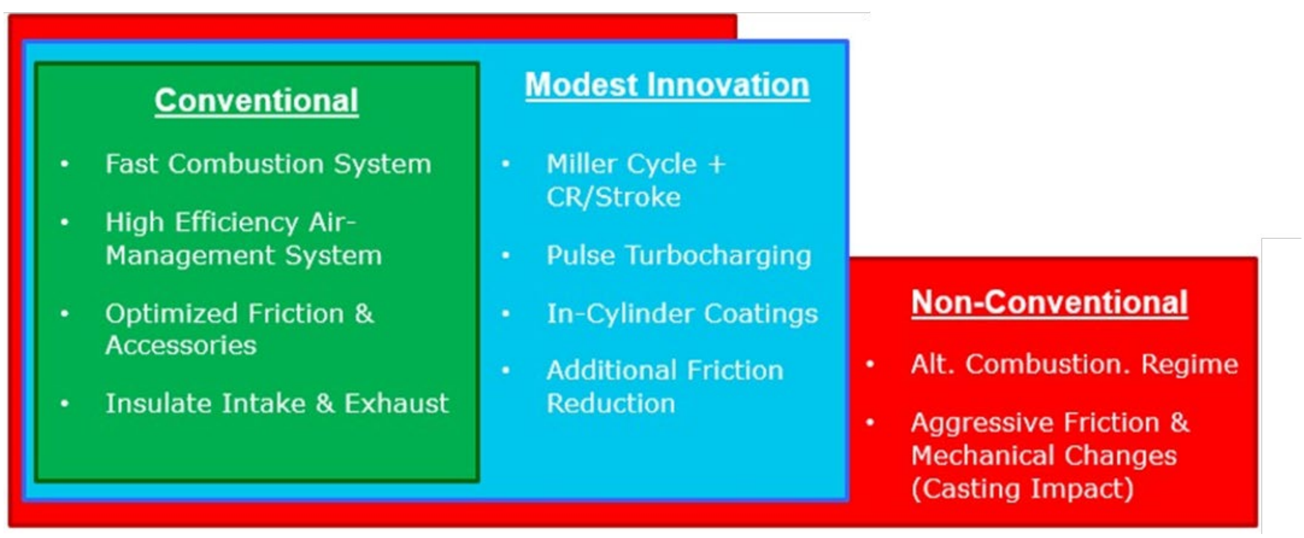


Figure 7: Conventional, Modest Innovation and non-conventional SuperTruck II engine technology concepts

Detailed simulations revealed that if implementing only the conventional concepts, the 55% BTE target will not be achieved. Therefore, the team fully adopted investigating the modest innovation design concepts and explored alternative combustion regimes from the non-conventional concepts in addition. New technologies that would result in a complete re-design of the current PACCAR MX-11 engine block were omitted due to the major manufacturing impact (in terms of required funding) this solution would have on current production lines.

All key development activities related to the conventional and modest innovation concepts explored under this program are included in this report starting with simulation-based performance potential analysis, concept selection, proof of concept testing, and final validation testing activities. Due to the complexity of technologies developed and their highly coupled interactions, the PACCAR SuperTruck II program followed a structured approach in evaluating new technology concepts early in the program with state-of the art simulation tools supported by validation on fundamental test-rigs, in an iterative approach. As a result, PACCAR was able to reach high-maturity levels on new technologies early, and no additional optimizations to the final engine and WHR system were required at the end of the program to reach the program goals. A summary of the implemented development approach is shown in Figure 8.

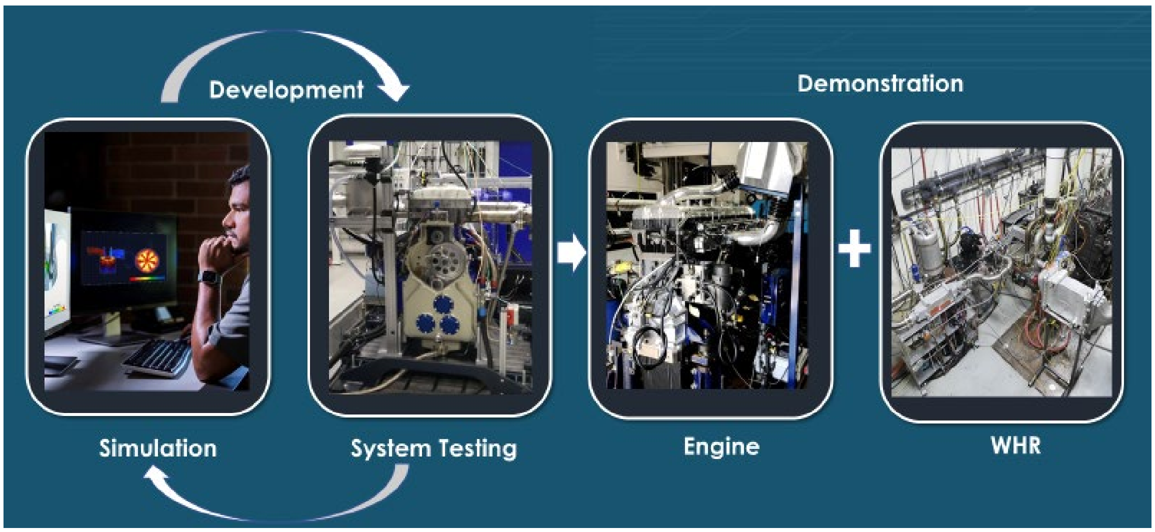


Figure 8: Development process PACCAR MX SuperTruck II 55% BTE engine demonstration

As stated above, the PACCAR 11-liter MX-11 was selected as the platform for engine development. Several factors were considered in this selection including performance, weight, efficiency, and architecture. With a peak power of 430 horsepower, the base engine will easily meet the performance targets, such as acceleration and gradeability. In addition, the engine is over 600 lbs. lighter than the 15L engine used in the 2009 baseline truck. In the first year as part of the simulation activities the team performed an in-depth study on the theoretical losses in the high-pressure and low-pressure thermodynamic cycles. The objective of this study was to have a better understanding of the achievable improvements. Additionally, with a better understanding of the losses and their sources, the BTE improvement plan can be better targeted.

For the sizing of the engine an analysis was carried out for the optimum brake mean effective pressure (BMEP). The peak BTE was found in the load range of 15 to 20 bar BMEP. Therefore a 6-cylinder engine with a bore of around 123 mm (similar to the PACCAR MX-11) is the right platform as the base engine for further development under the program. Supported by simulations, it became clear that at lower loads the influence of the friction and auxiliary losses becomes more important. Moving toward lower engine speed, one could reduce friction, but the heat transfer losses are greater than the reduction in friction losses. In addition, at higher loads the heat rejection in the intercooler is higher than the relative reduction in friction. Finally, as engine speed increases, the losses in the engine for turbo matching and pressure drop over the valves also increase.

The team investigated the robustness of the chosen platform for future combustion technologies. Literature suggests that extreme combustion processes (such as homogenous charge compression ignition) are capable of demonstrating high indicated efficiencies at low BMEP in research engines. To get 200 kW engine power for the BTE demonstration, a large swept volume is needed. Such an engine operating at low loads (and low power density) will fail to meet the targets for the vehicle freight efficiency as well as for cost. In a real-world engine, there will be some friction and auxiliary power needed as well, and these factors inevitably become higher in engines with larger volumetric displacement. Therefore, despite having high indicated efficiency performance, the net brake efficiency will be lower. Therefore, the chosen direction was towards faster heat release, but with less extreme (hence more stable) combustion concepts and limited increase of pressure as function of crank angle. Different technologies to shorten the heat-release were investigated in this project, including Gasoline Compression Ignition (GCI) combustion since the concept is still based on using a real-world pump fuel.

The WHR system requirements were defined in collaboration with potential partners/suppliers at the beginning of the program. It was apparent that extracting a 4% BTE contribution from an optimized engine, with a lower availability of waste heat power, is a real engineering challenge. Given the typical efficiency of the Rankine Organic cycle of a maximum of 14% (determined also by available temperatures and heat for the phase transition of the working fluid that cannot be recovered), this is only possible when all available heat sources from the engine are used. In addition, it requires very efficient heat exchangers to recover the maximum energy, without significant penalties such as back-pressure or flow restriction.

Vehicle Freight Efficiency Improvement Demonstration Route

A requirement of the SuperTruck II program is that the freight efficiency improvement must be demonstrated on several drive cycles, referring to the contractual technical milestone for budget period 5 in the program: "SuperTruck II Demonstrates > 100% Improvement in Freight Efficiency in Road Tests".

It is imperative that the drive cycle used must have several key characteristics, including:

- Representative of roads travelled by class-8 trucks within the United States
- Suitable for accurately demonstrating the freight efficiency gains of the SuperTruck II program, with a key goal being that these demonstrated gains will be reflective of real-world operation.
- Exhibit the balance between the often-conflicting goals of improved freight efficiency and proper vehicle performance. As such the drive cycle must contain events that demonstrate the vehicle's maximum longitudinal acceleration, and the vehicle's maximum grade climbing ability.

PACCAR Heavy-Duty Composite Cycle

The PACCAR Technical Center has vast experience in rigorously testing our vehicles for fuel economy and performance. PACCAR routinely uses a variety of drive-cycles and has developed a representative drive cycle for line-haul heavy duty trucks that is composed of local roads in western Washington wtate. This cycle is known as the Heavy-Duty Composite Cycle (HDCC).

The intent of the HDCC is to provide a drive cycle of reasonable testing duration that is representative of US highway miles driven by heavy-duty line-haul trucks, thus providing a representative measure of truck fuel economy that can be tracked over time. The HDCC is made up of several component cycles, which are run separately, with the results combined and weighted to achieve an overall representative fuel economy measure. The component cycles are:

- A northbound and southbound, section of Interstate 5 (I-5), with a combined round trip (loop) distance of 245 miles. This portion is run at a speed of 55 mph, and again at 60 mph. This section of road demonstrates rolling grades and includes grades more than 4%.
- A section of Snoqualmie Pass, following Interstate 90 (I-90) Eastward from Preston (east of Seattle) to the summit of Snoqualmie pass, for a total distance of 112 miles. This section of road represents the steep grades encountered in mountainous terrain and mountain passes and has grades up to 6%. The uphill and downhill sections result in a loop, having zero net elevation gain.
- A track cycle, run on the PACCAR Technical Center's closed oval test track, for a total of 500 miles. This portion has near zero road grade, and is performed at a speed of 65 mph. The inclusion of the track cycle is important to (1) better align the percentage of flat road content of the HDCC with real world data, and (2) include 65 mph road speeds.

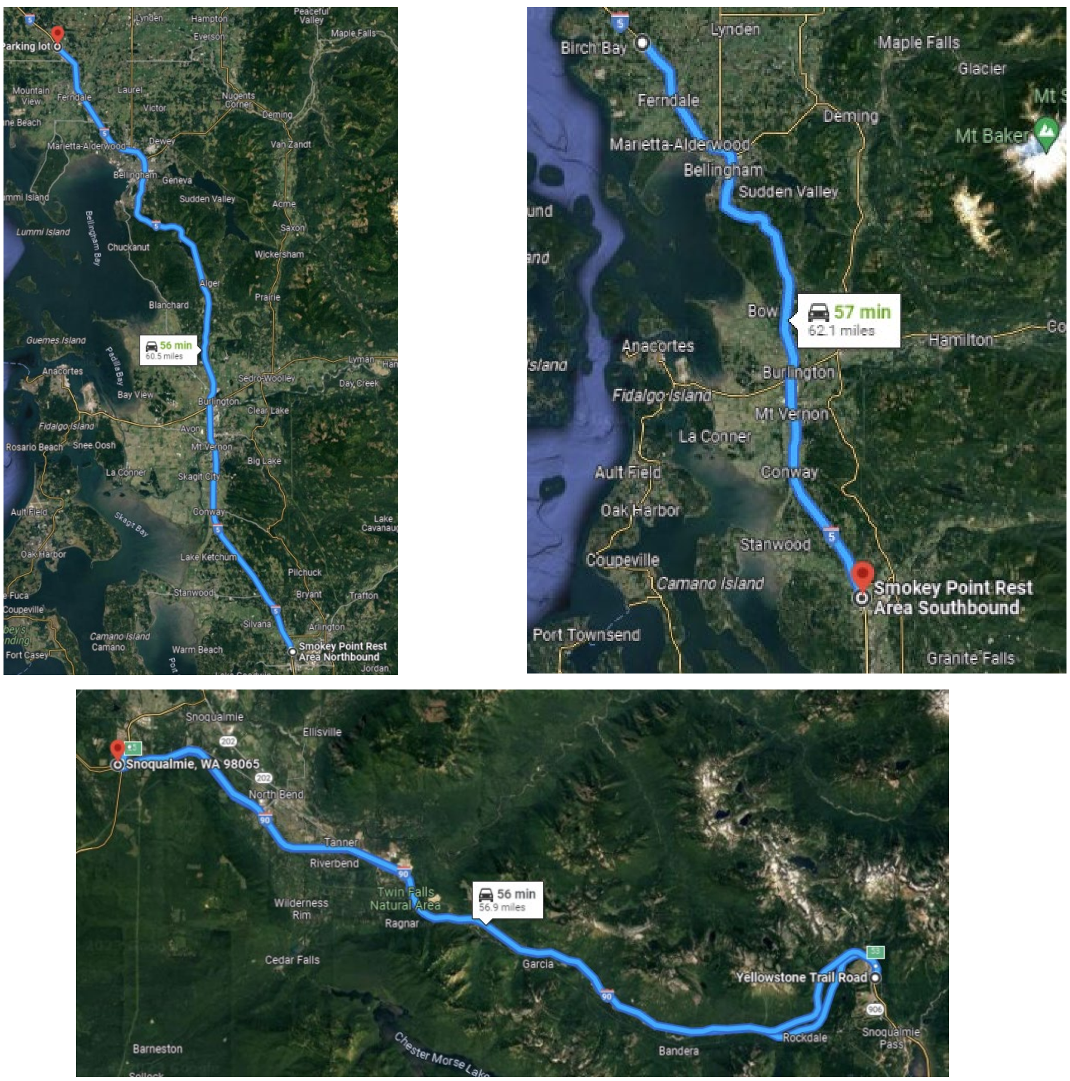


Figure 9: North and South sections of I-5 that comprise the PACCAR HDCC route(top) and map of I-90 section of the PACCAR HDCC route (bottom)

The component cycles are weighted and combined to achieve a single fuel economy measure. The combined overall cycle has a distribution of grade profiles that has been determined to be representative of the United States national highway system for vehicle miles traveled by heavy-duty trucks. Two key aspects of a drive cycle useful for evaluation and comparison are 1) the distribution of percent road grade, and 2) the distribution of "half-hill" lengths. A half-hill is defined as a section of road over which the road grade does not change significantly. A hill climb is characterized by positive grades, whereas a downhill section has continuous negative grades. The distribution of road grade and half-hill length for the HDCC are shown in Figure 10. Please note that roughly 40% of the roads exhibit grades in the range of 0.5% to 4% and that about the same percentage of hills are between 0.5 km to 3.5 km (0.3 - 2.2 miles) in length.

While the HDCC has proven to be a valuable test cycle for the evaluation of line-haul fuel economy, some inherent aspects of the cycle do not perfectly align with the SuperTruck II program. The HDCC emphasizes constant speed, highway cruising over long distances. While this is the most common operating condition, it overlooks other aspects of the truck operation that could be significant. The HDCC does not include any idle time, or any stops other than the few required to perform the given loops. In practice, the truck may see lengthy idle time each day – at the start of the driver's day, during pre-trip inspections, at any stops required by the route, etc. Furthermore, overnight hoteling can be a significant use of fuel, where either the main engine or an auxiliary power unit (APU) is used to sustain driver comfort during off-duty hours. In addition, a significant limitation is that the majority of the HDCC is run at vehicle speeds of 60mph, while nationwide speed limits are more commonly 65mph or higher. The HDCC includes 65mph testing on a track; however, this is all flat road conditions. It is desirable to have both higher speed operation and representative grades together, as this demonstrates truck performance and better represents actual loading conditions. For the reasons listed above, the team investigated an additional drive cycle for evaluations of vehicle design and performance use within the SuperTruck II program.

EPA Phase 2 Drive Cycles

In the Phase 2 greenhouse gas regulations that began in 2021, the EPA utilizes a simulation-based approach to certifying vehicle level CO_2 output. The Greenhouse gas Emissions Model (GEM) will use manufacturer specified vehicle inputs to drive each truck over a set of drive cycles to determine a representative value for fuel used and corresponding CO_2 emissions. The drive cycles used within the GEM model were carefully developed by the EPA, working in conjunction with National Renewable Energy Laboratory (NREL) which is the national-lab partner in this program, to provide vehicle operating conditions that reflect typical truck highway miles travelled by trucks within the United States.

EPA's drive cycle includes three component cycles, which are combined and weighted to achieve an overall representative cycle. The component cycles are:

- A transient cycle consisting of a varying speed vs time trace, with zero road grade. Vehicle speed varies between zero and 48mph, and idle time is included at two locations. Overall duration is 700 seconds, covering a distance of 4.57km (2.84 miles).
- A 55mph cruise cycle driven at constant speed, with varying road grade (minimum -5%, maximum +5%) and zero net elevation gain. The synthetic road grade profile was carefully designed with the goal of being representative of nationwide highway miles driven by trucks. The cycle is distance based to capture the fact that some trucks may not be capable of

maintaining the target speed on the steeper grades (power limited). The cycle is highly compressed, having a distance of only 21.6 km (13.42 miles).

- A 65mph cruise cycle that is identical to the 55mph cruise cycle in all aspects (distance and grade profile) except that the target vehicle speed is 65mph instead of 55.

The three cycles are weighted and combined to provide a single composite number. For class 8 sleeper cab line-haul trucks, the weighting is 5% transient cycle, 9% 55mph cycle, and 86% 65mph cycle. The 65mph cycle dominates the final resulting value. The distribution of grade and half-hill length are shown in Figure 10 with the HDCC data plotted as well for comparison. A condensed simulation time was a primary goal of the EPA, as manufacturers will be using GEM to certify thousands of different vehicle configurations every year, and a shorter drive cycle yields faster simulation time in GEM. To that end, the EPA drive cycles are very compressed – the national highway system is represented by a mere 13.42 miles.

The distribution plots show that the grade profiles successfully replicate the distribution of grade, and to a reasonable degree, hill length. However, this results in a situation where the grades may not completely represent real world operation. In the real world, the highest values of grade, although less frequent in occurrence on a percentage basis, often correspond to longer durations (for example, the grades encountered in mountain passes). In the condensed EPA cycle, however, 5% grade exists only for a single point in the cruise cycles. While these steep mountain grades occur infrequently when considering nationwide operation, when they do occur, they are long duration events. It is also obvious in the distribution of hill length that the cycle has very few hills making up the overall content. It would be more desirable to have a population of hill events, to better demonstrate the operation of powertrain system as it would occur in the real world. Another limitation of the GEM drive cycle is that, since it is completely synthetic, it cannot be demonstrated in the real world in a real truck – testing would be limited to simulation, a powertrain dynamometer, or a chassis dynamometer.

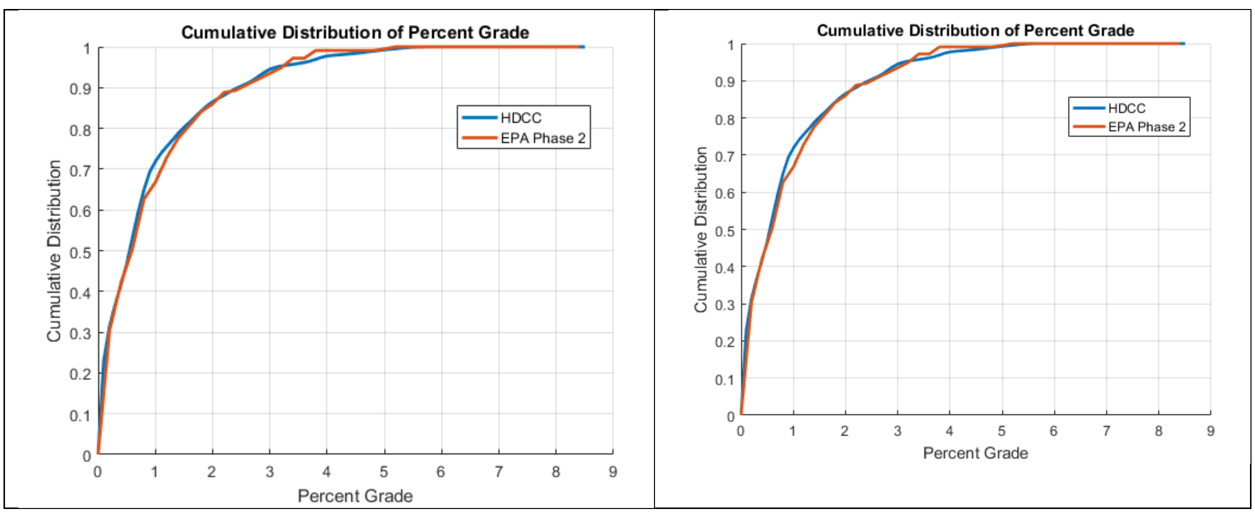


Figure 10: Cumulative Distribution of Road Grade (left) and Half Hill Length (right) for HDCC and EPA GHG Phase II Cycles

NREL Data Collection and Duty Cycle Analysis

Within the PACCAR SuperTruck II program, NREL worked with UPS to identify routes of interest and to arrange installation of data logging devices for a period of 2-3 weeks. Seven vehicles successfully logged data over long distance freight service routes. Over 88,000 miles of driving were recorded, representing line-haul operation including idle time. Figure 11 shows an activity map of the key UPS routes recorded. The data collection activity, in conjunction with available data on national road characteristics and usage, was utilized by NREL for the selection of the vehicle development drive cycle.

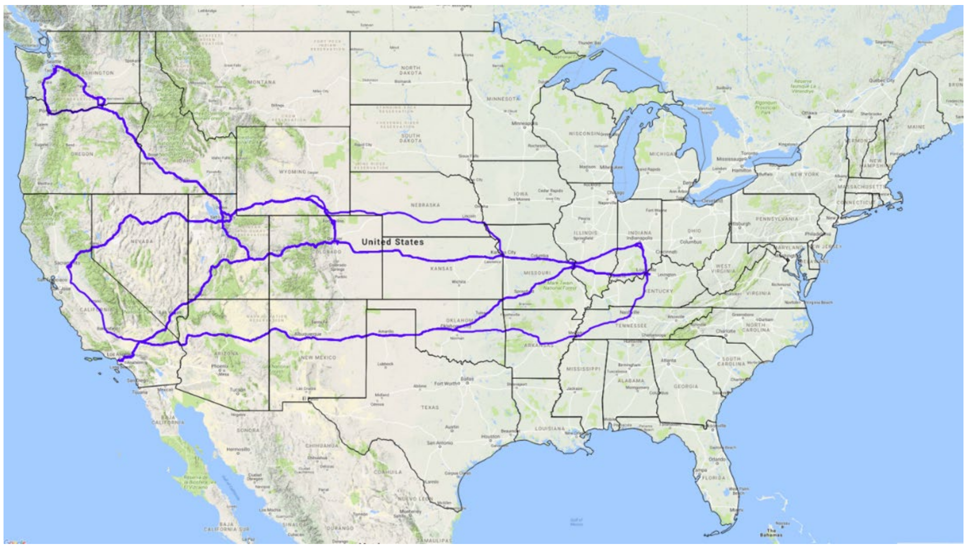


Figure 11: Data Logging Activity Map of UPS key line-haul routes

NREL utilized a specialized algorithm to search for nationally representative test routes that demonstrate road grade comparable to a national average. The algorithm was originally developed for the “EPA GHG Certification of Medium- and Heavy-Duty Vehicles: Development of Road Grade Profiles Representative of US Controlled Access Highways” effort. In that report NREL conducted a national analysis of MD/HD vehicle highway traffic from the EPA MOVES database to generate activity-weighted road characteristics including road grade and half-hill distances experienced by U.S. medium- and heavy-duty trucks on controlled access highways. Several opposing constraints affected the selection process.

The team utilized a pairwise comparison exercise to determine which key aspects of the drive cycles have the highest priority and ranked them in order of highest priority route consideration as shown in Figure 12. The most important features are those which could potentially preclude the ability to perform the test: seasonal weather and construction. Roads at reasonable absolute elevation (less than 5,000 feet) are preferred to ensure that the powertrain is operating under normal conditions without any engine derates etc. Speed limit is a key consideration due to the need to demonstrate operation at a typical operating speed and allowance of speed drop and overshoot for the purpose of energy management. The speed limits for trucks, by state, are shown in Figure 13. Test sites local to Washington State are not ideal unless the PACCAR technical Center test-track is included, due to the constraint of a 60 MPH speed limit.

Item #	Decision Criteria	Decision Criteria	1 Demonstrates Max Gradeability	2 Representative for % Grade	3 Representative for Hill Length	4 Representative for Vehicle Speed	5 Test Duration, Loop Distance	6 Min to Max Elevation	7 Max Elevation <5000	8 Elevation (Low)	9 Proximity to PTC	10 Traffic Density	11 Seasonal Weather (May-Aug)	12 Wind	13 Endpoint facilities	14 No Curvy Road (< Set Speed)	15 Construction	16 Weigh Station	17 Road Surface
1	Demonstrates Max Gradeability	1		2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17
2	Representative for % Grade	2			2	4	2	2	7	2	2	2	11	2	2	14	15	2	2
3	Representative for Hill Length	3				4	3	3	7	3	3	3	11	3	3	14	15	3	3
4	Representative for Vehicle Spee	4					4	4	7	4	4	4	11	4	4	14	15	4	4
5	Test Duration, Loop Distance	5						6	7	5	5	10	11	5	13	14	15	5	17
6	Min to Max Elevation	6							7	6	6	6	11	6	6	14	15	6	6
7	Max Elevation <5000	7								7	7	7	11	7	7	14	15	7	7
8	Elevation (Low)	8									9	10	11	12	13	14	15	16	8
9	Proximity to PTC	9										10	11	9	13	14	15	9	17
10	Traffic Density	10											11	10	10	14	15	10	10
11	Seasonal Weather (May-Aug)	11												11	11	11	15	11	11
12	Wind	12													13	14	15	12	17
13	Endpoint facilities	13														14	15	13	17
14	No Curvy Road (< Set Speed)	14															14	14	14
15	Construction	15																15	15
16	Weigh Station	16																	17
17	Road Surface	17																	

Sorted Criteria		
Seasonal Weather (May-Aug)	1	High
No Curvy Road (< Set Speed)	1	High
Construction	1	High
Max Elevation <5000	4	High
Representative for Vehicle Speed	5	High
Representative for % Grade	6	Med
Representative for Hill Length	7	Med
Min to Max Elevation	8	Low
Traffic Density	9	Low
Endpoint facilities	10	Low
Road Surface	10	Low
Test Duration, Loop Distance	12	Low
Proximity to PTC	13	Low
Wind	14	Low
Elevation (Low)	15	Low
Weigh Station	15	Low
Demonstrates Max Gradeability	17	Low

Figure 12: Pairwise comparison ranking (left) and priority of route consideration features(right)

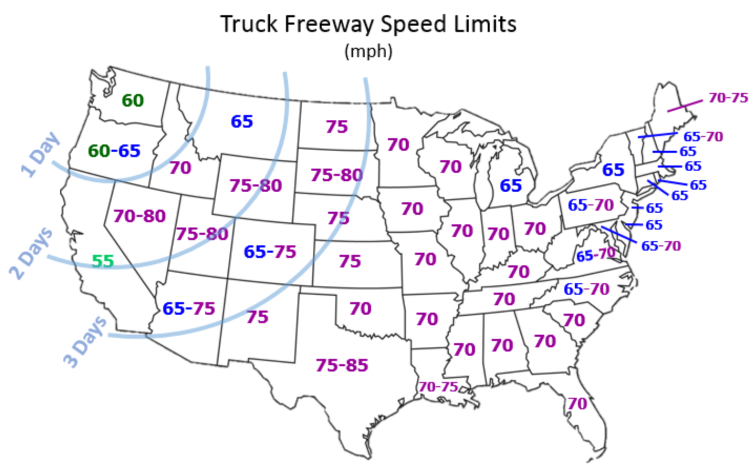


Figure 13: Truck freeway speed limit map and PACCAR Technical Center travel ranges (in days)

Within the constraints and considerations listed above, a test route for the development of the PACCAR SuperTruck II freight efficiency demonstration vehicle from Beach, ND to New Salem, ND on I-94 was selected, as shown in Figure 14. This route offers a grade profile and half-hill metric that is comparable to the national average in terms of overall distributions of absolute road grade and half hill distance, respectively. This is well below the 0.05 and 0.10 targets developed for the EPA analysis. The 256-mile round-trip route will be repeated twice per test for a realistic day of just over 500 miles of interstate highway travel. Without weigh stations or ports of entry to add uncontrolled variables to the testing, PACCAR will have the ability to simulate transient portions of the drive cycle in a repeatable and controlled manner. The 75 MPH speed limit permits validation of an extended range of fuel savings strategies above the 65 MPH target speed and within lawful limits.

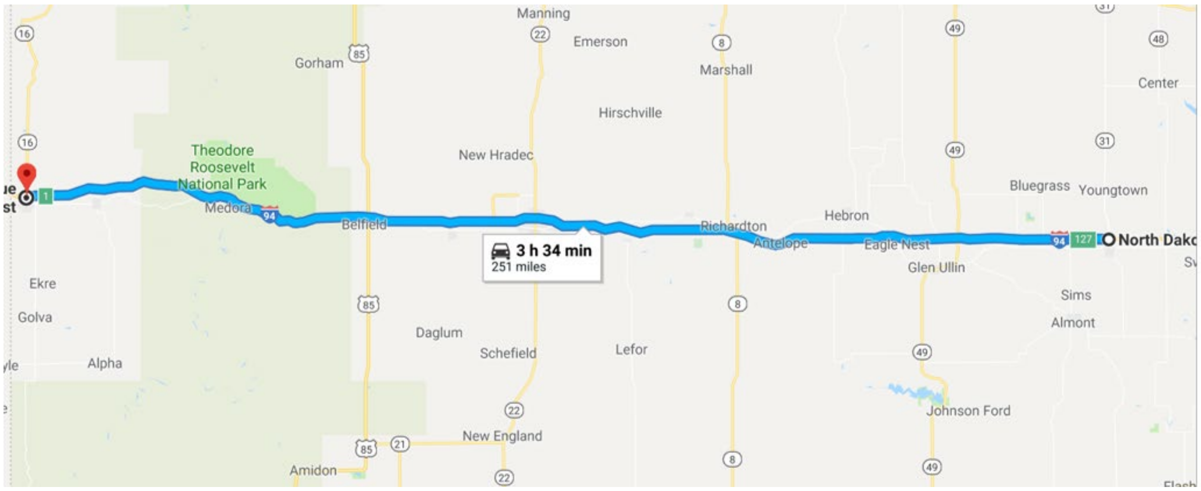


Figure 14: Beach, ND to New Salem, ND route

A high-resolution GPS measurement system was installed in a vehicle that drove the route to measure the exact grade profile and elevation, and to confirm the analysis based on database values and selection process. Figure 15 shows the resulting metrics for grade distribution and half-hill length. Plotted for reference are the curves for NREL's statistics on national US highway miles driven by trucks.

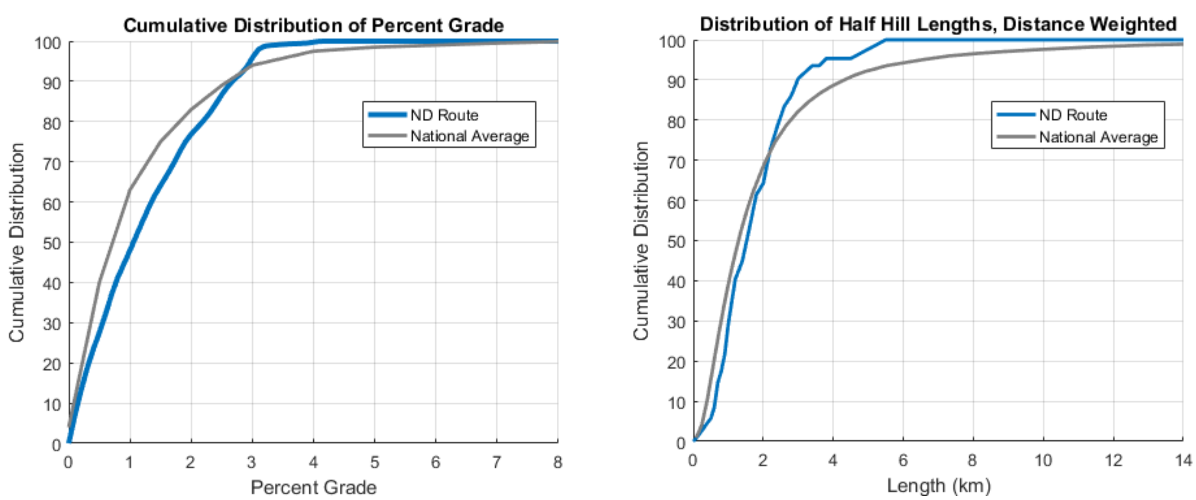


Figure 15: Road grade percent cumulative distribution(left) and half-hill length cumulative distribution (right)

Vehicle Gradeability

The SuperTruck II freight efficiency target includes a condition stating that performance on grades should be comparable or better than that of the 2009 baseline vehicle. Gradeability is directly proportional to available power. In the case of conventional powertrains, the available power is provided by the internal combustion engine; in the case of parallel Diesel-Electric hybrid powertrains, the available power is the result of the combined power of the internal combustion engine and electrical motor. Regardless of the power source, the required power to climb a 6% grade can be calculated using vehicle aerodynamics and tire rolling resistance as inputs. Simulation results in Figure 16 show that the 2009 baseline vehicle is capable of climbing a 6% grade at 59 km/h (37 mph). The label Preston in Figure 9f refers to the section of the HDCC drive cycle which includes a maximum grade of 6%. In order to maintain the same gradeability, one of the SuperTruck II configurations under investigation, including low friction tires and a 3.5 CdA will need between 250 and 325 kW of power to climb at the same speed.

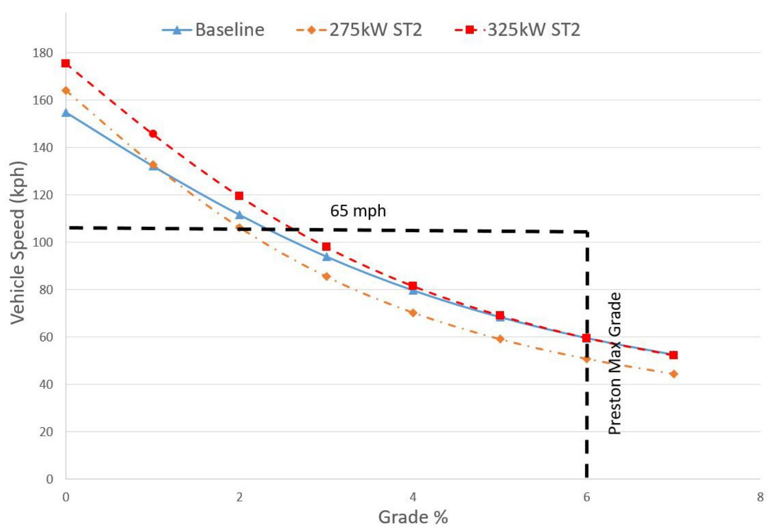


Figure 16: Vehicle Speed as a Function of Road Grade for Different Power Ratings

Ambient conditions and hoteling loads

NREL performed analysis of national temperature and solar data to determine appropriate ambient conditions for representative simulations and testing. From all 1020 typical meteorological year (TMY) weather station locations nationwide, 200 stations most closely corresponding to truck linehaul operating coordinates were selected. This down selection was done based on truck vehicle miles traveled (VMT). Each road segment VMT was assigned to the closest weather station and the least represented weather station was eliminated in an iterative process. Figure 7A shows the original 1020 TMY weather stations in the United States (Alaska and Hawaii not shown). Figure 16 shows the down selected 200 weather stations. The size of the circles shows the relative VMT weighting of each station. Selecting the representative weather stations based on truck traffic ensured that the data correlated to where trucks are operated and eliminated weather stations that were not representative (e.g. Northern Alaska). Figure 17 shows the temperature percentiles nationally for the 200 representative weather stations, while Figure 18 shows the solar percentiles nationally for the 200 representative weather stations. NREL is developing a method to determine national hours spent at weather conditions weighted by VMT for each weather station and a time-of-day distribution. This will provide a national distribution of relative time spent at weather conditions during idle events.

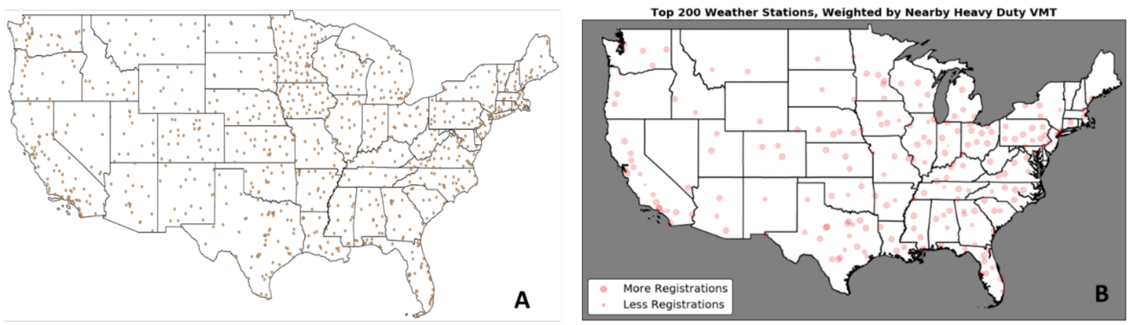
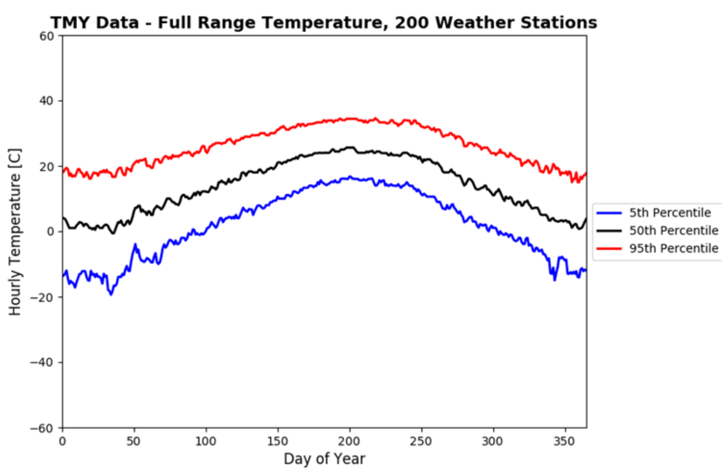
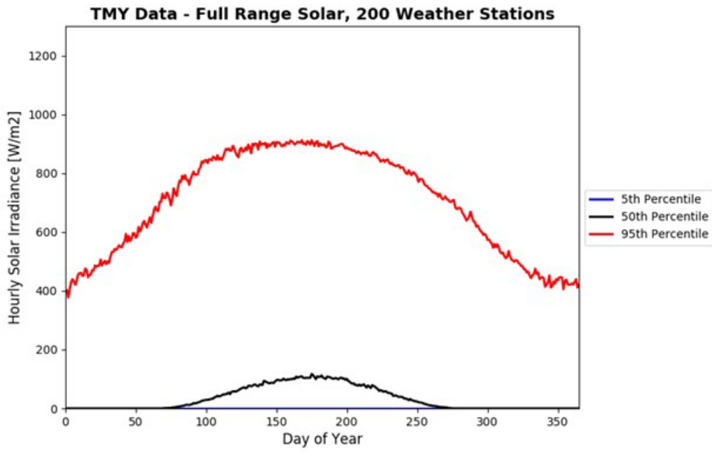


Figure 17: Weather station selection based on truck VMT. All 1020 TMY weather stations (left) 200 most representative weather stations based on truck VMT (right)



	Min	Mean	Max
5th	-19.4	1.7	16.7
95th	15.0	26.0	34.6

Figure 18: Temperature percentile bands based on 200 most representative stations



	Min	Mean	Max
5th	0	0	0
95th	377.0	704.7	912.1

Figure 19: Solar percentile bands based on 200 most representative stations

With a road segment selected and verified for the demonstration testing, work was done to define a hoteling cycle to represent typical overnight usage. The hoteling portion of the demonstration cycle was performed in a climate chamber see Figure 19, at the PACCAR Technical Center. This will allow for repeatable testing at consistent ambient conditions that can be set to be representative of typical customer usage. The truck interior will be controlled to a temperature set-point, and an electrical load profile representing typical occupant usage of devices and appliances will be used.

Duration of the hotel cycle will be 10 hours to correspond to a daily schedule of 14 hours on-duty time, and 10 hours of off-duty time, to align with federal regulations. Given that HVAC system loads are typically the most significant power draw during hoteling, the cycle will be demonstrated both for conditions of a typical warm day requiring the HVAC system to operate in cooling mode, and for a typical cold day requiring the HVAC system to operate in heating mode.

Figure 20: Kenworth SuperTruck II vehicle in the PACCAR Climatic Chassis Dynamometer

Selected Demonstration Route for Vehicle Development and Final Demonstration

The vehicle team consistently used the North-Dakota route, based on the analysis performed by NREL for all vehicle development activities. This approach ensured that the vehicle technologies developed under this program are capable of performing well under real-world driving conditions and that the estimated freight efficiency benefits are based on a national average representative class-8 drive cycle.

Due to program time constraints, PACCAR performed the final vehicle test activities on routes in Washington state which are part of the earlier discussed HDCC cycle. The I-90 (or Preston) section, which is favorable for the freight efficiency improvement of the SuperTruck II demonstrator vehicle due to its potential to use regenerative braking and predictive vehicle features, couldn't be used for final demonstration due to winter weather conditions. Nevertheless, the team concluded that the Kenworth SuperTruck II demonstrator vehicle design included sufficient margins to robustly meet SuperTruck II program deliverables for freight efficiency improvement without these particular route sections.

In order to determine the freight efficiency improvement impact over the route sections actually used for the final vehicle demonstrator, the team updated the weighting factors for the composite cycle of I-5 and track test to match closely to the national average of percent grade. Below are plots showing the percent grade and half hill lengths showing the close resemblance of the national average class-8 truck road usage in the US.

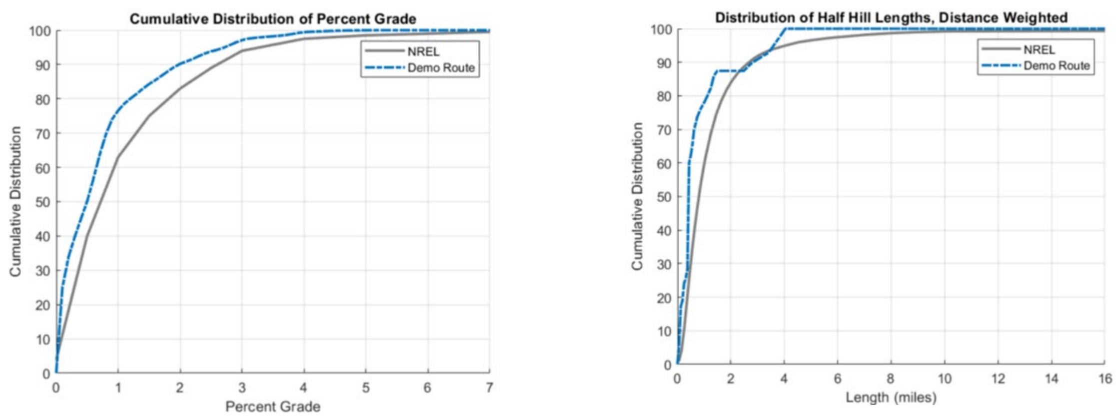


Figure 21: Road grade percent cumulative distribution(left) and half-hill length cumulative distribution (right) for actual road demonstration including PACCAR Technical Center test track and public road I-5 sections.

Vehicle Technologies Selection

The internal PACCAR SuperTruck program targeted to achieve a 120% freight efficiency improvement over the baseline 2009 vehicle. A systematic approach was taken to identify the relationships between critical components and evaluate the interdependencies such that we could prioritize our efforts where they would provide the greatest benefit to the whole. After a round of initial opportunity evaluations the team identified and allocated improvement targets across vehicle technologies as shown in Figure 22.

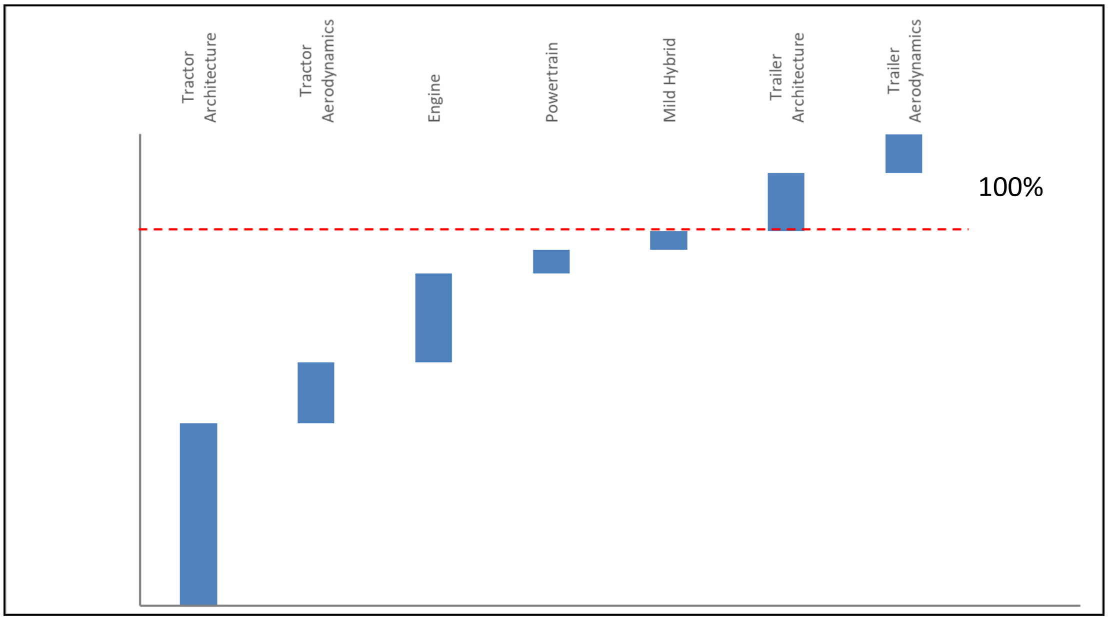


Figure 22: Freight Efficiency Improvement Allocation

Tractor architecture is a comprehensive term intended to capture all the architectural monuments that our production vehicles have. Many of these have been incurred as a function of having to cover a variety of use cases in the North American market, others have been the product of historical momentum within the market. The SuperTruck II vehicle would look past these and be a clean sheet design that is optimized for freight efficiency and to best serve the truck drivers of the future.

The layout of the chassis is an enabler for vehicle aerodynamics given that the layout must accommodate all the running gear within the bounds of the exterior shape. An iterative process of aerodynamic analysis and chassis refining led us to our general vehicle layout, a mid-ship engine placement with a SFFFA (Set Far Forward Front Axle) that was narrowed to enable full integrated wheel closeouts. This enabled a COE (Cab Over Engine) configuration that closely matched the ideal shape as identified in the initial aerodynamic studies.

Weight reduction is another area where the chassis architecture was tailored to enable several optimization efforts. By reducing the number of nodes by which loads are transferred into the chassis, material use can be optimized to yield an assembly that provides significant reductions in mass as well as improved durability and vehicle ride. Leveraging these nodes, front suspension, cab suspension, engine mounting, battery box, and fuel/air/DEF tank designs were optimized for performance and weight reduction. A summary of these refinements and their associated impacts to weight are illustrated in Figure 23.

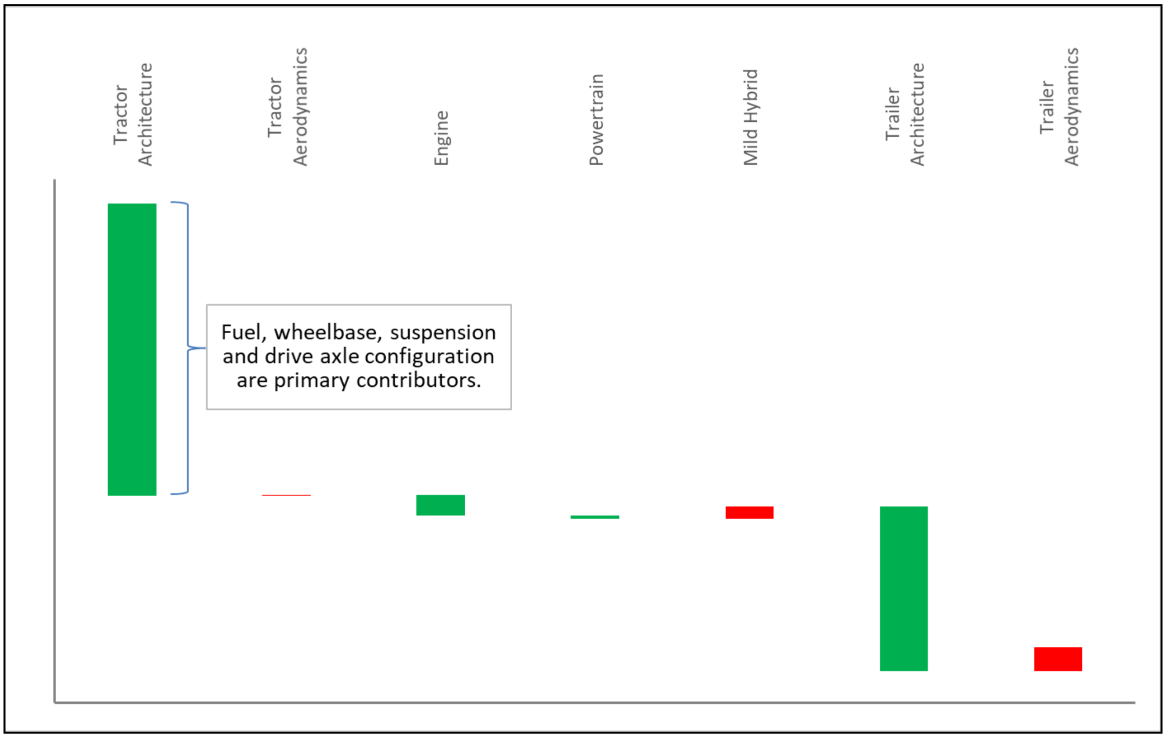


Figure 23: Weight Reduction by Source

A major contributor to the weight reduction is the use of a lighter PACCAR MX 11-liter engine in place of the baseline vehicle's 15-liter engine. The PACCAR engine's increased engine efficiency also reduces the amount of onboard fuel required to complete the same mission, which allows us to optimize the fuel and DEF tank volumes. This being an example of one of the aforementioned relationships.

The remaining components in the powertrain were selected to yield a vehicle that is optimized for line-haul and regional-haul applications yet remains flexible enough to serve most customers interested in a more fuel-efficient configuration just outside the bounds of these use cases by adding additional hardware such as additional axles, heavier duty springs, and larger tires. In this way ensuring that our development efforts would have a streamlined path to production.

The cooling package has a significant interdependency with external and underhood aerodynamics as well as overall powertrain performance and power consumption. As a first step, separating the needs of typical line- and regional-haul applications from bulk haul and other vocational applications with higher cooling needs allowed us to reduce the size, mass, and airflow needed. This enabled us to engage in a multi-factor optimization effort to deliver a module that would package within a space claim that delivered the basic exterior shape that our aerodynamic studies had identified as ideal.

Elimination of unnecessary components and consolidation of component functions were also made possible through the full chassis redesign and node consolidation. Initial analysis suggested that significant baseline chassis weight could be eliminated by this approach. This philosophy was utilized in the creation of several multi-function modules, each of which only transfers its loads through 3 nodes. The structural bracketry that typically supports these components was eliminated or consolidated.

- Fuel/DEF Tanks and Air Tanks
- Battery, Charge Air Cooler, and Battery Controller
- High/Low Temp Radiators, Low Temp Pump, Engine Air Cleaner, and Electric Fans
- Engine, Transmission, and P2.5 Hybrid Assembly
- Front Suspension: Leading Radius Arms, Air Springs, Axle, EPS Rack
- Rear Suspension: Sway-bar, Upper Control Arm, Air Springs

The layout of the traditional cab interior environment was also reevaluated for this program. One mayor departure from traditional cab architecture that was implemented is a center driver seating position. The greatest motivation for this is the benefit to aerodynamics from a narrower front cab shape. The change is justified by the percentage of time where a single driver is occupying the vehicle in this segment (90-95% of the time). This position required consideration of a few key topics including visibility outside of the cab, ergonomics within the cab environment and the ability to interact with the exterior environment. Advanced technologies can aid situations such as backing and docking but need to be refined from their current state to enable market acceptance. This test property will be used to develop the state of technologies to ease the transition to production with real world experience. As an example, camera mirrors need to provide some better depth perception, akin to what a driver gets from moving their head to gain additional perspective. Backing aides, including trailer tracking and backup lines, are also under investigation.

To optimize the driver packaging and visibility, several iterations of design mockups were generated along with ergonomic simulations, which enabled engineers to have a quantitative and qualitative assessments of each layout design before physical prototypes are created. Within the simulated cab environment aspects such as driver visibility, reach, fit, and ingress/egress were evaluated and optimized. Assessment of the driver's direct visibility to the outside environment includes simulated real-world objects (cars, pedestrians, bicycles, intersections, road signs etc.) to ensure cab design concepts reduce driver fatigue and increase safety. The driver's indirect field of view can also be assessed (the reflected images the driver sees in the mirrors of the environment around the truck) to validate that new cab shape and driver position would not create distractions for the driver. Within each of the simulated cab environments the digital human model is being moved to simulate reach for controls, climb in and out of the truck, and other common tasks that a driver performs. Results were compared and contrasted to the current product with a cross-functional team that included drivers and ergonomics experts. The new cab provides a superior environment to existing products, allowing the driver to focus on safe and efficient operation of the vehicle.

Notable interactions between the driver package layout and the chassis/drivetrain were identified and optimized for best effect. The development of the steps into the cab was an area of focus in accordance with our commitment to make the vehicle as safe as possible, which includes ingress and egress. Not only does that ensure that the steps up to the cab are fewer and shorter, but this yields a larger interior volume for any given roof height. The larger volume makes for a more livable interior with greater flexibility for optimization to a variety of future use cases. This relationship was part of the rationale for the mid-ship engine location such that we could lower the engine within the chassis without affecting the front axle space claim. The frame rails were also affected by this relationship, where the lowered chassis enables us to reduce the drag area as well as enabling a greater amount of suspension height variation within the legal height limits for the vehicle. This being a critical element for an aerodynamic vehicle since the outer body components at the bottom edge of the vehicle are often damaged by operators encountering road debris or uneven pavement.

To address this concern, we developed new front and rear suspensions for the SuperTruck that enabled 200mm of suspension travel. The increase in travel gave us the ability to run a lower static ride height at high speeds for reduced aerodynamic drag as well as an elevated ride height at low speeds to improve ground clearance, approach angle, and departure angle. In addition to these benefits, the new suspensions were lighter than existing offerings and provided a higher roll center. The latter was incorporated to resolve the effect of another relationship, in this case, the relationship between weight distribution and vehicle dynamics. The SuperTruck tractor was expected to be significantly lighter than any of our existing production vehicles meaning that a fully loaded trailer would yield a tractor and trailer combined center of gravity that had moved rearward beyond the bounds of our previous experience.

To evaluate the effectiveness of our approach at addressing this concern, we assembled a proof-of-concept "mule" vehicle that was fitted with an early version of our intended chassis, the new front suspension with a power steering rack, and an existing rear suspension that also provided the appropriate roll center. A current production cabin was modified and fitted in the correct COE position, yielding representative loads. This test property was further instrumented with over 120 strain gauges on the rails, crossmembers, and suspension components so that we could correlate our chassis FEA model to what was seen on our test track. At the PACCAR Technical Center, the vehicle was subjected to a selection of events on our durability track before working its way up to our final challenge, a subjective handling evaluation at high speeds. The data was then used to further optimize the chassis for the demonstration vehicle ensuring that the vehicle is safe to operate.

Powertrain Selection

Based on the demonstration trucks developed under SuperTruck I, it was evident that in order to meet the SuperTruck II freight efficiency targets a Diesel/electric hybrid powertrain needed to be developed. At the beginning of the program, PACCAR actively explored several hybrid concepts, based on earlier internal demonstration programs spanning a wide range of applications. These projects provided a knowledge base of applicability, strengths, and weaknesses. Four hybridization programs formed the basis for SuperTruck II input which are summarized below.

(1) CONVENIENT and (2) ECOCHAMPS are EU funded projects to develop proof-of-concept regional haul heavy-duty parallel hybrid vehicles capable of braking energy recovery and limited electric-only operation through urban zero emission zones.

Figure 24: DAF CONVENIENT (left) and ECOCHAMPS (right) project hybrid vehicles in Europe

(3) HECT (High Efficiency Cargo Transport) and ZECT (Zero Emissions Cargo Transport) are California South Coast Air Quality Management District (SCAQMD) funded projects to develop proof-of-concept heavy-duty series hybrid vehicles focused on emissions-free operation through emissions non-attainment zones around ports for drayage applications.

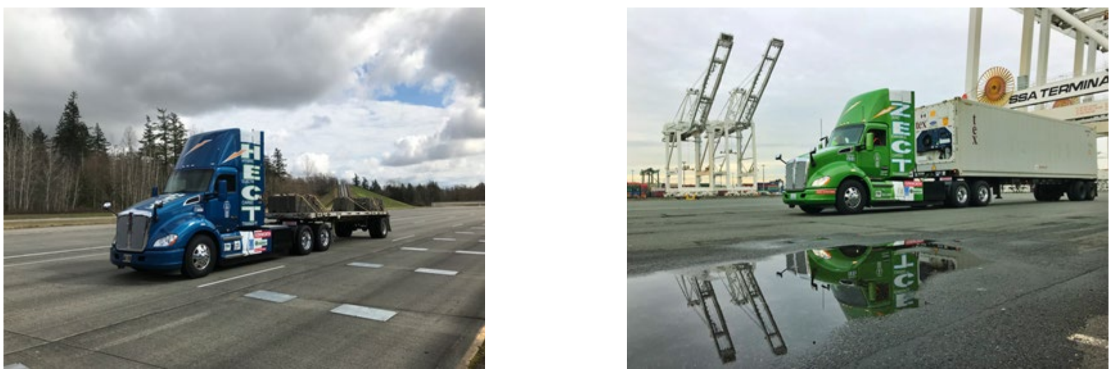


Figure 25: Kenworth HECT (left) and ZECT (right) project hybrid vehicles in North America

(4) The Electrically Regenerative Accessory Drive (eRAD) project is a 48 Volt mild-hybrid powertrain developed by Eaton in collaboration with PACCAR. This concept is a 2-speed transmission-mounted Power Take-Off (PTO) unit connected to an HVAC compressor and a small sized (10kW) electric motor. Figure 25 shows the location of the eRAD unit and highlights some of the components that can potentially be removed from the engine.

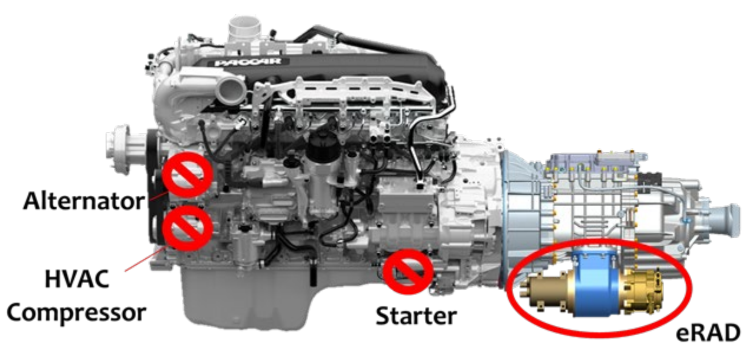


Figure 26: Eaton's 48V eRAD system with 10 kW e-motor

Table 6 shows the hybridization scenarios considered for the hybrid powertrain analysis. The properties that were varied are e-motor power, battery configuration, system voltage and integration method. The analysis covers both mild and strong (or full) hybrids with the size of the electric motor and voltage level varying accordingly. In the context of the current project, a strong hybrid is characterized by providing electric traction power, whereas the mild hybrid primarily provides power to the accessories and supports regenerative braking. In all cases, energy storage was sized to accommodate regenerative braking energy over the relevant routes. The mild hybrid is a 48V PTO-mounted system similar to the eRAD concept discussed above - whereas the 650V strong hybrid configurations feature an electric motor integrated with the transmission.

eMotor Power	15kW	30kW	45kW	60kW	90kW	120kW
Battery (energy / power)	e10kWh	e10kWh	p10kWh	p10kWh	p10kWh	p10kWh
System Voltage	48V	48V	650V	650V	650V	650V
Mounting	PTO	PTO	PTO	Integrated	Integrated	Integrated

Table 6- Hybridization Scenarios

Table 7 shows the decision criteria considered when selecting the hybrid configuration developed for SuperTruck II. In addition to the stated program goals for freight efficiency and payback, much consideration was given to the commercial feasibility of the technology solution. The system has to be serviceable and leverage economies of scale, for example in the area of accessory electrification. Market data also suggests there is a cost threshold that dictates what the customer is prepared to pay for new vehicle content. Cost estimates are based on 120,000 annual miles driven, $3/gallon fuel, and 3 nights of hoteling per week.

Criteria	SuperTruck II Goal
Freight Efficiency	> 100% Improvement
Customer Payback	36 Months
Serviceability	Production Viable
Volume Potential	Production Viable
Customer Cost	Below Acceptance Threshold
Commercialization Cost	Production Viable

Table 7 - Hybrid Decision Criteria

1D simulation models were applied to evaluate the potential of each powertrain architecture. By using a speed control algorithm that allows the use of vehicle kinetic energy as a storage mechanism, regenerative braking and the use of electrical propulsion at low power demand. The primary control mechanism assesses the vehicle state based on vehicle speed and current power demand (either positive or negative). The vehicle model uses a Lower Speed Limit (LSL) and an Upper Speed Limit (USL) defined based on the drive cycle target velocity. If the vehicle speed is between the USL and LSL and the power demand is zero, the vehicle is allowed to coast with the transmission in neutral and the engine turned off. Above the upper speed limit, regenerative braking is applied via the electrical machine until the upper vehicle speed limit is reached, or battery state of charge reaches 90%. The vehicle's foundation brakes are used when the e-machine's maximum power absorption is exceeded.

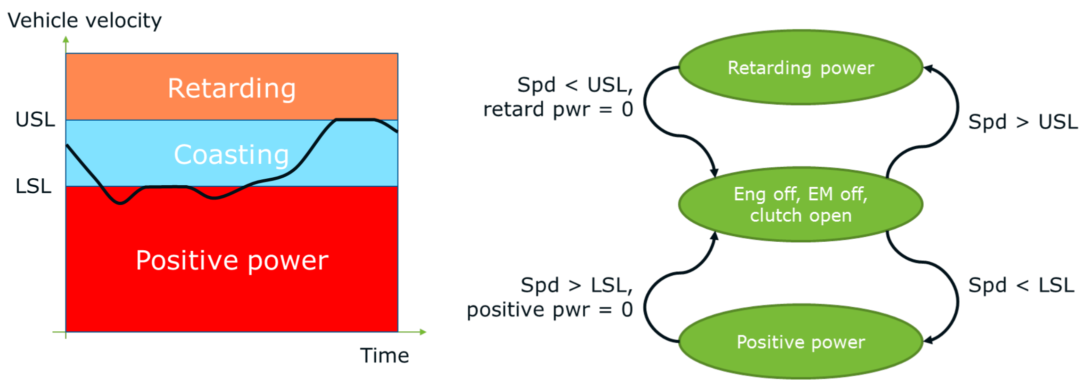


Figure 27: Overview of the speed control model used for hybrid powertrain selection evaluations

Using weight estimates for electrified powertrain components, the freight efficiency of each scenario was calculated and compared to a non-hybridized baseline scenario. The results are shown in Figure 27. From this study the following conclusions were drawn. (1) Increasing levels of hybridization (e-motor size) have a very marginal benefit on freight efficiency. (2) Routes characterized by rolling hills and relatively low vehicle speeds tend to favor a strong hybrid more than routes and vehicle speeds representative of the national average. Several factors are responsible for this result. Higher speeds tend to reduce the hybrid benefit by reducing the opportunity to capture braking energy. Instead, most of the energy is required to overcome losses related to the aerodynamic drag. While a larger motor does increase the ability to capture energy, it comes at increased weight, thus penalizing the ability to coast. Large batteries were ruled out at the beginning of the analysis due to their negative impact on freight efficiency

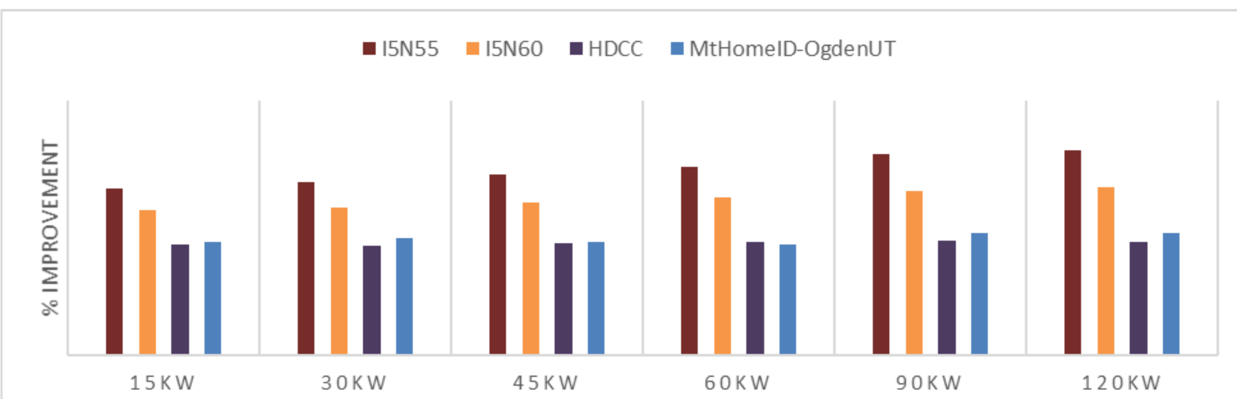


Figure 28: Freight Efficiency Improvement vs. eMotor Size

Customer payback time was first evaluated for the lowest power system that can provide the desired hoteling and engine-off coasting benefits, with a breakdown of cost and benefits as shown in Figure 28. The payback time is well within the SuperTruck II goal of 36 months. This is expected to improve substantially for applications with more hoteling.

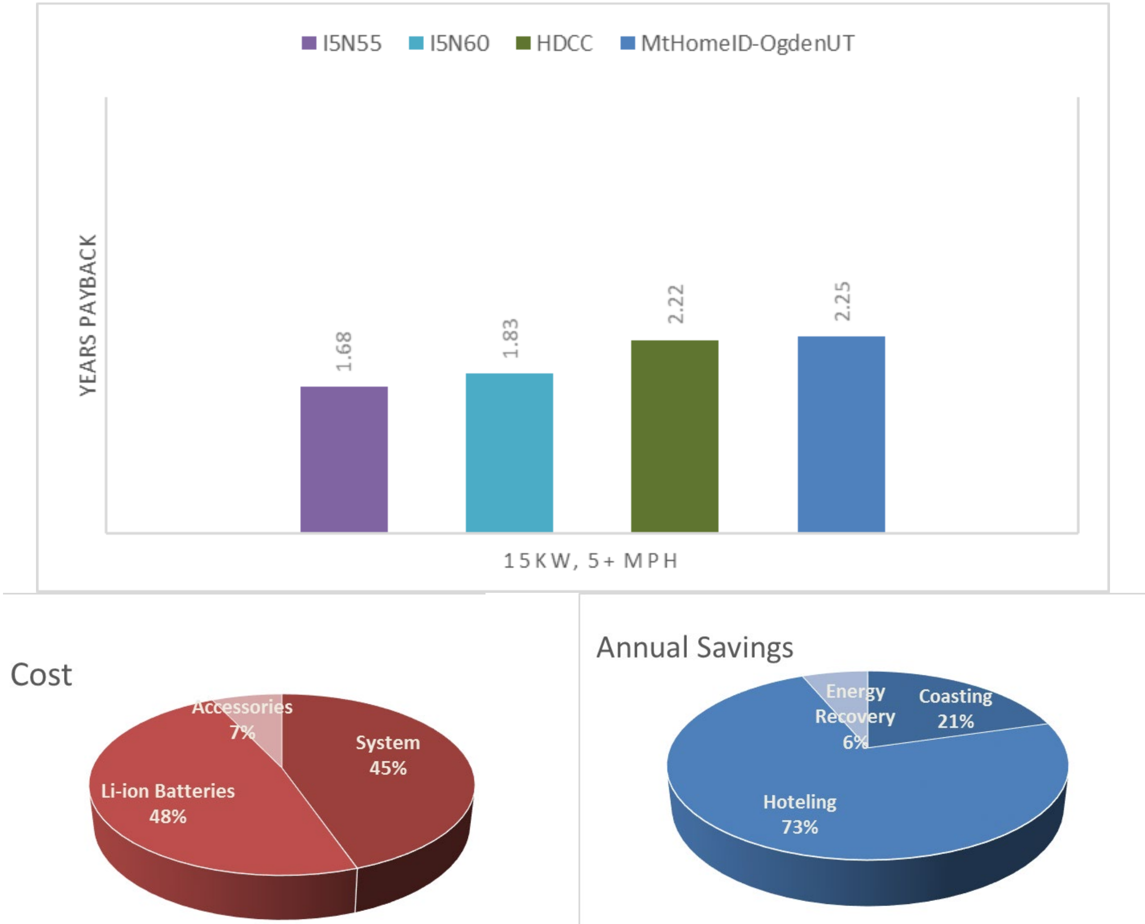


Figure 29: Payback of Mild-Hybrid System vs. Conventional (top), Breakdown of Incremental Cost for Hybrid System (left bottom), and Hybrid System Freight Efficiency Contributions (right bottom)

Next, the incremental payback time for the higher hybridization levels were compared to the 15 kW system. This analysis revealed that for a 3-year payback time the optimal configuration is somewhere in the 15-30 kW range. Moving to higher levels of hybridization quickly pushes out the incremental payback time to where there is no benefit over the lifetime of a truck.

Summary of other evaluation criteria:

- Serviceability: 48 Volt systems were ranked better for serviceability than 650 V systems because high voltage training would not be required to service the system at dealerships.
- Volume potential: Current adoption of battery HVAC systems are in the 10-15% range for on-highway truck fleets, but the North American Council for Freight Efficiency (NACFE) projects that this could reach 30-40% over the next decade. While strong hybrids are being considered to meet local Zero Emission Vehicle standards, the volumes are not expected to exceed 10% for heavy-duty trucks over the next decade. For this reason, mild hybrids received a higher score in this category.
- Customer cost: Fleets have limited budgets for equipment purchases. Even if a particular technology provides a reasonable payback, it will likely not be considered if it significantly reduces the ability of the fleet to grow its business.

- Commercialization cost: The development costs associated with a fundamentally new powertrain design are substantial. Following the initial proof-of-concept phase, the OEM has to spend a large amount of resources ensuring the concept is commercially viable, validating the product and preparing for manufacturing and aftermarket operations. Concepts associated with a lower development cost increase the likelihood of commercialization and obtained a better score in the evaluation.

Figure 29 shows the scorecard for each hybridization scenario against the selected criteria. A 15-30 kW 48V mild hybrid was determined to be the optimal hybrid configuration for SuperTruck II and will be the focus of future work.

eMotor Power System Voltage Integration	15 kW 48 V PTO	30 kW 48 V PTO	45 kW 650 V PTO	60 kW 650 V Integrated	90 kW 650 V Integrated	120 kW 650 V Integrated	120kW 650 V Integrated
Contribution to ST-II Freight Efficiency	G	G	G	G	G	G	R
Customer Payback	G	Y	R	R	R	R	R
Serviceability	G	G	Y	Y	Y	Y	Y
Volume Potential	G	G	Y	R	R	R	R
Customer Cost	G	Y	Y	Y	R	R	R
Commercialization Cost	Y	Y	Y	R	R	R	R

Figure 30: SuperTruck II Hybrid System Scorecard

7. Engine Technology Development

This chapter provides an overview of different key engine technologies developed under the program. The SuperTruck II base engine design includes a new updated cylinder block, crankshaft, pistons, and connecting rods, which increases the engine swept volume of the production PACCAR MX-11 from 10.9 to 11.9 liter. The increased stroke in combination with Late Intake Valve Closing (LIVC) Miller valve timing results in a higher expansion ratio, which increases the gross indicated thermal efficiency. The design will also feature 2-stage turbo charging, electrified accessories with no Front End Accessory Drive (FEAD), low viscosity oil, and modifications to the oil system for reduction of friction and increased BTE. To ensure high commercialization potential, reciprocating assembly designs have been implemented to ensure they meet production derived simulation strength and robustness requirements. The key components that supported the 55% BTE demonstration under this program are summarized in Figure 30 below:

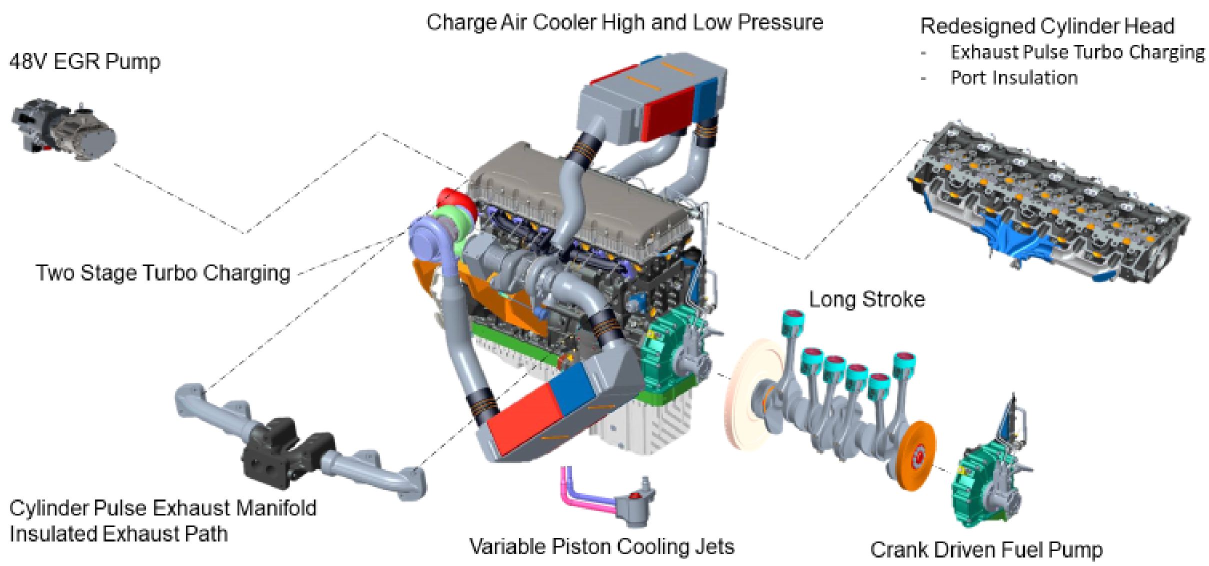


Figure 31: New engine technologies for the PACCAR SuperTruck II engine

Combustion System

The design of a new combustion system started with in-depth CFD simulation activities. The in-cylinder combustion models used were initially validated against test data from a single cylinder engine, a dedicated test-facility mainly used for combustion system optimization. A computational design of experiments (DoE) has been devised and carried out to find the best combination of design parameters of the combustion system that meet the project objectives. Compression ratio and injector design parameters such as number of holes in the injector nozzle, included angle of the nozzle, fuel injection pressure, timing, and flow rate were the focus of the investigation.

The simulation matrix (1700 cases) provided an optimum solution with a predicted Gross Indicated Thermal Efficiency (GITE) improvement close to 1%, while keeping the same NOx emissions as the baseline system. While performing a timing sweep with the optimum design parameters, the peak cylinder pressure was predicted to increase but to remain close to current production limits, as shown in the figure below.

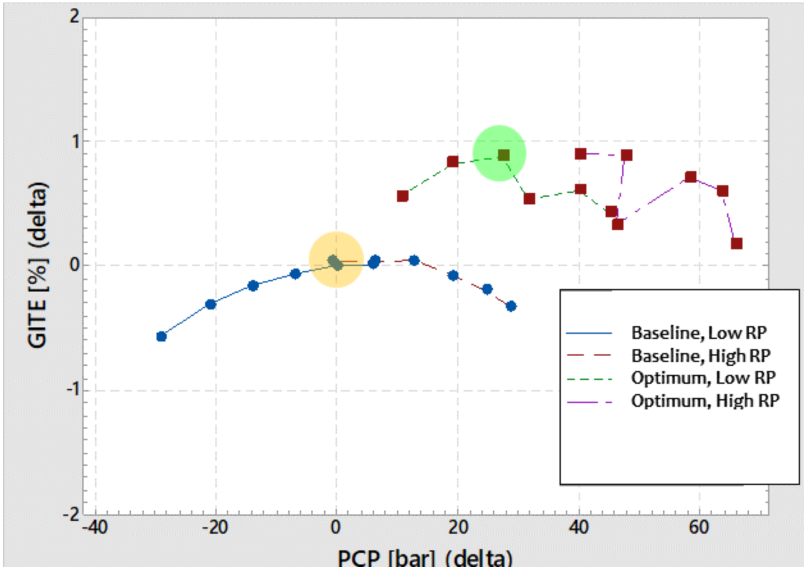
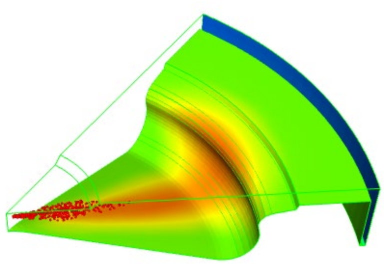


Figure 32: Efficiency vs peak cylinder pressure trade-off for baseline (orange) and optimum (green)

The team continued to investigate the impact of piston bowl profile and various injection configurations on the gross indicated thermal efficiency of the SuperTruck II combustion system design. The piston bowl optimization was initialized using the previous iteration of the ST-II bowl parameters. The injector parameters investigated include variation of the number of nozzle holes, rail pressure, nozzle inclusion angle, and start of injection (SOI). An automated procedure of 3-D CFD combustion analysis was created to investigate possible GITE improvements for the ST-II combustion recipe. The piston bowl design exploration was initialized by the geometric parameters belonging to the ST-II combustion system recipe bowl profile. These parameters are illustrated in the figure below. This investigation showed that higher GITE (roughly a 0.5% improvement) is possible by increasing the number of injector nozzle holes and having a slightly deeper piston bowl profile compared to the previous ST-II combustion recipe.



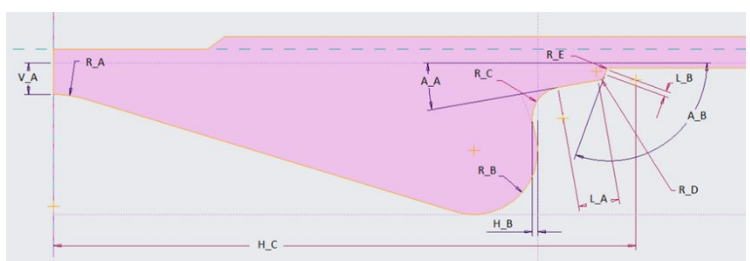


Figure 33: Piston Bowl Parameters Study using 3D CFD

The effect of injector flow rate was evaluated from 1.54 l/min to 2.80 l/min for the anticpated SuperTruck II operating point boundary conditions. The resulting injection profiles and GITE-NOx trade-off is shown in the figure below. As the injector flow rate increases, the injection duration decreases, thus governing the combustion duration. As a result, GITE is increased at constant NOx.

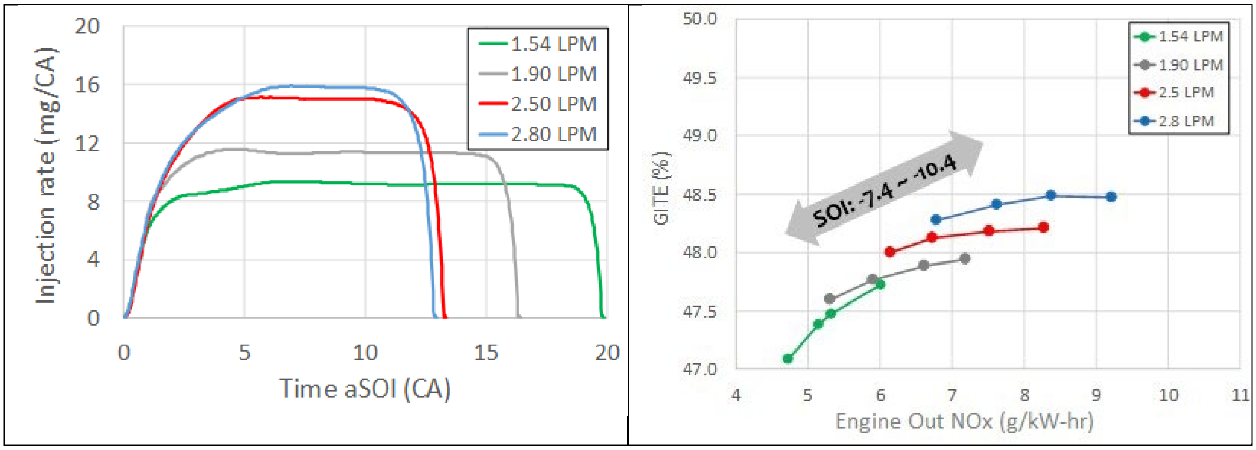


Figure 34: Injection profile comparison (left) and GITE NOx performance comparison

The single cylinder engine test was performed at the SuperTruck II reference load point using 5 different inlet boost levels and a Design of Experiments testing method for injection timing and pressure. Hardware combinations used for the model validation consisted of a modified piston bowl (based on earlier combustion optimization work), 40° retarded intake valve closure and 3.2 liter per min flow rate injector with 158° included spray angle. The in-cylinder pressure predictions correlated well with the measured data, providing confidence in the model.

Cylinder-head Design

High boost pressure is a critical component in the implementation of the SuperTruck II engine concept. To support this goal, an intake port geometry optimization study was carried out for maximizing the flow coefficient, defined as the ratio of mass flow into the cylinder versus the isentropic flow across the valves. 3D CFD models were set up to successfully replicate a physical flow coefficient and swirl test conducted in-house as shown in the figure below.

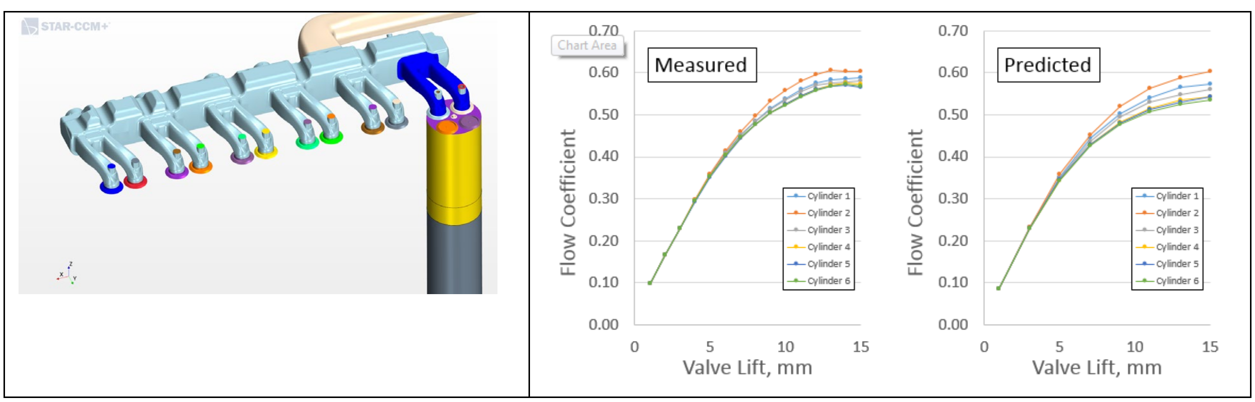


Figure 35: STAR-CCM+ CFD model of the MX-11 intake and ports

For the port geometry optimization, the flow coefficient was defined as the cost function. Cylinder 4 was isolated and a new CFD model was created to capture single-cylinder geometry. A new workflow was developed that utilized automated solver and mesh adoption techniques. After many iterations of solver parameters and control point locations, a solution with a significant increase in flow coefficient emerged. The optimized shape was used to create a new STAR-CCM+ model and a sweep of valve lift values was investigated. The figure below compares the optimized geometry with the baseline. The port areas are increased, particularly for the optimized one on the right.

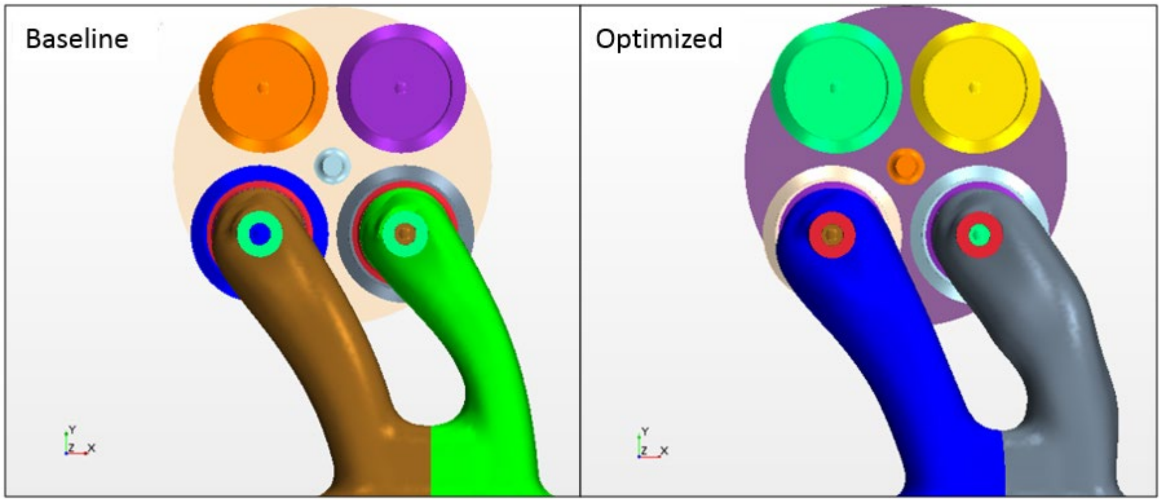


Figure 36: Intake ports, baseline vs. optimized

The resulting improvement in the flow coefficient with the optimized port geometry is shown in Figure 61. A 5% increase in the flow coefficient at 14mm lift was the maximum change found during the iterations. At lower lift values (from 1 to 5 mm) there was very little change from the baseline shape. Improvements increased as larger lifts were investigated. A CAD feasibility check showed that the new geometry would impinge into the cooling passages in the head. An updated CAD geometry is to be made minimizing conflicts and to be evaluated with the CFD model.

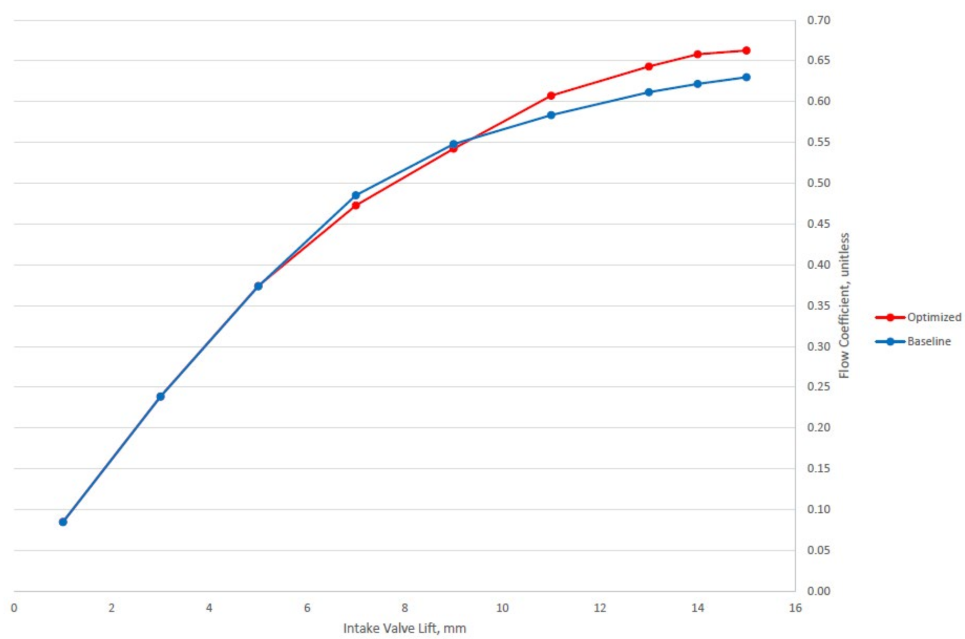


Figure 37: Flow coefficient baseline vs. optimized port geometry

The SuperTruck II intake manifold design was also driven by simulation tools to optimize intake port temperature and airflow ratio variation between cylinders. Analysis revealed that air-fuel ratio (AFR) spread across the cylinders was over 5%, due to the single-entry point being located off the center. Because of the efficiency impact of this spread, intake manifold design improvements were required. The new SuperTruck II twin entry symmetrical manifold design with delivers a measurable reduction in AFR and intake temperature variation between cylinders to enhance brake thermal efficiency. The manifold reduces cylinder-cylinder AFR spread down to less than 2%. In addition, the baseline production manifold showed more than a 50°C intake charge temperature difference in-between cylinder 2 (coldest) and cylinder 6 (hottest).

Figure 38: Baseline PACCAR MX-11 intake (left) and SuperTruck Dual Entry intake (right)

A new head casting optimized the port designs and open the current integrated intake manifold for improved flow distribution to the cylinders. The new head will reduce the exhaust port volumes for improving turbocharger turbine efficiency. The exhaust port is changed from rectangular to a round circular cross section at the port outlet. To test the reduced exhaust volume before the new cylinder head casting was ready, exhaust port inserts were fabricated to transition to the new exhaust manifold design.

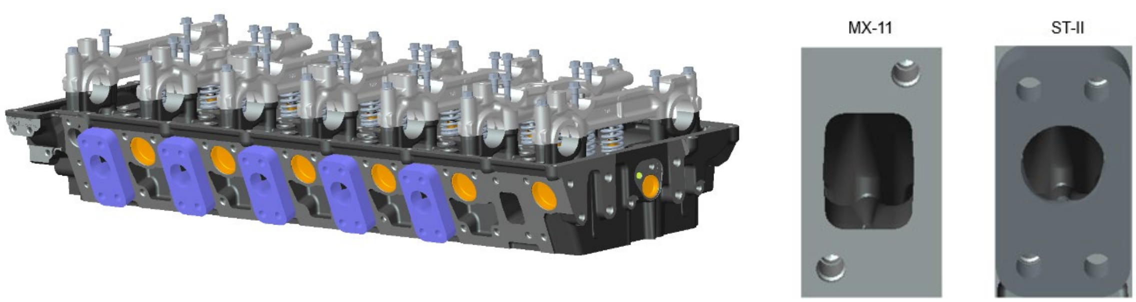


Figure 39: Cylinder Head with Exhaust Port Inserts for Reduced Exhaust Volume

Valve-timing

A main effect of Miller timing based on the LIVC strategy is reduced gas temperature and accompanying reduction in NOx formation. This NOx margin can be leveraged for improved thermal efficiency and reduced wall heat loss. With LIVC, piston compression work is reduced, but on the flipside the turbocharger must provide a higher boost to compensate for the reduced effective CR, causing additional pumping losses. This concept therefore only works if turbocharger efficiencies are significantly improved compared to the baseline situation.

1-D GT-Power engine simulations were used to evaluate the total efficiency gain with the LIVC at the SuperTruck II engine reference point, using a LIVC with reduced EGR rate plus a high efficiency two-stage turbocharger concept. Three different IVC timings were tested with the intake valve opening (IVO) unchanged to maintain the original valve overlap.

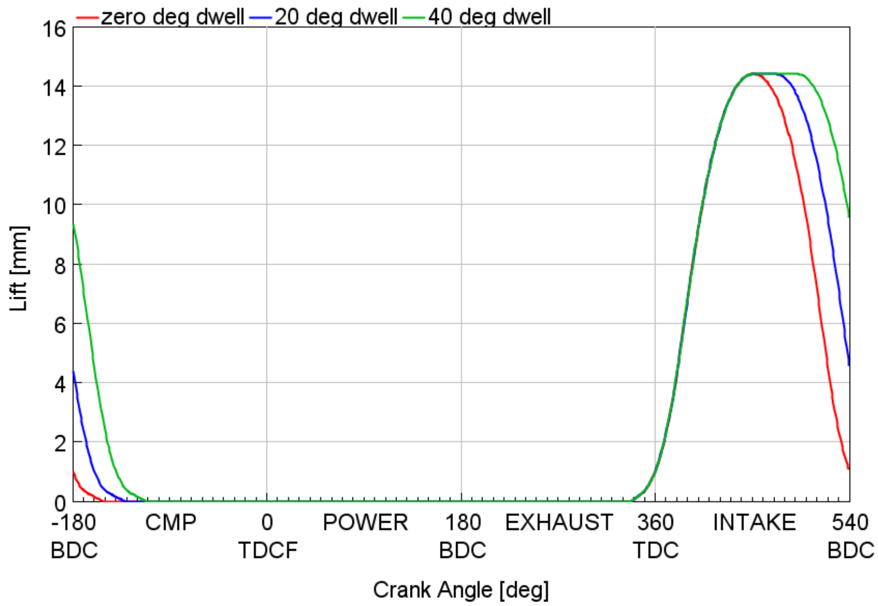


Figure 40: Valve lift comparison used for LIVC Miller timing concept analysis

A two-stage turbocharger can produce the same boost at much lower exhaust backpressure than a single-stage turbocharger. An intercooler between the compressor stages can help improve compressor performance.

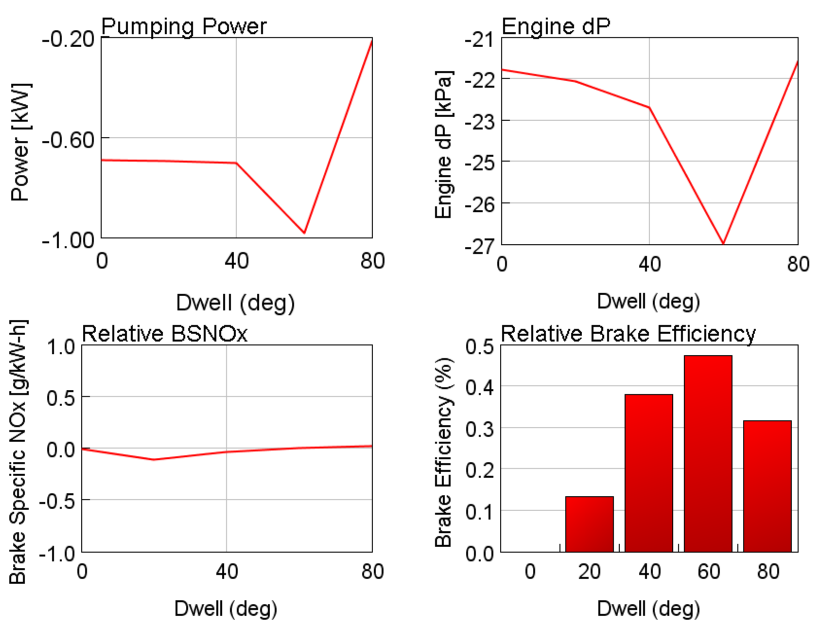


Figure 41: Pumping power, engine dP, relative brake specific NOx and relative brake efficiency, Configuration 4

In the figure above, pumping power is shown negative, meaning intake is at higher pressure than exhaust. This is added to the cycle work, resulting in further BTE improvement. Maintaining NOx level, 60-degree retarded IVC is shown to give the most BTE benefit by almost 0.5% while further optimization of air management system provided additional BTE improvement.

To validate the engine concept explored using simulations a Multi Cylinder Engine was operated for several weeks to examine the efficiency potential of different Miller cam configurations. Both Early Intake Valve Closing (EIVC) and Late Intake Valve Closing (LIVC) cam configurations were tested. To simulate the Borg-Warner 2-stage, high efficiency turbocharger system that was still under development at that point of time, PACCAR utilized an external “artificial boosting system”, and operated the engine with the expected boost and back pressure values as predicted in our 1-D simulation models. A 70% turbocharger efficiency was utilized to reverse calculate the corrected real-world BTE values for these configurations.

The data from all the cams tested is shown in figure 41 below. The stock camshaft with standard valve timing is the black line of the graph, listed as IVC_00. The number following either EIVC or LIVC is the number of degrees away from stock that the cam closes. For example, EIVC 50 means that the intake valves close 50 degrees earlier than the stock configuration. LIVC 50 means that the intake cams close 50 degrees later than the stock configuration. As can be seen in the graph, more aggressive Miller timing tends to produce the highest levels of BTE. For example, the LIVC 60 and LIVC 70 cams produce roughly 49.3% BTE. However, the aggressive Miller cams also had the negative effect of making the engine very difficult to start or operate at low speeds and low loads. As a result, a compromise valve timing needed to be found to allow for increased efficiency without harming the ability to start or operate the engine using the complete engine map. LIVC 40 was selected as this final design point.

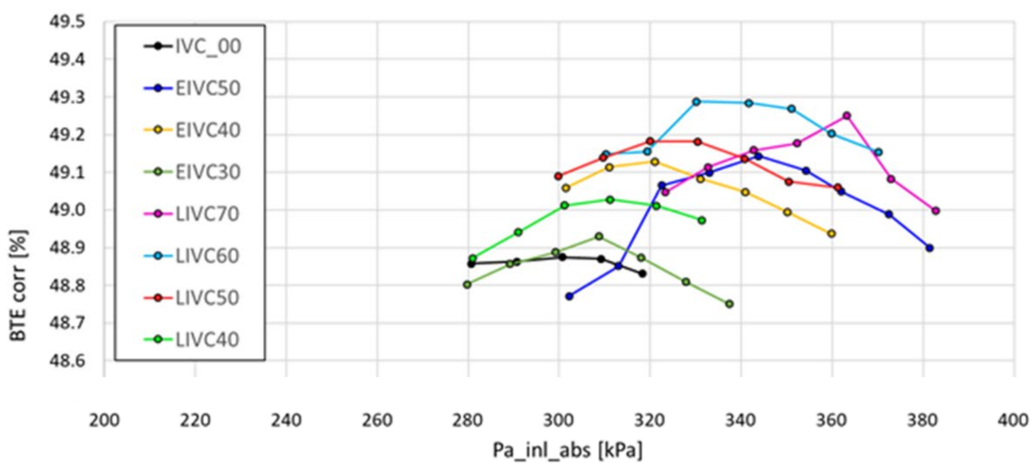


Figure 42: Miller camshaft multi-cylinder engine testing results

Long-stroke crankshaft

One direct method to increase the thermal efficiency is to increase compression ratio. However, this has a tendency to reduce the piston bowl volume, which reduces the air available during combustion. One way out of this is to increase the stroke. The efficiency benefit of a long-stroke concept gets further amplified when combined with the high-boosted Miller timing concept since it shortens the effective compression stroke while extending the expansion stroke for optimal work output.

The concept was analyzed in detail leveraging coupled 1D GT-Power and 3D CFD simulations. The baseline for the analysis work included the high-flow fuel injector resulting in the highest GITE. a longer stroke and late intake valve closing (LIVC) strategy combined. The simulations confirmed the hypothesis that the increased stroke generates more piston work with a higher compression ratio (CR) while maintaining TDC volume and LIVC reduces compression work. On the downside, a longer stroke increases CR, resulting in significant increases in peak cylinder pressure (PCP) and the limits on maximum allowable PCP must be considered.

Three stroke variants were simulated with three different Intake Valve Closure (IVC) timings. The base configuration features a 152 mm stroke and -133 after Top Dead Center (aTDC) IVC timing. Pressure at IVC was decided from the polytropic compression curve, Figure 19, starting from IVC = -133 aTDC. The original goal was to reach the same TDC pressure with IVC variants for each stroke setting, but this is only possible with higher boost. For instance, with 172 mm stroke and -103 aTDC IVC, in-cylinder pressure at IVC is required to be 4.87 bar. This presents significant challenges to the air management system. An iterative study is planned to find an optimal point that maximizes BTE while compromising with the other constraints. The total displacement volume for 165 mm and 172 mm strokes are 11.8 liter and 12.3 liter (respectively). The geometric CR becomes 20:1 (using 165 mm stroke) and 20.8:1 (using 172 mm stroke).

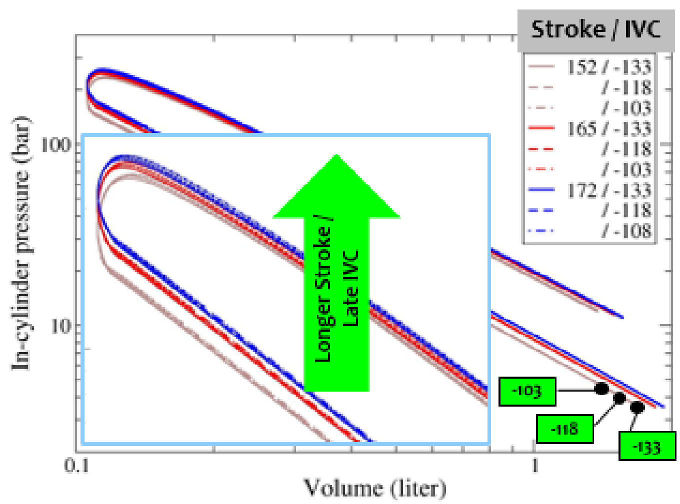


Figure 43: Stroke vs. IVC

The GITE vs. NOx plot shown in Figure 43 confirms that the longer stroke and LIVC combination can improve GITE while also reducing NOx. Within the simulation range (including SOI sweep), the predicted maximum GITE improvement was 2.6% while NOx was reduced by approximately 1 g/kW-hr.

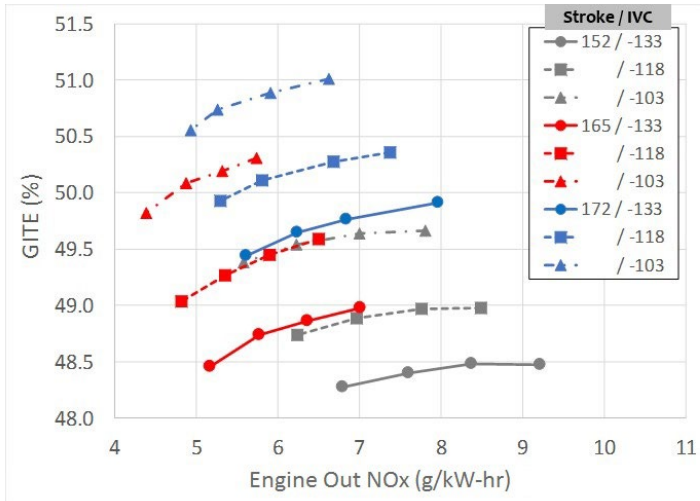


Figure 44: NOx vs. GITE

GITE improvement with the current recipe can be attributed primarily to increased expansion work and reduced heat loss to the combustion chamber walls, as a result of lower gas temperatures. As shown in Figure 44, the predicted averaged gas temperature is decreased by 140K compared to the baseline. This reduces the energy available to the turbocharger and WHR system.

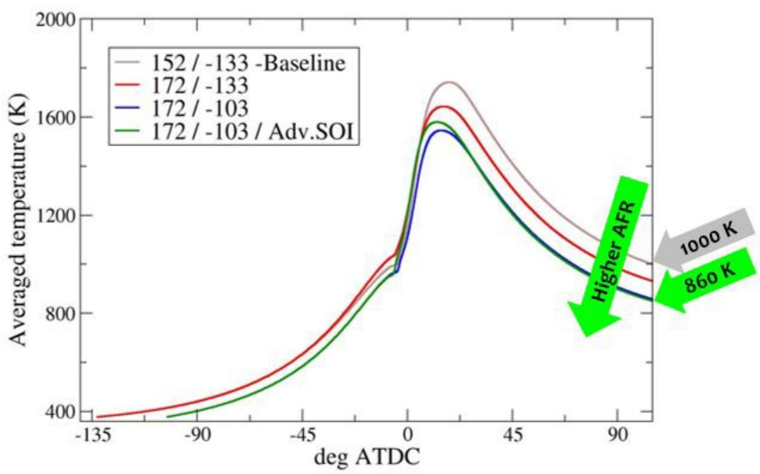


Figure 45: Averaged gas temperature

Single Cylinder Engine Testing

A key activity in the engine development process were the Single Cylinder Engine (SCE) investigations in to validate the cylinder-head, valve-timing and combustion system and long-stroke concept designs all together at relevant inlet boost and exhaust back-pressure operating conditions. Figure 45 shows the newly designed long stroke crankcase and gear train for the SCE dedicated for SuperTruck II work. In addition, the newly adapted camshafts for the long stroke engine are shown as well.

Figure 46: New long stroke SCE version of PACCAR MX-12 SuperTruck II engine (left), new Miller timing SCE camshafts (middle), and long stroke SCE test cell installation (right)

The engine was modified to the long stroke engine design (Figure 46), which converts the baseline PACCAR MX-11 engine platform. Comprehensive tests were performed with the long stroke configuration, including Miller cam hardware and different combustion system hardware configurations.

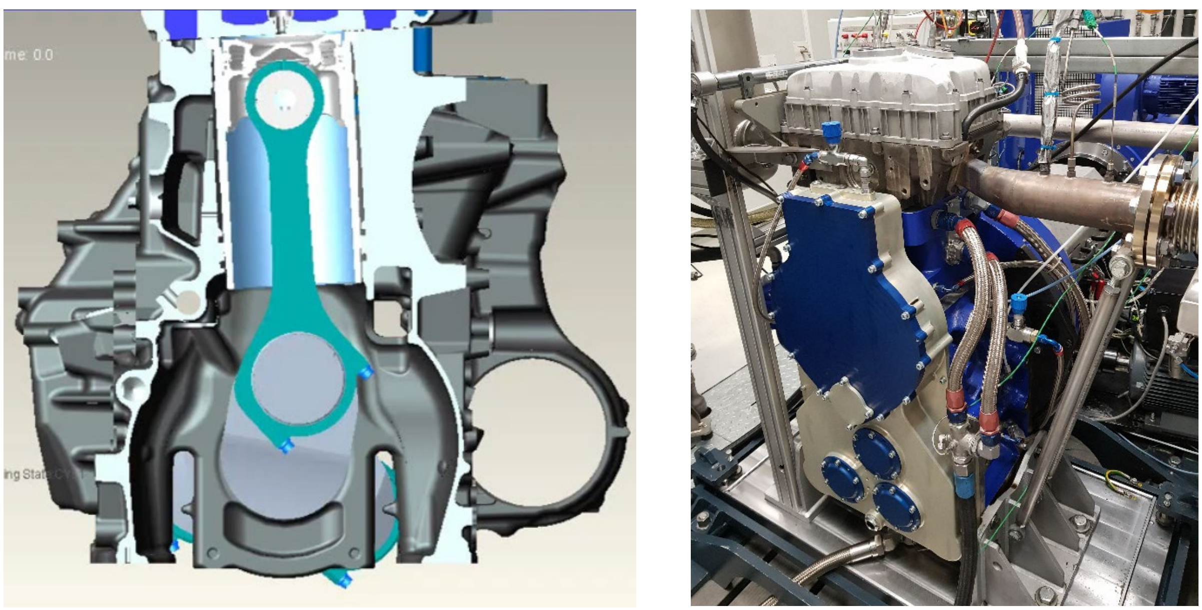


Figure 47: long stroke concept (left) and SCE test cell installation (right)

An overview of the test results, including earlier communicated data from the end of 2019, are shown in figure 47. In 2019 the SCE work ended with the optimized piston bowl (T1), high flow injector (3.4lpm - 7holes) and Miller cams 60 & 80. The maximum BTE reached was 49.5%. At the start of 2021 the stroke was increased from 152 to 167 mm and as a result the compression ratio increased from 20.8 to 22.8, while using the same T1 piston. With this setup the following hardware variants were tested at different boost levels and exhaust backpressure setting assuming 70% turbocharger efficiency:

- Miller 60, Miller 80 & Miller 70 (fast closing)
- Variation of hole numbers: 7, 10, 12
- Variation of nozzle flows: 3.4, 3.7, 4.0 lpm

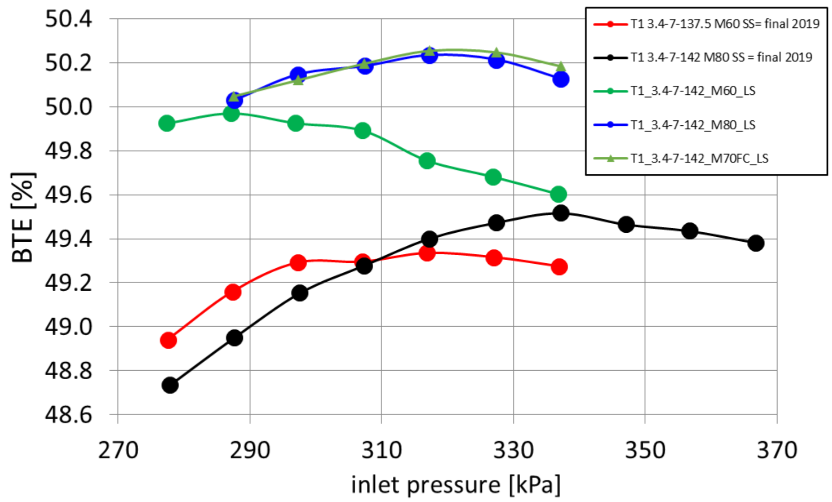


Figure 48: SCE test results showing the efficiency benefit of the long stroke concept

The long stroke engine achieves a 0.5%. higher BTE compared to the optimum found in earlier work, resulting in a peak BTE of 50.2%. Optimum BTE was found using the following hardware:

- Piston: T1, piston bowl design using combustion CFD under the SuperTruck II program
- High flow injector: 7 holes, 3.4lpm
- Fast closing Late Inlet Valve Miller Cam: 70 deg later compared to the baseline camshaft
- Long-stroke concept: +15 mm, based on the design changes as discussed in previous reports

The SCE proof of concept work supports the theory that using a higher compression ratio while keeping the same bowl volume, results in higher thermal efficiency. The additional work at the end of the combustion stroke as a result of the longer expansion stroke is clearly visualized in the p-V diagram plotted on a double logarithmic scale, as shown in figure 5. The red curve represents the long stroke concept and is compared to the black curve baseline using the same fuel input.

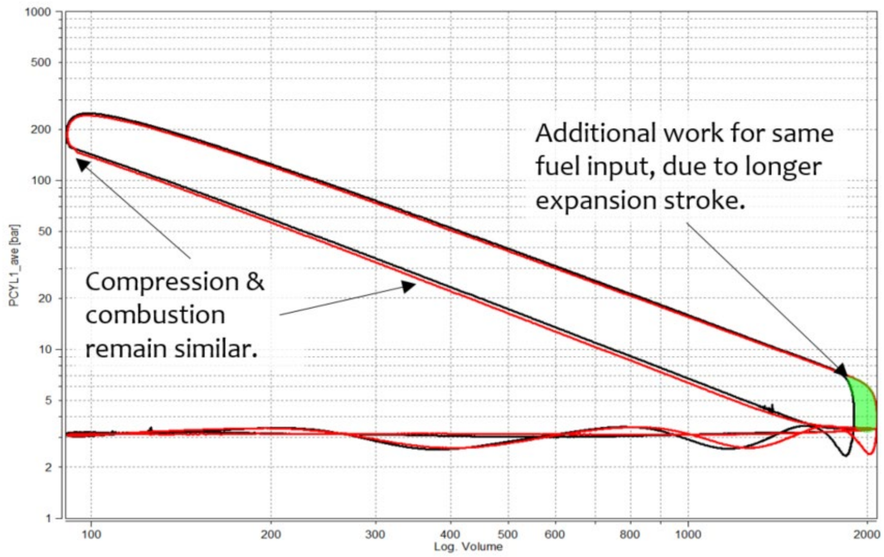


Figure 49: SCE long-stroke test result showing the additional work obtained during expansion

Turbocharging

The figure below shows the approach taken in the development of the air management system to optimize the gas exchange process.

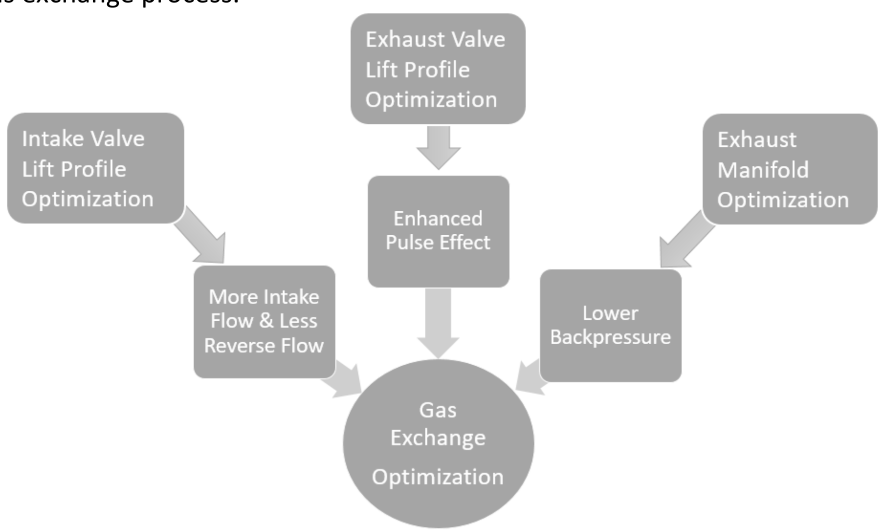


Figure 50: Modeling approach to gas exchange optimization

A major challenge in development of the ST-II engine concept is to design an air management system capable of supporting the high boost pressure required to maximize the gross indicated work. The GT-Power baseline engine model of the MY17 engine was modified to include ST-II concept (longer stroke and late IVC). The high pressure EGR loop was replaced with a low pressure EGR loop to reduce the exhaust backpressure before the turbine.

Boundary conditions were given to the turbocharger supplier Borg-Warner, who was responsible for the development of the 2-stage turbocharger for the PACCAR SuperTruck II engine. The operation point of the engine at the SuperTruck II point on the turbocharger stages map provided by Borg-Warner show that the proposed turbocharger architecture is well suited for our engine application, as can be seen in the figure below, with the SuperTruck II demonstration point region shown in blue ovals.

The two compressor maps, high-pressure and low-pressure, show that the operating points at the SuperTruck II point is close to the best efficiency. The low-pressure turbine map (bottom right) shows that the low-pressure turbine stage is also operating at an optimum condition. The high-pressure turbine stage is more difficult to consider as it operates at a high range of pressure and flow due to its location but it also mostly operates at the map’s best efficiency zone.

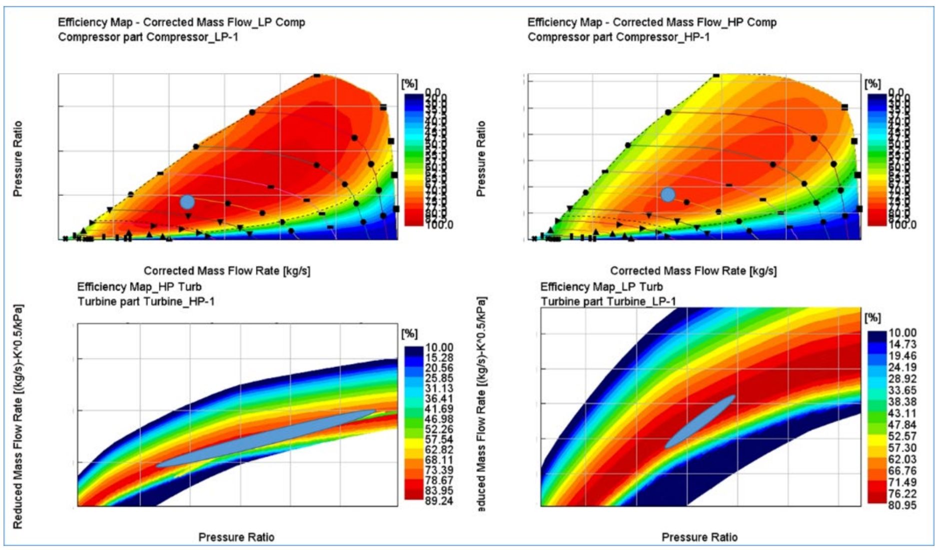


Figure 51: Turbocharger Maps and SuperTruck II Engine Demonstration Point

To further improve overall turbocharger efficiency, the PACCAR team investigated a pulse conserving manifold through simulation and bench testing. Exhaust pulse conservation occurs when interference between exhaust phases is avoided and expansion through the turbine is maximized. As one exhaust event begins, producing a blow down in the exhaust manifold, another cylinder is in the latter part of its exhaust phase, thus having to produce more pumping work to finish its exhaust process.
In the case of an in-line 6 cylinder, a simple separation of cylinders in the middle of the bank is enough, i.e., cylinders 1, 2 and 3 do not interfere with each other and can be grouped in the same half manifold, the same goes for cylinders 4, 5 and 6. The exhaust flow needs to be separated all the way to the turbine to avoid interference between cylinders. It is expected that this set-up can provide some significant improvement in the energy available to the turbine.

Exhaust manifold geometry was modeled and used in the GT Power model. The turbocharger was also set up with two entry ways. Both the model and the prototype test setup were run to compare the turbine inlet condition with a pulse conserving manifold and without. To validate the pulse capturing concept the team completed the design, build, and commissioning of a split manifold and integrated a twin-scroll turbocharger to the baseline model year 2018 engine in the engine testbed. Figure 4(a) shows a CAD schematic of the designed split-manifold integrated with a twin-scroll turbocharger. Figure 4(b) shows the same hardware installed in the test cell with its associated instrumentation.

The concept of pulse-turbocharging is to recover the exergy of the high pulses which would otherwise be lost in mixing. The “no pulse” pressure trace was recorded by installing a spacer between the manifold and the twin-scroll turbocharger allowing the pressure equalization between the two banks. The results of these tests are being used to validate simulation models to help co-optimize the manifold and turbocharger for effectively utilizing pulses at the ST-II point.

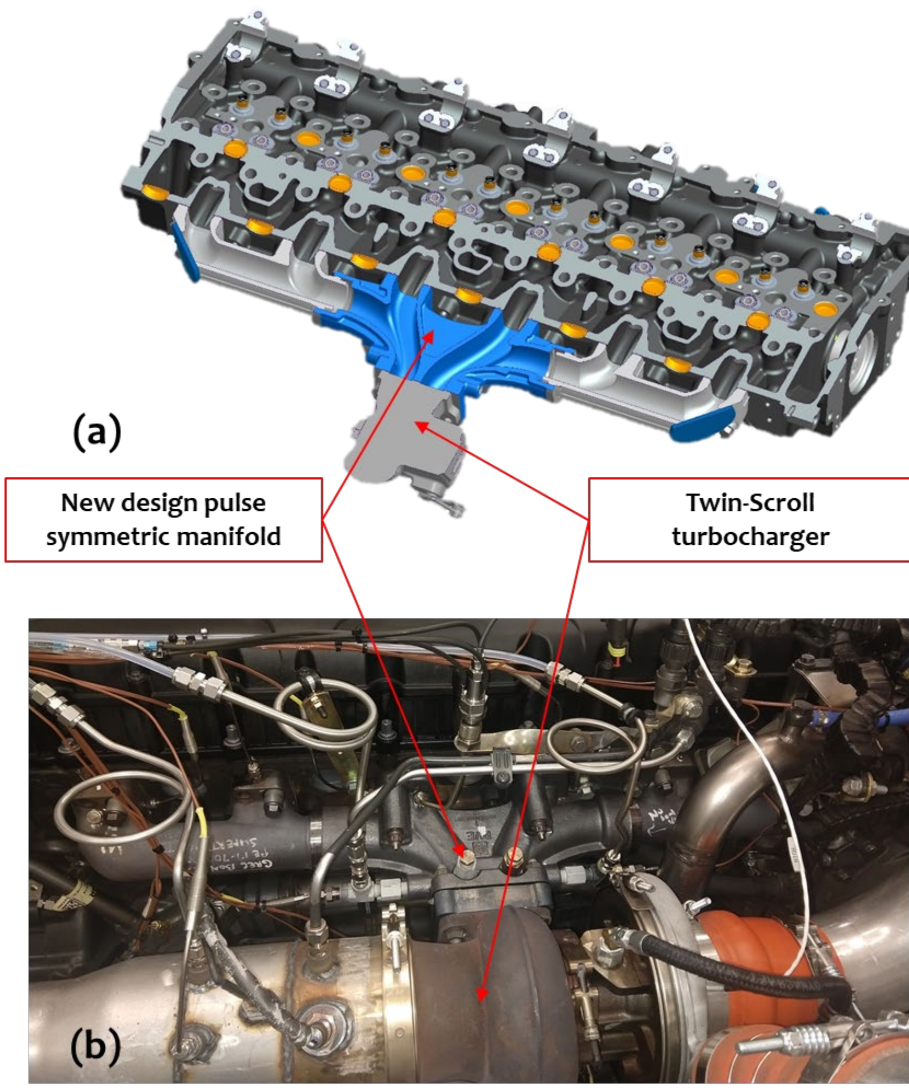


Figure 52: (a) CAD Visualization Of Split Manifold And Twin-Scroll Turbocharger Installation On The MY17 Engine (b) Image Of Test Cell Installation Of Turbocharger With Instrumentation

It was found that separating the engine exhaust in two banks allowed for an improved separation of the exhaust pulses. Additionally, as each side of the exhaust is smaller than the whole system, the exhaust pulse amplitude becomes larger. The team investigated the effect of changing the exhaust manifold internal diameter, thus further changing the exhaust volume.

Three exhaust manifold CAD models were used to generate three GT Power models of manifold with various internal diameters: 44.3mm (baseline), 40mm and 35mm. Other features of the manifolds were kept unchanged. The manifold models were then run in GT Power at the SuperTruck operating point. As each manifold could impact how the turbine would operate, it was decided to add a sweep of turbine sizes using the mass scaling factor. Performing a sweep of turbine sizes with each manifold allows for a better assessment of their impact, should the optimum turbine size be altered by the manifold.
All the manifolds investigated show a pulse factor (β) that increases with an increase of the turbine size. As β is always positive, it suggests that the real expansion work available to the turbine is higher than would be seen from time averaged values such as on-engine measurements.

Furthermore, all the manifolds investigated showed a peak BTE for the same turbine scaling value. As the turbine model is based on a single map, there may be some inaccuracies regarding the modeling of dynamic gas flow through the turbine. Finally, the 44.3mm and the 40mm case show the best peak BTE. The accuracy of the BTE prediction will improve when the twin scroll model is used for the HP turbine. These results show that the exhaust geometry can influence the amplitude of pressure pulses in the exhaust. The metric developed to evaluate the presence of pulse in the exhaust confirms that the turbine inlet conditions are impacted by the exhaust manifold diameter. We also found that the turbine size can influence the exhaust pressure pulse and the efficiency with which they are used. The result is that the optimum BTE is not necessarily reached at the highest air flow, but by balancing improved pumping and moderately reduced GITE.

As concluded from the investigations described above, a two-stage turbocharger system combined with pulse conserving manifolds are both required to reach the optimum pressure ratio, leading to maximum engine efficiency. For the final SuperTruck II demonstrator engine the turbos are mounted 90-degree to each other for the straightest engine exhaust routing. The exhaust manifolds have a reduced volume to reduce dynamic to static pressure losses. Low flow loss & dual core intercoolers were developed for the Waste-Heat-Recovery system heating and airside temperature reduction (see WHR section for more details). Insulated ceramic spacers between the cylinder head and ceramic coated exhaust manifold are being evaluated to minimize exhaust heat loss.

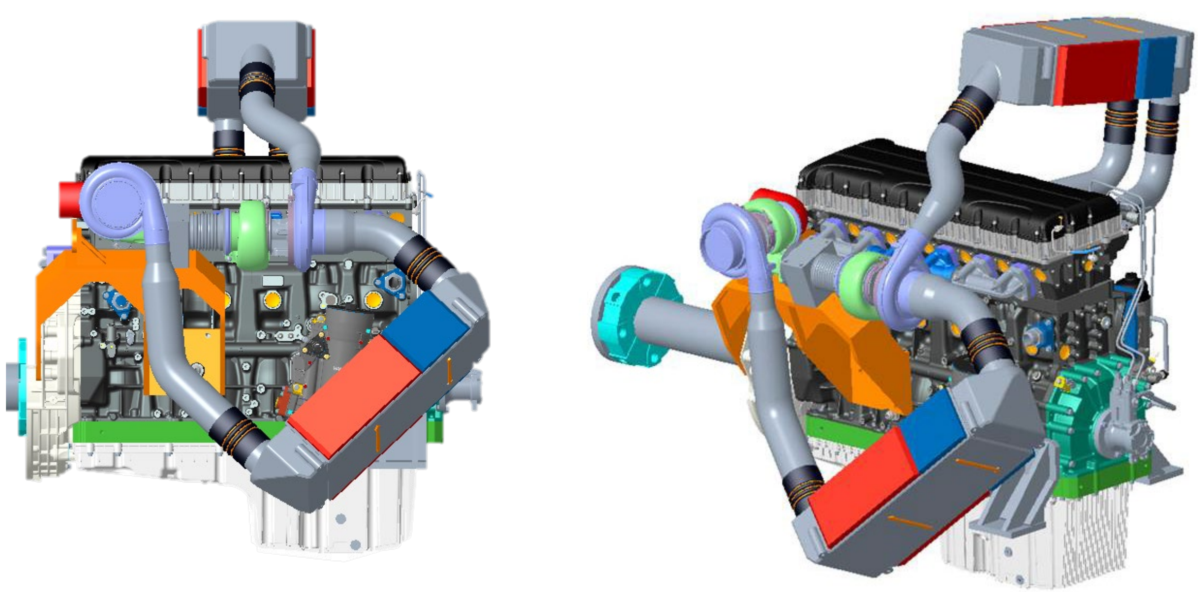


Figure 53: Two Stage Turbo System with Charge-Air-Coolers

EGR Pump

Eaton's EGR pump is a modification of Eaton's TVS Supercharger product, with new designs to resist soot fouling and corrosive combustion byproducts. Using an electric driven EGR pump results in a positive displacement device. The 48V pump used for the SuperTruck II engine demonstrator is optimized for use in exhaust conditions appropriate for pumping EGR but is fundamentally the same as a roots-type supercharger. The EGR pump is driven electrically off the 48V bus and can be controlled via a CAN bus speed command. This ability to vary pump speed allows precise control of the flow of exhaust gas to the intake manifold according to engine operating point.

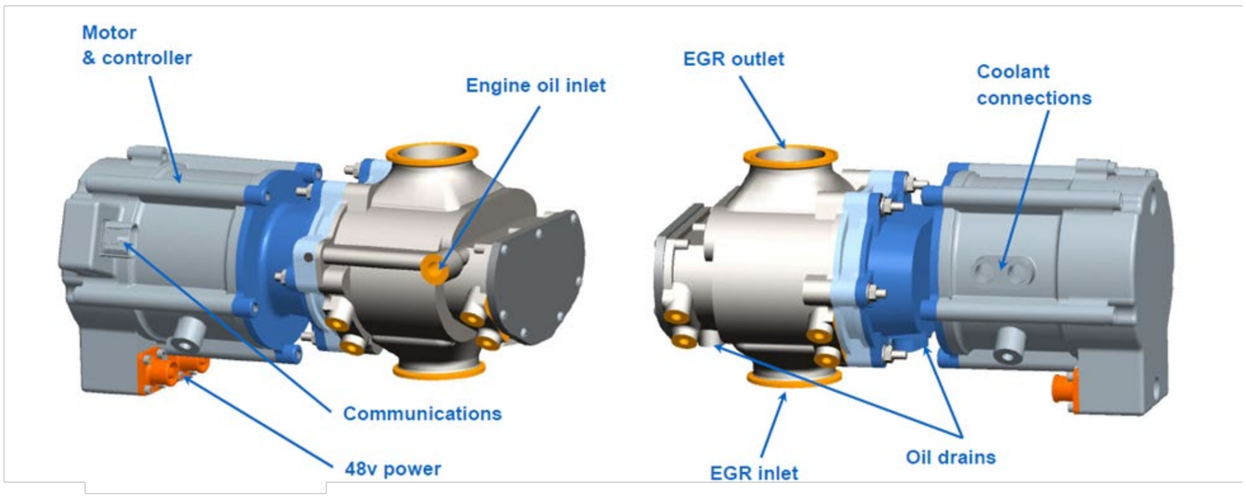


Figure 54: Eaton's EGR Pump

In boosted diesel engines, the vane position in a variable geometry turbocharger architecture is used to control boost as well as EGR demand. As a result, independent control of boost and EGR is not feasible and the engine obtaining the correct boost and EGR for efficient operation becomes increasingly challenging. Replacing the VGT with a fixed 2-stage geometry turbocharger and a separate EGR pump was the selected architecture by PACCAR to improve both turbo-charging efficiency while maintaining EGR flow controllability.

During the program EGR pump prototypes were built and characterized. The EGR pumps were bench tested and a control system was developed. Controls development included: CAN communication, volumetric flow table calibration, EGR control calibration. The figure below shows the test rig developed to test the EGR pump controls. The rig consists of a bi-directional power supply, EGR pump with micro-autobox interface and a forced oil and coolant supply. The outlet of the pump was looped back to the inlet for reducing noise while the pump was in operation.

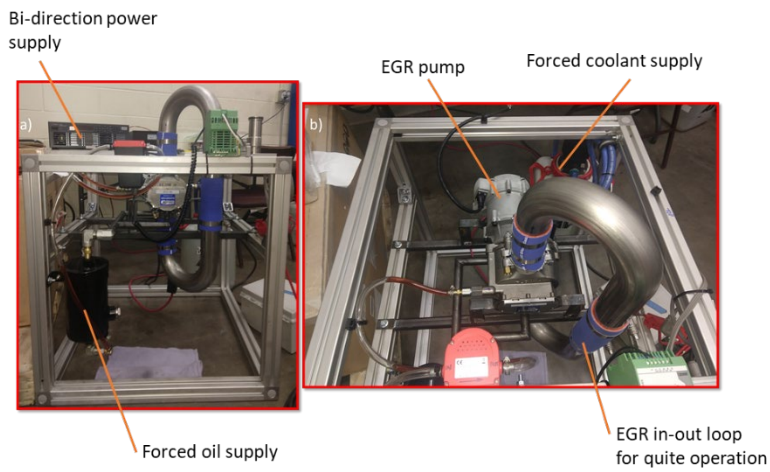


Figure 55: EGR controls layout

Engine Block Design

A new engine block was designed to improve gross indicated thermal efficiency with a longer stroke crankshaft that will increase the compression ratio from 18.5 to 20.2 [-] with the existing bowl volume. The block was extensively redesigned by extending the deck height by 40mm and tilting the cylinder liners by 2-degrees with a 16mm crankshaft offset, which produces an angled fire deck. The crankshaft offset results in reduced piston skirt friction at high cylinder pressures and low engine speed, i.e. the main SuperTruck II operating range.

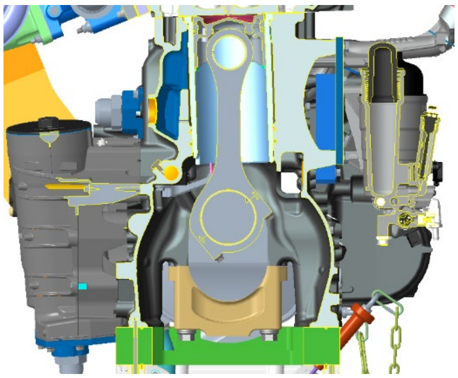



Figure 56: Block Deck Height Increase with Angled Fire Deck

The increased deck height requires a larger diameter block-to-head idler gear. The larger gear covers three existing flywheel-housing bolts that are being replaced with an additional bracket to clamp the production flywheel housing.

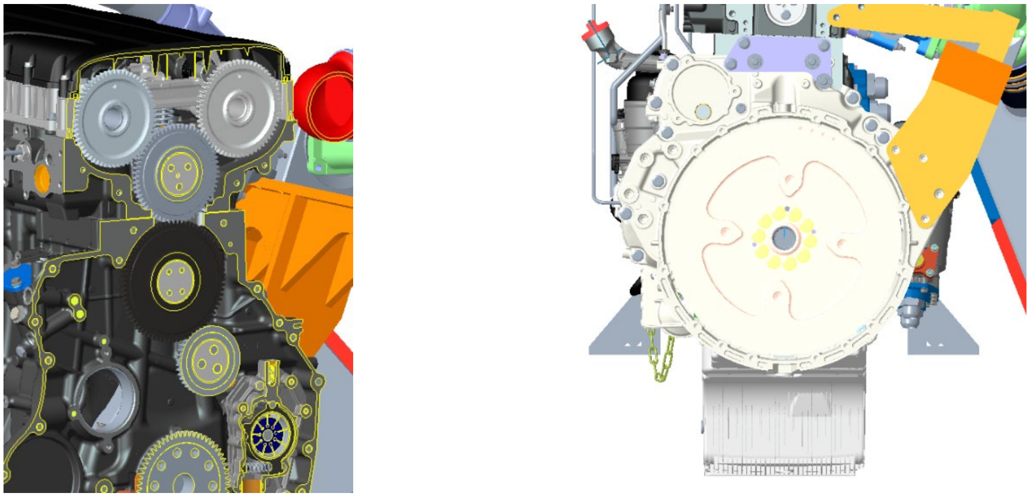


Figure 57: Increase in Deck Height Idler Gear and Flywheel Bracket

The electrified accessories allow the FEAD to be deleted. To take advantage of this, the new block design will include a new front cover that encloses the crankshaft damper and eliminates of the front dynamic crankshaft seal. Removal of the crank seal results in a calculated ~100 Watts of reduced rotating friction. The front cover will also mount a new crankshaft driven high efficiency fuel pump with ~175 Watts improvement at the ST-II point versus the current design. The new design also removes the fuel pump cam gear, which is projected to save a further ~20 Watts at the SuperTruck II point. The fuel pump design required a new high-pressure and low-pressure lines, and repositioning of the fuel rail and filters.

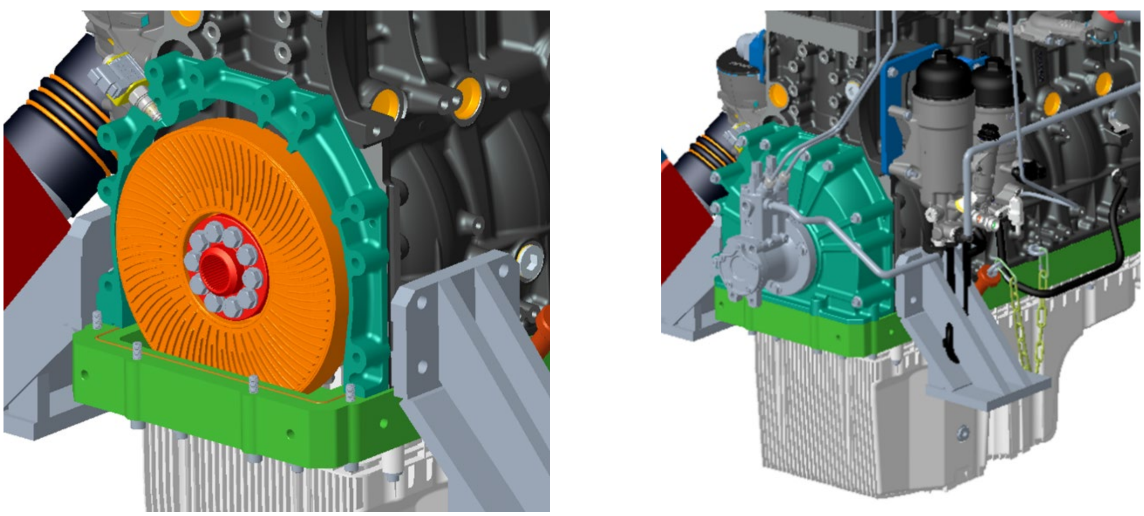


Figure 58: Front Cover / Damper & Crankshaft Driven Fuel Pump

Viscous Damper

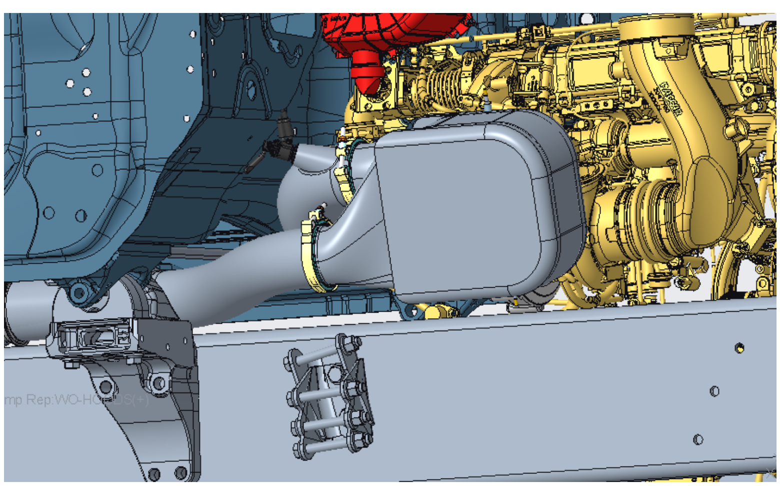
Within the SuperTruck II program, Geislinger was asked to develop a crankshaft damper for the 55% engine demonstrator. One of the initial challenges with looking at the viscous damper were the temperatures inside the engine cover and the heat generated by the viscous damper. The viscous damper is filled with a silicon fluid and is more sensitive to heat, so typically in these cases fins are added to the surfaces to increase the flow of air around the damper. The increased air flow allows for a greater heat dissipation around the damper. It was that at the operating speed for the system the damper still had a high thermal load and so the development began for a completely new viscous damper design. This new innovative design incorporates fins into the housing that allows additional cooling for the viscous damper so that a higher heat dissipation can be achieved during operation. The new optimized viscous damper meets the torsional requirements and hardware is now available for the SuperTruck II engine build.

Figure 59: Viscous Damper Production Piece

Ultra-Low NOx Aftertreatment

The SuperTruck II includes demonstration and implementation of an Ultra-low NOx aftertreatment system. The development efforts for the SuperTruck II demonstrator are related to the installation and specific design of the metallic SCR substrate. Although the Ultra-Low NOx (ULN) concept doesn't contribute directly to the freight efficiency goals of the SuperTruck II program, the team recognizes that challenges for future Diesel powertrains include both freight efficiency and ULN emissions.

The goal set by the PACCAR team is therefore to meet the most stringent upcoming ULN legislation target: CARB 2027 -90% tail pipe out NOx. There are two pathways explored: GCI (on a testbed demonstrator) and close coupled aftertreatment system (SuperTruck II vehicle demonstrator). It is worthwhile to mention that both approaches were already included in the original program scope.



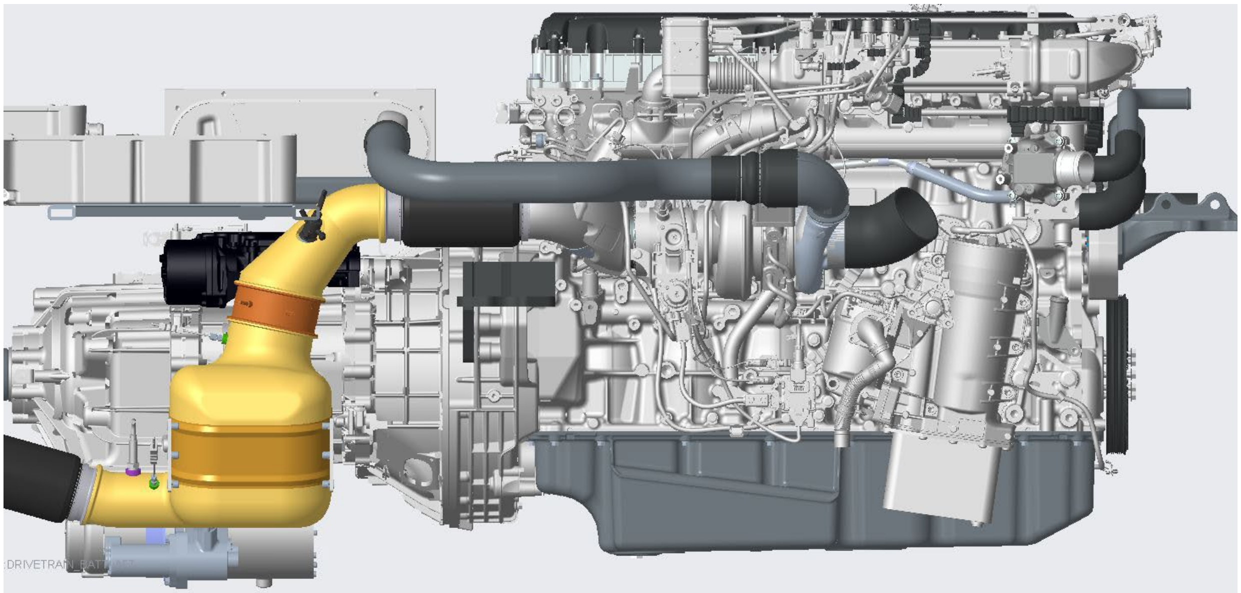


Figure 60: Close coupled aftertreatment design for existing PACCAR vehicles (upper) and optimized for the SuperTruck II vehicle (lower)

Gasoline Compression Ignition

Gasoline Compression Ignition (GCI) is explored in this project because of the high potential, based upon the literature, to achieve very high BTE while using a commercially available pump fuel. This effort is being undertaken in parallel with the main thrust of the PACCAR ST-2 engine effort. The opportunity to create a combustion system that provides extremely rapid combustion coupled with very low (NOx and PM) emissions signature is compelling, despite the known challenges to GCI systems. PACCAR contracted with the Wisconsin Engine Research Consultants (WERC) to perform some screening simulations, first in 1-D and later in 3-D, to determine which combination of operating conditions will lead to the optimal fuel efficiency for the GCI concept.

The simulation work executed by WERC provided analytical support to establish performance expectations and hardware/calibration recommendations for GCI engine development. By using 3-D CFD combustion modeling tools, engine hardware configurations (compression ratio (CR), intake valve closure (IVC) timing) and calibration settings (air fuel ratio (AFR), exhaust gas recirculation (EGR) percent, intake manifold temperature (IMT), injection strategy) were optimized to enable high (>51%) engine BTE while operating in GCI mode.

Figure 61 shows the DOE results with respect to GITE and PCP (including the best case found in the previous DOE). For the cases with the most retarded IVC, there was insufficient cylinder pressure/temperature for ignition and misfires then occurred. Conversely, the highest efficiency was obtained with the highest IVC pressure (ie, highest boost).

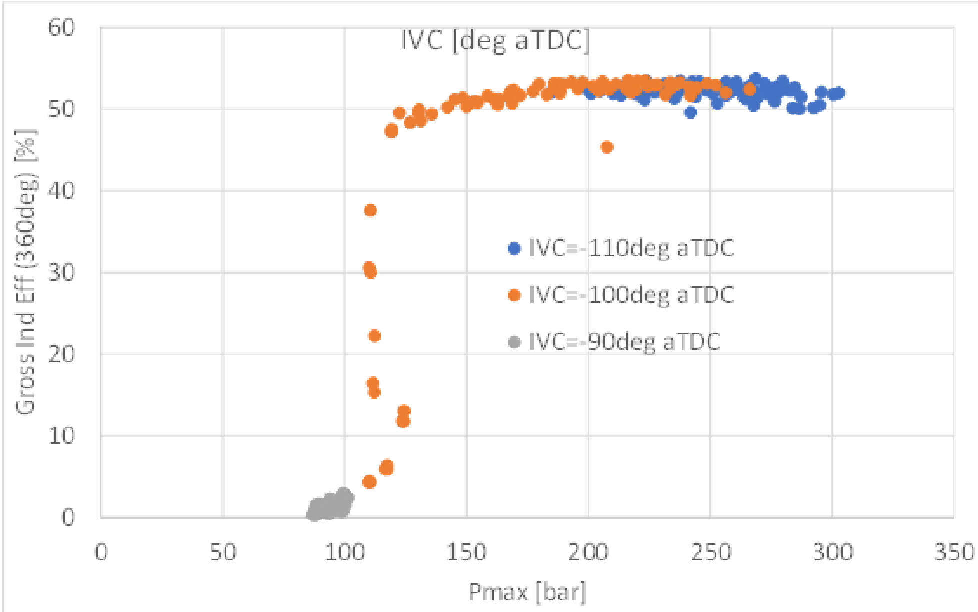


Figure 61: DOE results to investigate sensitivities of engine parameters

The GCI simulations provided possible hardware and calibration settings to enable very high engine indicated thermal efficiency while operating in GCI mode for the SuperTruck-II engine reference point. It was found that the best GITE could be achieved by CR=18, high IVC pressure (4 bar), minimum intake temperature (T_{IVC}=72 °C), 10-15% EGR configuration with mid IVC timing (-100 CA° ATDC). It was also found that a split injection strategy performed better than that for the premixed charge. While the first injection timing was fixed to IVC timing (with ~70-80% of the total fuel mass), the optimal timing for the second injection was found to be near TDC. The six best points were then identified, providing the highest GITE with reasonable PCP values. Peak GITE values are between 53.0% and 53.8% when placed into the PACCAR calibrated GT-Power model for brake efficiency.

Based on the simulation results PACCAR decided to continue GCI investigations in parallel under the program realizing that the Diesel combustion concept would remain prime path for the SuperTruck II engine demonstrator. PACCAR partnered with Aramco who provided the test-cell and engineering support. The goal of the experimental project was to implement GCI on the PACCAR MX-11 Diesel engine platform, and leverage GCI's fundamentally low soot-NOx trade-off characteristics to implement efficiency improving strategies on the engine (higher compression ratio and late intake valve closure) to offer simultaneous reduction in engine-out NOx and CO2 emissions. The technical targets for the project are to offer 3% CO2 reduction while reducing engine-out NOx to 2 g/hp.h (>50% reduction compared to MY-2021 production engine).

A closed-cycle 3D full geometry simulation was performed from intake valve closing (IVC) to exhaust valve opening (EVO). Figure 62 shows the selected validations between 3D CFD simulations for conventional diesel combustion (CDC) case compared with experiments operated in Detroit Research Center (DRCC) under low NOx BSNOx condition. Under Super-Truck 2 (ST2) operating condition, the 3D CFD model capture overall pressure trace and heat release rate profiles accurately. Similar agreements were observed under other operating conditions (e.g., B50, B100) and other dilution (or BSNOx) levels, demonstrating that the 3D model with boundary conditions were reasonably set-up.

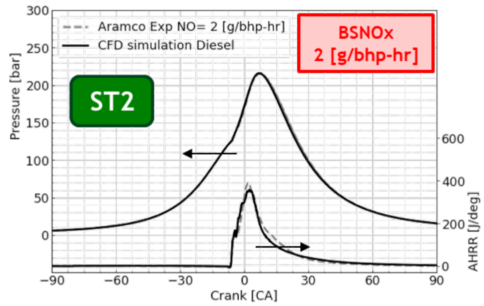


Figure 62. Selected 3-D CFD model validation against Aramco experiment with CR16.5 under Super-Truck 2 operating condition.

With successful model validations, the baseline diesel case was calibrated for low NOx condition (2g/hp-hr). Starting from the reference boundary conditions from 1D GT model, start-of-injection (SOI) timing and exhaust-gas-recirculation (EGR) level were optimized for 2g/hp-hr ISNOx condition. CA50 was fixed at ~5 aTDC degrees. For GCI combustion recipe, two split injections with partially premixed combustion were utilized. Thus, pilot injection timing (SOI_{pilot}) and main injection timing (SOI_{main}), both of these two timings were adjusted together with splitting ratio (Q) during optimization. Figure 63 shows a schematic diagram of piston design optimization with geometry parameterization. The piston bowl profile was characterized by several design variables (independent variables) and compression ratio was monitored as a target metric. The independent variables can be varied within ranges of interests, and CAESES was used to automatically generate CFD-ready design files. Two examples set of design variants were displayed where Rim Radius was effectively varied under the fixed piston angle (Design Set 1), and fully varied variants (Design Set 2).

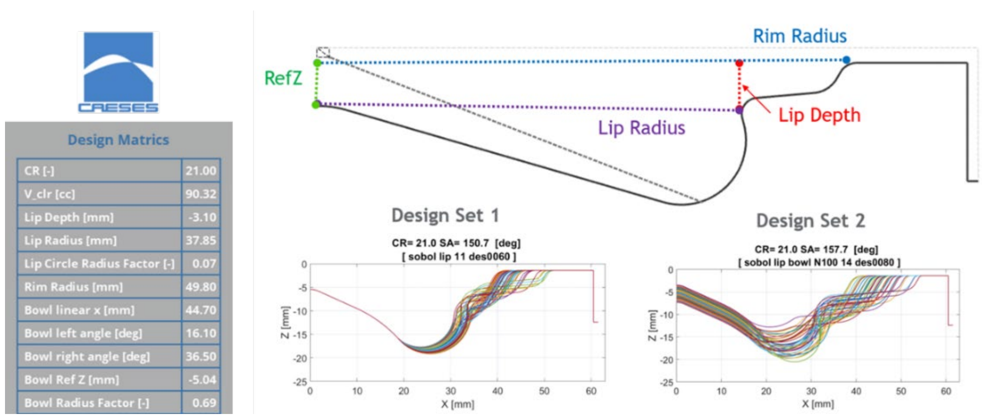


Figure 63. Characterization of piston features with automated Design parameterizations using CAESES.

The modified pistons with a geometric compression ratio of 21 were installed on the PACCAR MX-11 engine with a LIVC-40 CAM, resulting in an effective compression ratio of around 19. No changes were made to the air handling system of the base engine (turbocharger, EGR system). The modified engine was tested with Gasoline Compression Ignition (GCI) under steady state operating conditions of the super truck evaluation point (ST-2), B-50, B-75 and A-100, to assess fuel efficiency improvements over the base engine at engine-out NOx emission of 2 g/hp.h (~ 2.7 g/kW.h).

Figure 64 below provides an overview of GCI and CDC combustion using key parameters important from efficiency and after treatment perspective. GCI combustion on the modified engine demonstrated 4% lower CO_2 emissions compared to the base Diesel engine at 2 g/hp.h engine-out NOx. At fuel rail pressure fixed at the production calibrated level, smoke emissions were ~ 20% lower with GCI compared to CDC, which may be further reduced using higher fuel injection pressure. Compared to the production Diesel engine, the modified engine operating with GCI exhibits significantly lower relative air-fuel ratio, required relatively less EGR increase for low-NOx operation, and consequently, resulted in ~ 40°C higher turbo-out exhaust gas temperature. The lower exhaust mass flow (i.e. lower ATS space velocity) and higher feed gas temperature are beneficial for ATS performance.

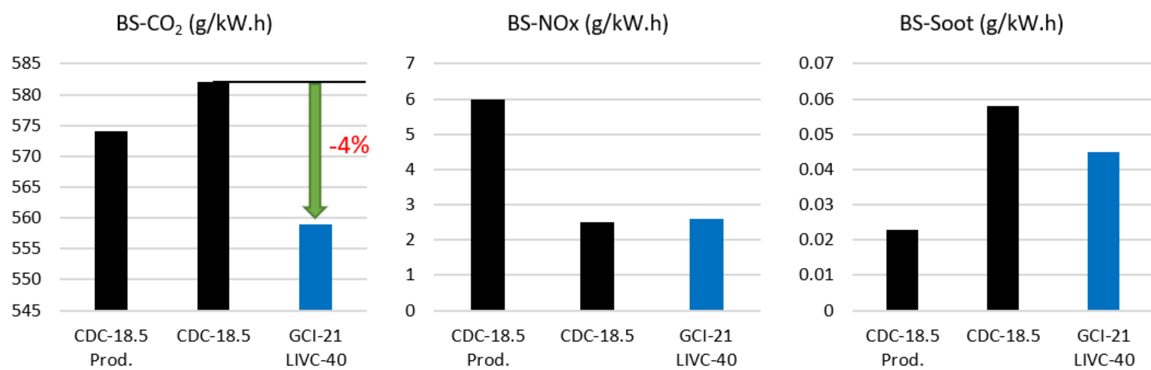


Figure 64. Performance overview at ST-2 operating condition (1100 rpm, 1600 Nm)

Waste Heat Recovery

Detailed engine simulations performed at the beginning of the program revealed that the 55% BTE target is not achievable with engine technology improvements alone. A Waste Heat Recovery (WHR) system based on the Organic Rankine Cycle (ORC) is required to achieve the program goals. After a detailed supplier evaluation, PACCAR partnered with Cummins in 2019 to develop a WHR system tailored to the PACCAR MX SuperTruck II engine. Based on the complexity and added weight of the WHR system, it was concluded early in the program that a pathway towards commercialization is not feasible in the foreseen future and therefore engine efficiency gains were prioritized over WHR efficiency contributions. The only exception to this statement is the engine coolant temperature since it shows a strong trade-off for WHR system efficiency improvement while having a minimal impact on engine efficiency and it doesn’t result in engine design changes either. Based on the first trade-off studies performed by PACCAR, the BTE% contribution target for the WHR system was set to 4% i.e. 14.5 kW net power at the by PACCAR provided engine conditions.

The overall WHR architecture selected to support this ambitious goal is shown in Figure 65. The WHR system has a unique architecture since it is based on a dual-loop ORC consisting of a (1) low temp, low pressure and (2) high temp, high pressure loop to optimize the efficiency of the system working at different conditions. The HPL (high pressure loop) is optimized for high-temperature sources and will utilize heat from the engine exhaust, extracted in the WHR tailpipe boiler. The LPL (low pressure loop) is optimized for heat sources at moderate temperature and will utilize heat from the charge coolers, engine oil cooler, and coolant heated boiler. The system includes a recuperator in the HP loop to recover some remaining low-grade heat from the combined turbine discharge flow, prior to the condensation process. A dedicated gearbox and dyno will be used to measure the system power output. A chilled water coupled off-the-shelf condenser will be used in the test cell for heat rejection, simulating the condenser capacity available in an optimized class-8 vehicle. The WHR system is using two feed-pumps and a unique dual-entry turbine expander. The working fluid selected, based on experience from Cummins with various WHR development program sis R1233zd(E) which is a Hydrofluoro-olefins(HFO) class of refrigerant from Honeywell. The benefits of the fluid are that it is non-flammable, ozone friendly and non-toxic. The key drawback of this refrigerant is its global warming potential if burned, and therefore any leakage towards the combustion system of the engine through the charge air coolers needs to be avoided. With the architecture selected Cummins started the design work of component sizing based on extensive simulations.

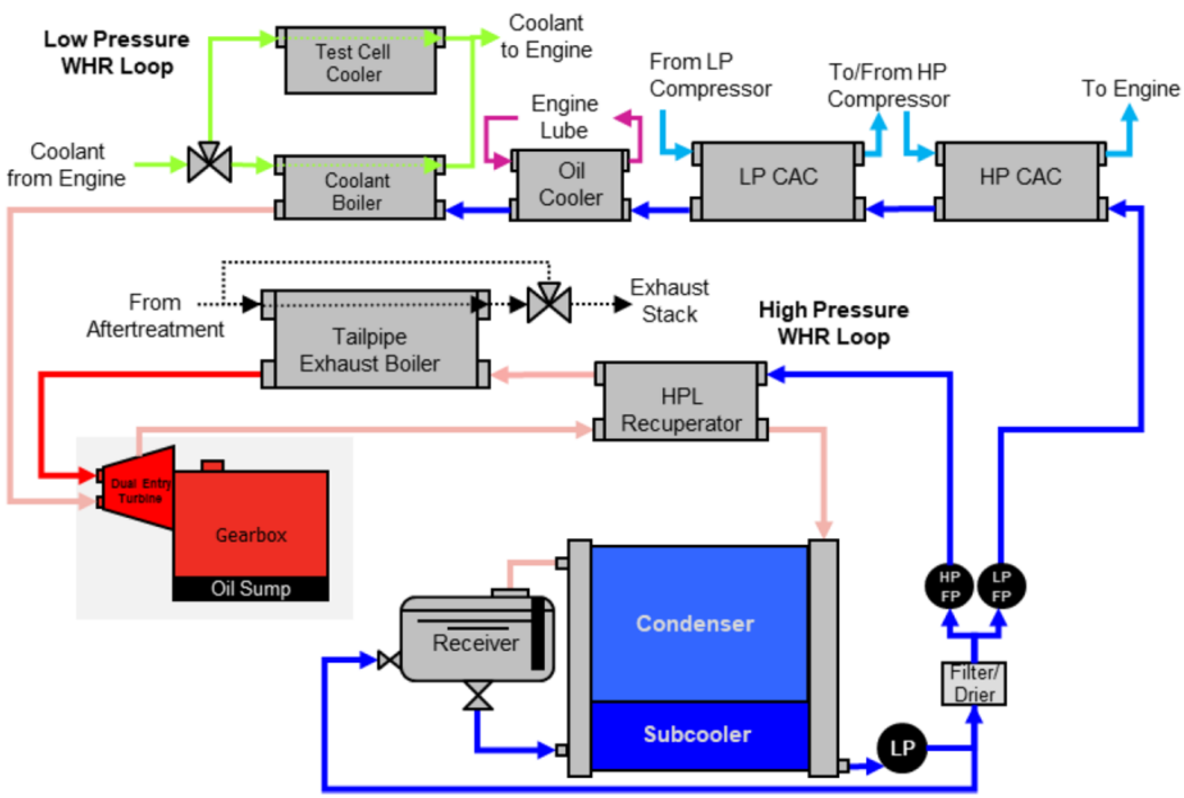


Figure 65: Figure X Cummins WHR architecture tailored to the PACCAR MX SuperTruck II Engine

The Engine and WHR teams worked together to define the CAC sizes. Both LP and HP CACs feature dual cores, with a WHR core providing partial cooling of the incoming charge air and a water core providing the finish-cooling. The design supports the engine inlet temperature target for maximum engine BTE, without driving the WHR system to operate in a less efficient mode, as would be required to achieve additional charge cooling. Based on specifications provided by the engine simulation team as well as specifications from the finalized WHR architecture study, core sizes for the CACs have been determined. Due to the relatively similar cooling requirements for both the LPCAC and HPCAC, common core sizing is possible.

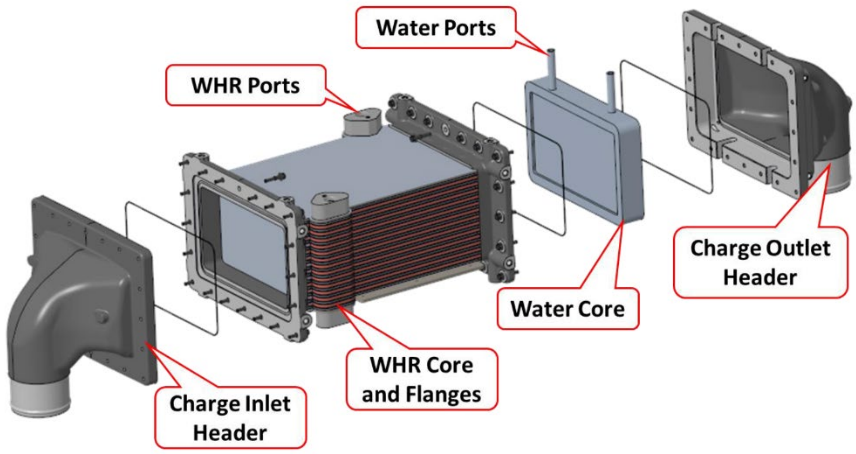


Figure 66: Dual-core CAC exploded assembly

The tailpipe boiler design is shown in Figure 67. A bypass routing is implemented to extend the lifetime of the boiler when WHR is de-activated. The proposed material of the exhaust boiler is a no carbon-steel or iron to avoid corrosion of the components. A Diesel Particulate Filter (DPF) has been installed upstream the exhaust boiler to avoid degradation of system performance due to internal soot build up. To ensure maximum performance all exhaust components upstream the tailpipe boiler will be insulated as much as possible.

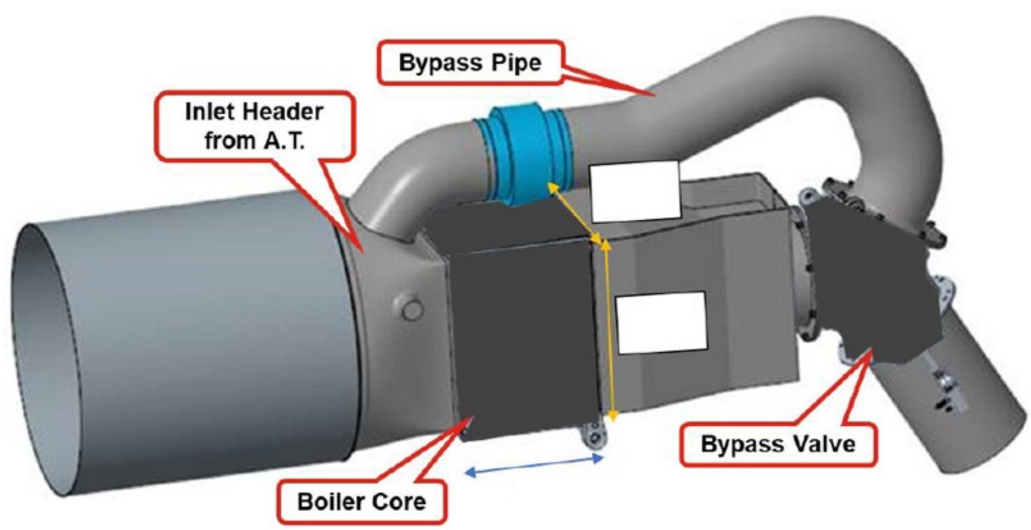


Figure 67: WHR exhaust boiler design

For the dual entry turbine expander: aerodynamics, thermal, and rotor-dynamics studies were completed, and the design was tested on dedicated test-rigs at Cummins to ensure performance met the simulation models. Testing activities included both the WHR turbine and gearbox hardware, necessary for experimental evaluation of turbine expander performance. The used testing set-up at Cummins is shown in figure 68. In this experimental setup, existing WHR hardware and engine was used to simulate PACCAR operating conditions at turbine interfaces.

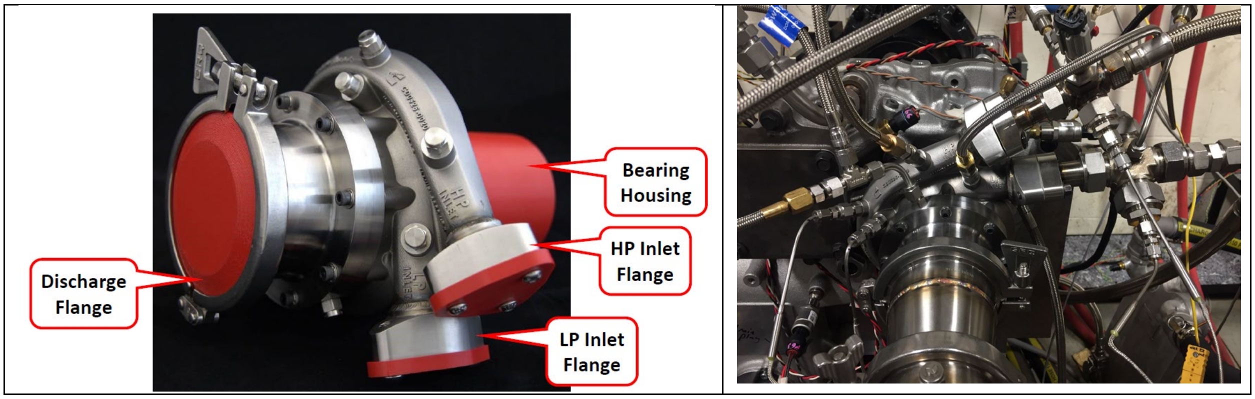


Figure 68: WHR Dual-Entry Turbine Expander

Coolant Boiler

The WHR coolant boiler is the primary source of heat for the LP working-fluid loop. WHR working fluid, which is partially pre-heated in the CAC WHR cores, enters the coolant boiler in a subcooled state where heat is transferred from the engine coolant to the WHR working-fluid. The working-fluid is heated to saturation, fully vaporized, and exists the coolant boiler in a superheated state, suitable for the expansion process. Engine coolant, providing heat to the WHR system, is maintained in a liquid state during this process. The coolant boiler design employs a brazed aluminum bar-plate construction. The core design has been finalized and procurement of the coolant boiler has been started.

Tailpipe Boiler

The tailpipe boiler is the primary source of waste heat for the HP working-fluid circuit. This heat exchanger recovers heat from the engine exhaust gases, downstream of the aftertreatment system. Working-fluid preheating, boiling, and superheating processes are completed in the tailpipe boiler, similar to the coolant boiler, though at higher working-fluid pressure. The design is currently under review with component suppliers. An all-stainless steel microtube design is proposed, offering low gas-side restriction and high heat transfer performance per volume, as well as corrosion resistance to exhaust condensate. As with several other components, the design is sized specifically for current PACCAR SuperTruck II specifications. An exhaust bypass valve will be supplied as part of the WHR system and have been procured.

Recuperator

The preliminary design for the recuperator is shown in the figure below. The core sizing has been finalized based on WHR system specifications and uses a brazed aluminum bar-plate construction. Header design work is in-progress with minor revisions pending to accommodate updated gas-side connections and integrated oil-management features. Mounting for the recuperator is expected to be off-engine, either to the WHR gearbox/dyno stand or condenser stand, as the WHR system benefits from close-coupling of the turbine, recuperator, and condenser. The working-fluid passes between these components in a low-density vapor state making reduction of plumbing lengths an important factor in optimizing turbine performance. Procurement of the recuperator assembly has been started.

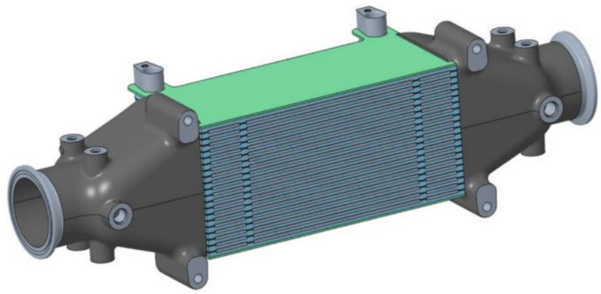


Figure 69: WHR recuperator assembly

Condenser

A condenser is required for heat rejection from the WHR system. The recuperator vapor-side outlet feeds into the condenser inlet, where the working-fluid is further cooled, undergoes complete condensation, and some level of liquid subcooling. In a conventional on-highway application, the WHR condenser is often air-cooled and located in the vehicle front-end cooling module, similar in concept to an HVAC condenser, though typically larger in size. For the current test-cell demonstration a water-cooled condenser will be employed. This allows the flexibility to simulate a wide range of vehicle condenser sizes and reduces the cost and complexity of the test cell hardware. Based on the WHR system specifications and test cell chilled water interfaces, a stainless steel brazed-plate heat exchanger has been procured for use as the condenser.

Simulated system performance

A testbed sensitivity study was performed around the nominal working conditions (based on PACCAR SuperTruck II engine simulation results. The results from this study are summarized in the radar plot shown in figure 70. likely changes to the engine operating conditions (higher exhaust temperatures, higher charge temperatures, lower overall engine BTE) are expected to benefit WHR performance. The highest identified risks to WHR performance consist of low exhaust temperature, low coolant temperature, elevated engine power (without increased waste heat temperatures).

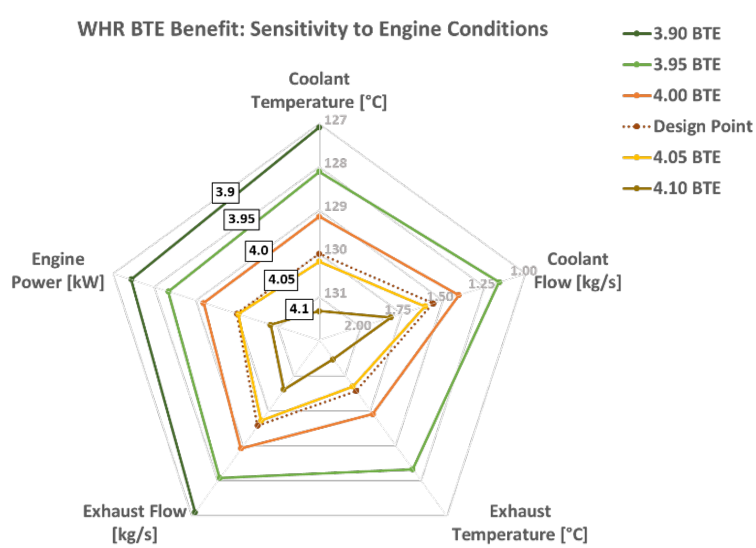


Figure 70: WHR Performance as a function of engine conditions

Component and system level testing

During the WHR system validation testing, a thermal test engine was used to provide sufficient waste heat to simulate the operating conditions of the high-efficiency demonstration engine. Significant effort has been made to ensure that the proper thermal conditions are achieved and that the WHR system will operate in the same manner and with the same performance as would be achieved if coupled directly to the high-efficiency demonstration engine. Connections were made between the engine heat sources and WHR system heat exchangers. Controls and data acquisition systems, including all WHR instrumentation and actuators were installed and functional checks completed. Details of the installation and assembly of major sub-systems are provided for the turbine expander and gearbox, main WHR test stand, charge cooler assembly, coolant boiler, and tailpipe boiler. The fully assembled WHR system is shown in Figure 71.

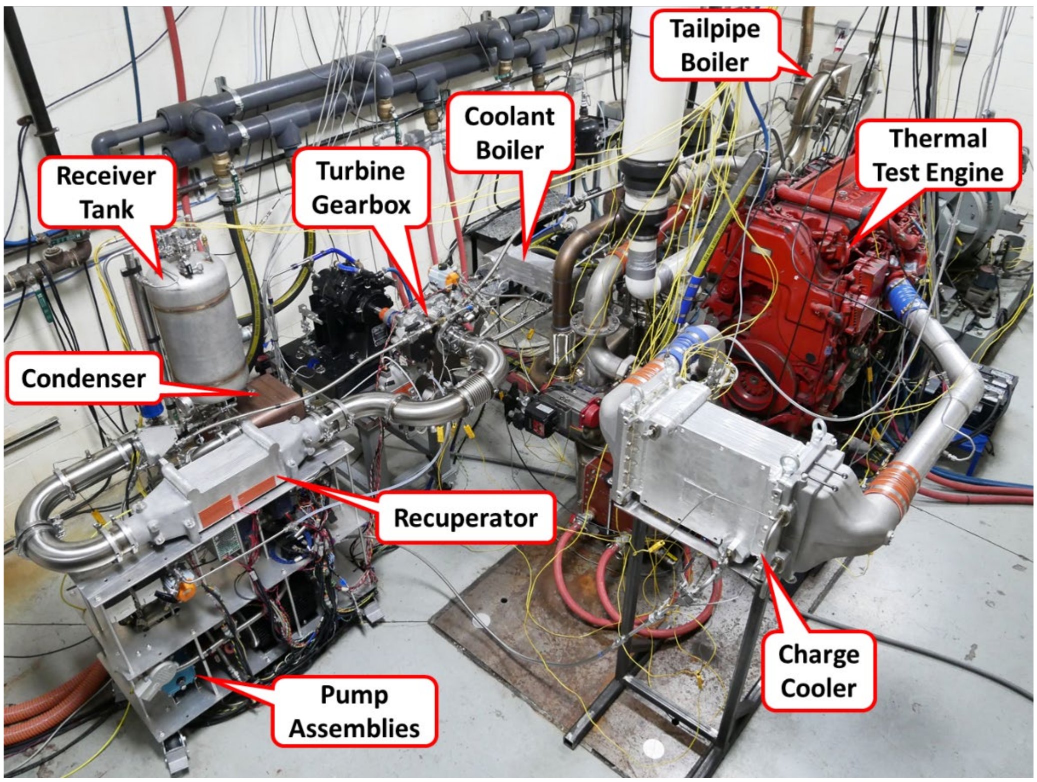


Figure 71: WHR Demonstration system in test cell

The first dual entry turbine testing results, the key novelty developed under this program, were promising with a measured power increase of >1kW versus the simulated power. (17.4 versus 16.2kW). The oil management was acceptable at design conditions, and no design issues were observed with turbine, gearbox, and dyno assembly. The condenser, fluid inventory control, and fluid pumps were all successfully validated and no significant hardware revisions necessary to proceed with system level. Overall, it was concluded that no significant hardware revisions were necessary going from the design to final validation phase in the program to complete final system level testing.

PACCAR provided multiple operation modes to Cummins based on their single cylinder engine testing activities with SuperTruck II hardware to use in evaluating the WHR system performance. These modes included the design point and six alternate operating points (Cases 1-6). To provide an updated estimate of the WHR performance, the waste heat conditions for the latest engine operating point considered as Case 7, shown in table 8.

Cases	Exhaust Temp	Exhaust Flow	Charge Flow	LP CAC Gas Temp	HP CAC Gas Temp	Coolant Temp	Coolant Flow	Coolant Heat	Comments
	°C	kg/s	kg/s	°C	°C	°C	kg/s	kW	
Demo Point	207.8	0.3458	0.3373	90.6	81.2	130	1.56	56.3	Design conditions
Case 6 Test	268.9	0.2832	0.2699	77.9	82.1	130	1.58	53.1	Single cylinder test data
Case 7 (Current)	279.7	0.2862	0.2778	79.2	89.2	130	3.84	51.2	Latest demo. engine data
Deviation (7 vs. 6)	+10.8°	+1.06%	+2.93%	+1.3°	+7.1°	-	+143.0%	-3.58%	

Table 8: Demo Point and Alternate Engine Operating Points (Cases 6 and 7)

During the test cell demonstration of the WHR system, extensive data was collected at the design point and alternate operating conditions (Cases 1-6). The latest waste heat conditions from engine testing (Case 7) are shown above as well. Apart from the increased coolant flow rate, the waste heat conditions used for testing are very similar. Elevated coolant flow rates in this range were tested during WHR system sensitivity studies, as reported earlier. From table X, it can be concluded that the coolant heat availability, which has decreased modestly (3.58%), all other waste heat conditions have improved or are unchanged. Additionally, given the similarity of the measured engine operating conditions compared to the measured WHR input operating conditions used to build the models, the existing simulations provide accurate (and verified) results.

With significant additional data available from the test cell demonstration of the WHR system at the design point, improved component calibrations have been implemented in the WHR simulation tools. Specifically, performance of the system heat exchangers has been revised, gearbox and turbine correlations were updated, and more accurate limits for the subcooling and pump performance were implemented.

A comparison of WHR test and simulation results are shown in Figure 6. Test results were reported previously for the optimized design point and Case 6 and have been shown for comparison to simulation results for those conditions. The optimized design point test result was measured at 15.54 kW net power from the system. The calibrated simulation result matches this well at 15.58 kW. Comparing the test result and simulation result for Case 6 (the test point most closely matching the current engine waste heat conditions) respectively, values of 16.65 kW and 16.70 kW were obtained, matching to within 0.3% which is in line with the test-bed measurement accuracy.

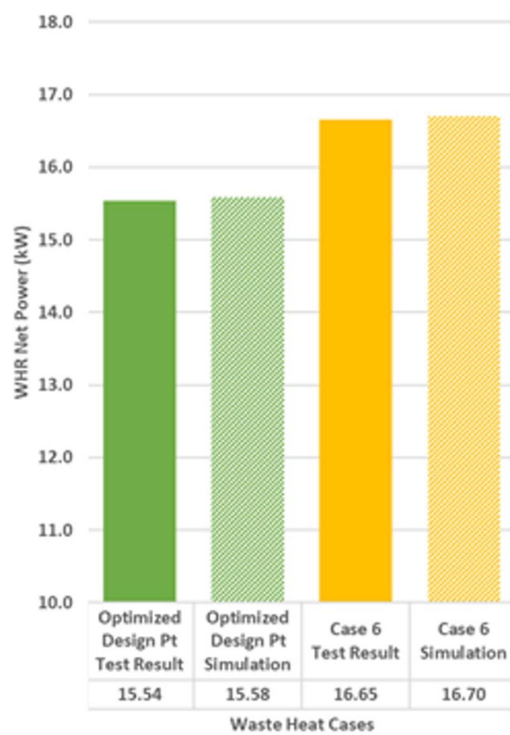


Figure 72: Comparison of WHR test and simulation results

This modest increase of 410 W, or 2.5% in net power, from Case 6 to Case 7 is primarily due to the increase in exhaust temperatures (+10.8°C), which improves the amount of heat recovery and turbine inlet pressure in the WHR high pressure loop. Only a modest improvement in performance is attributable to the high coolant flow rate, which does result in a decrease in temperature difference between coolant at WHR working fluid. The optimization of the design point resulted in an increase of 0.70 kW, or 4.7% net WHR power. This optimization was conducted using measured testbed results around the WHR design point only. As such, additional WHR hardware optimization benefits based on the latest engine waste heat output conditions, has not been considered in this optimization study. Based on this, the results provided for Case 7 are expected to be conservatively low.

With the test cell demonstration of the WHR system having been concluded and documented in previous test campaigns, work continued focusing on the revision and calibration of the WHR simulation tools to provide the most accurate WHR results as the final PACCAR SuperTruck II engine testing continued in parallel. While directly implementing the actual measured heat rejection values from the PACCAR MX Supertruck II engine, it was concluded the WHR was operating under more favorable conditions as compared to the original design point. The final WHR system net power measured 17.1 kW. Given engine brake power of 185.0 kW, the WHR BTE contribution was 4.7% which was significantly above the original design target of 4.0% BTE.

In parallel to the WHR testing, the final optimization of the PACCAR SuperTruck II engine took place. During the engine build the following hardware was selected and installed from previous POC engine testing campaigns:

- Camshaft with 70 deg LIVC (i.e. Miller timing)

- Injectors with 3.4 L/min flow, angle of injection at 142 degrees and 7 injection holes
- Piston with the T1 design

As the engine build neared completion the fabrication of hardware needed in the test cell initiated. This included: the charge air coolers (one for the low-pressure turbocharger and a second for the high-pressure turbocharger), the stands and the piping to connect them to the engine and test cell. The flywheel adapter to dynamometer and encoder to monitor precise engine speed. The connections for liquids: fuel supply and return, coolant inlet and outlet, and all the chilled and process cooling connections.

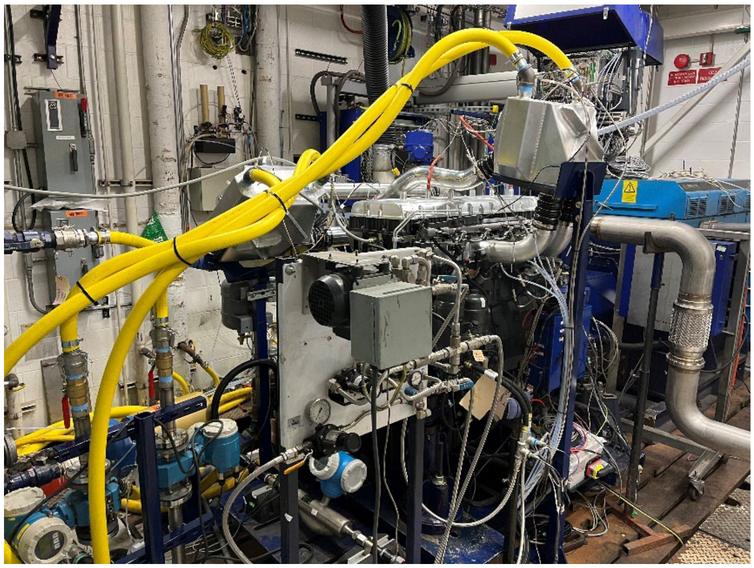


Figure 73: PACCAR MX SuperTruck II Engine in the Test Cell

With the engine installation complete and the engine functional check successful, the engine break-in initiated. With the engine break-in successful the engine restrictions, inlet and exhaust conditions were set to:

Parameter	Value	Unit
Intake Temperature	14-18	°C
Intake Restriction	-1.4	kPa
CAC Outlet Temperature	17.8	°C
Exhaust Restriction (to mimic EAS)	4	kPa
Relative Humidity	50	%

Table 9: SuperTruck II engine test-cell settings

Data collected at the 65-mph road load point at various engine settings was analyzed to determine the optimal settings to obtain the greatest BTE. For the engine speed, it was found that the optimal speed was 1075 rpm:

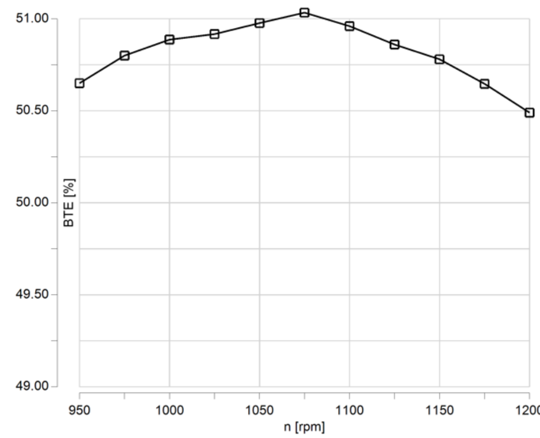


Figure 74: BTE / engine speed trade-off

The other key optimization variables are the fuel injection pressure and timing. For the optimal BTE the rail pressure is selected at 1200 Bar and 8 degrees advanced of TDC. A typical trade-off for timing and pressure found is shown in figure 75:

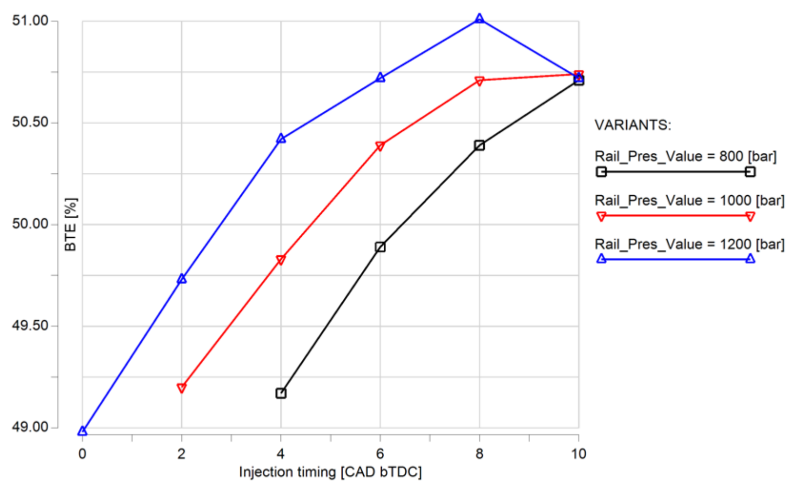


Figure 75: BTE as a function of injection timing and pressure

It is a well know relationship that increased back pressure decreases BTE. For the PACCAR SuperTruck II demonstration engine this tradeoff is plotted in figure 76. Also included in this plot is the effect of coolant temperature on BTE. Despite earlier POC measurements on more standard production engines, it was unclear if increasing the coolant temperature was going to improve BTE by way of reducing heat transfer through the combustion chamber walls or reduce the BTE by way of increased loss of gases at higher temperatures flowing through the ports on the final demonstrator engine. From the data it can be concluded that increasing the coolant temperature to 130 ^{0}C reduced the BTE, thus leading us to believe that the increased loses in the (not insulated) ports is the dominant factor. However, the increased coolant temperature has been selected due to the increased Waste Heat Recovery system performance, hence overall increase in BTE.

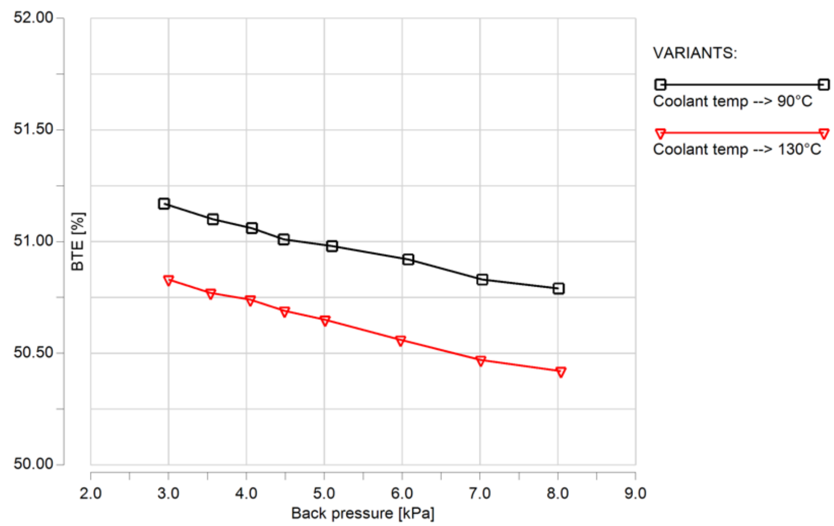


Figure 76: BTE as a function of exhaust back-pressure and engine coolant temperature

The effect of oil pressure and oil temperature on BTE. Decreasing the pressure reduces the load on the engine increasing the available torque to the dynamometer increasing the BTE. In contrast to increasing coolant temperature, as oil temperature increases the BTE of the engine also increases.

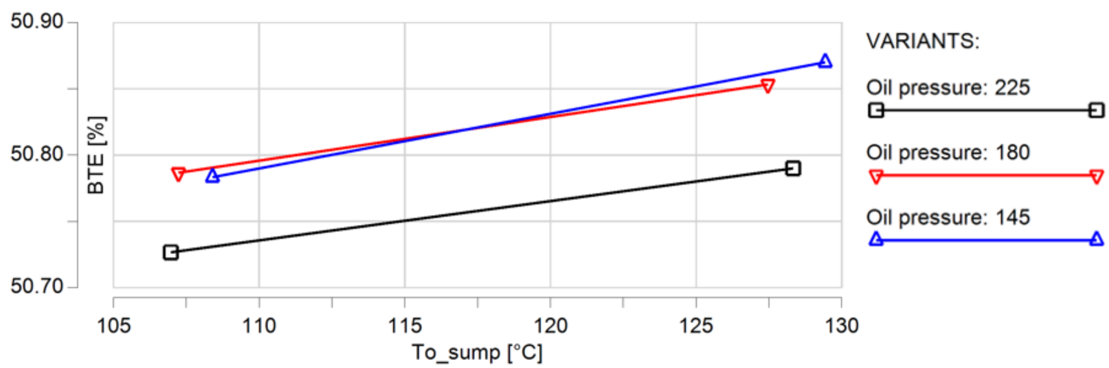


Figure 77: Variation in oil temperature and oil pressure

The calibration/optimization data shown above was analyzed to determine the optimal BTE performance at the 65-mph road load point. Once this point was identified, the engine was run at this point for 15 minutes of continuous operation. During this 15-minute demonstration, an average BTE of 51.0% BTE was measured.

With the optimum BTE point identified the EAS and EGR systems were installed to start work on the emissions calibration of the engine. Once the EAS and EGR systems are installed, calibration of steady state points will initiate. Followed by the calibration of transient conditions. Once initial calibrations are created the emissions test cycles will be attempted. The calibrations will be iterated based on the results of emissions test cycles both in performance and emissions results.

Testing continued to determine the calibration settings for the steady state points. The Supplemental Emissions Test (SET) cycle points were used to develop the injection timing, fuel rail pressure and waste gate maps of the engine. The below plots show the timing and rail pressures determined for the SET cycle points (not including the full load points for A, B, and C speeds). This data will be used to make the steady state engine maps to support completion of the SET test.

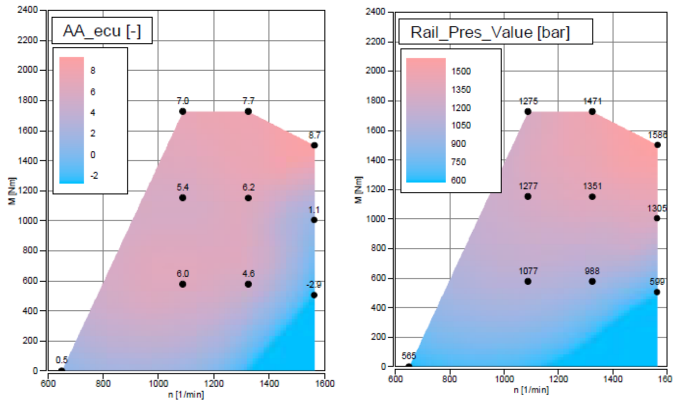


Figure 78: SET Injection timing (AA_ecu) & Rail Pressure settings

The exhaust aftertreatment system was installed to conduct the SET cycle and Federal Test Procedure (FTP) cycle test protocols. The exhaust aftertreatment system contains the following catalysts: diesel oxidation catalyst, diesel particulate filter and selective catalytic reduction and installed in the order listed. The diesel oxidation catalyst is used to reduce any unburnt hydrocarbons in the exhaust stream that escape the combustion chamber. In addition, the diesel oxidation catalyst will also oxidize any carbon monoxide to carbon dioxide. The diesel particulate filter is used to capture any particulate matter generated in the combustion process. The selective catalytic reduction is used to reduce the oxides of nitrogen to diatomic nitrogen and water vapor. After the catalysts the exhaust stream is combined prior to the instrumentation sampling points for the tail pipe. The diesel oxidation catalyst and diesel particulate filter are passive systems in this installation and do not require a control system. The selective catalytic reduction system requires diesel exhaust fluid to reduce emissions. To support the control an off-engine control box was created to manage the diesel exhaust fluid injection system. Below is a picture of the exhaust aftertreatment system installed in the test cell.

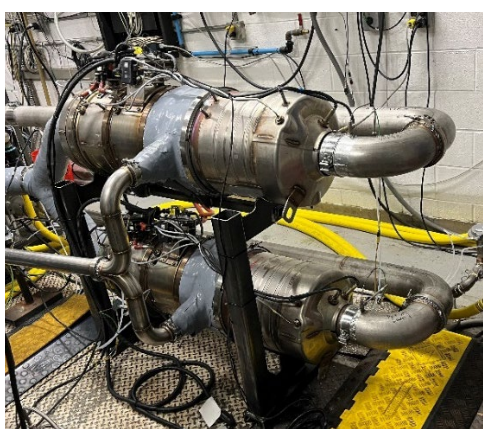


Figure 79: Production exhaust aftertreatment system installed in test cell

To support the exhaust aftertreatment system in mitigating emissions an exhaust gas recirculation (EGR) system was also installed. The exhaust gas recirculation system includes an Eaton EGR pump previously reported on. This system is installed as a low-pressure loop system. The exhaust gas is sourced after the diesel particulate filter and fed back into the combustion air supply at the low-pressure turbocharger compressor inlet. There are two EGR coolers to reduce the exhaust gas temperature. One is located before the EGR pump to reduce the exhaust temperature below the maximum inlet temperature of the pump but not below the condensation temperature of water. The second cooler is located after the EGR pump to reduce the exhaust temperature nearing the inlet air. There is also a valve installed to block any flow through the EGR system. This valve is in place to allow testing with the EGR system installed but does not provide any EGR.

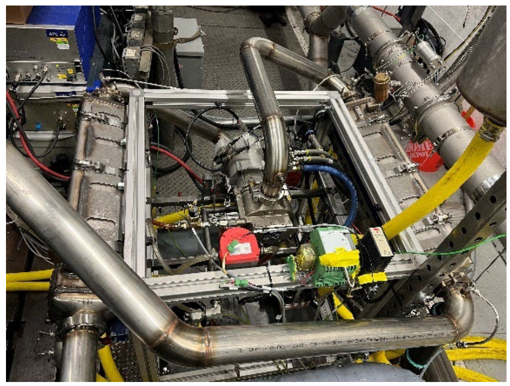


Figure 80: Eaton EGR pump and support systems and two EGR coolers

8. Vehicle Technology Development

Predictive Features

Withing the PACCAR SuperTruck II program, the team explored a combination of predictive technologies such as cruise control, neutral coast, downhill speed control, transmission shifting and engine off coast with powertrain electrification. These advanced predictive features are defined as follows:

- Predictive Cruise: Adjusts the vehicle set speed based on topology information (with the objective of keeping the total trip time neutral)
- Predictive Neutral Coast: Idles the engine with the transmission in neutral, when possible, to save fuel
- Predictive Downhill Speed Control: Manages the vehicle speed while driving downhill in a way to optimize fuel use
- Predictive Shifting: Maximizes drivability by reducing the number of gear shifts
- Predictive Engine Off Coast: Shuts off the engine with the transmission in neutral, when possible, to save fuel

Figure 81 illustrates an application of the predictive driving assist technologies investigated. By looking at the grade ahead, predictive shifting will makeshift decisions, like an expert driver would; taking into account the exact road grade, vehicle mass and available torque, it will avoid unnecessary shifts improving performance, drivability and fuel economy. Recovery of downhill, or braking, energy contributes to the overall freight efficiency improvement over the drive cycle. Kinetic energy recovery allows the truck to increase its speed while going down hills, turning downhill potential energy into kinetic energy that can subsequently be used to coast or reducing power required for the next hill climb. 1 kWh of energy corresponds to about 30 feet of elevation, or 10 mph vehicle speed change.

From the investigated predictive features, all were implemented in the final vehicle demonstrator except engine off coasting. Although engine off coasting contributes to fuel savings, PACCAR learned that the technology has low drivers'/ costumer acceptance due to safety concerns (despite of the electrification of critical auxiliaries such as power steering).

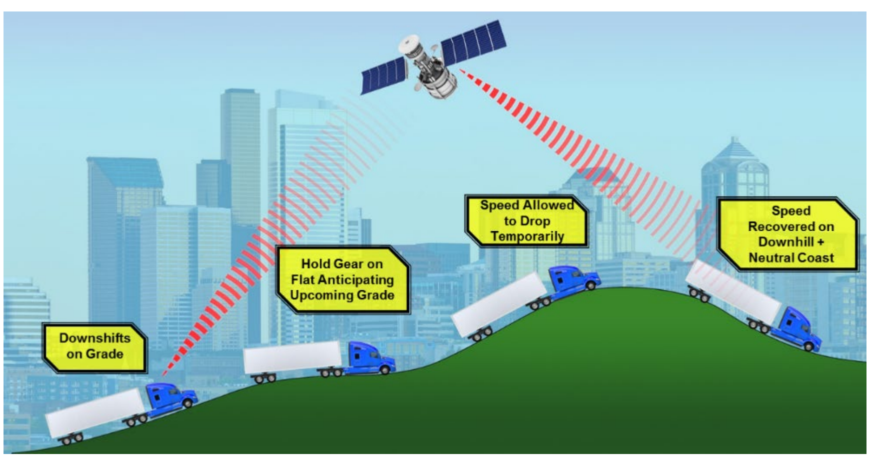


Figure 81: Suite of Predictive Driving Assist Technologies

Mild Hybrid Powertrain Development

The mild-hybrid concept was selected based on an independent, collaborative Eaton/PACCAR development project called the electrically Regenerative Accessory Drive (eRAD). The eRAD system uses a transmission-mounted, two-speed gearbox and an electric motor with electronic controls to either mechanically or electrically power traditionally belt-driven accessory loads. The unit is connected to the Power Take-Off (PTO) of the transmission. Generating at 48 volts allows for increased power capacity and throughput with reduced conductor size, and an integrated DC to DC converter provides power for existing 12-volt systems. The proposed SuperTruck II mild hybrid system is shown in Figure X. In this architecture Front-mounted, belt-driven engine accessories have been removed and electrified – including the engine cooling fan.

A mild hybrid system with the capability to run 30kW regeneration power for one-minute was selected based on hybrid powertrain architecture evaluations. The limits on the regeneration timing were considered acceptable after reviewing route simulations which showed the majority of the regeneration events having less than a one-minute duration. The team concluded that the lost regeneration energy that occurs on very long downhill descents (greater than one minute in duration) was not worth the extra cost of the upsized hybrid system.

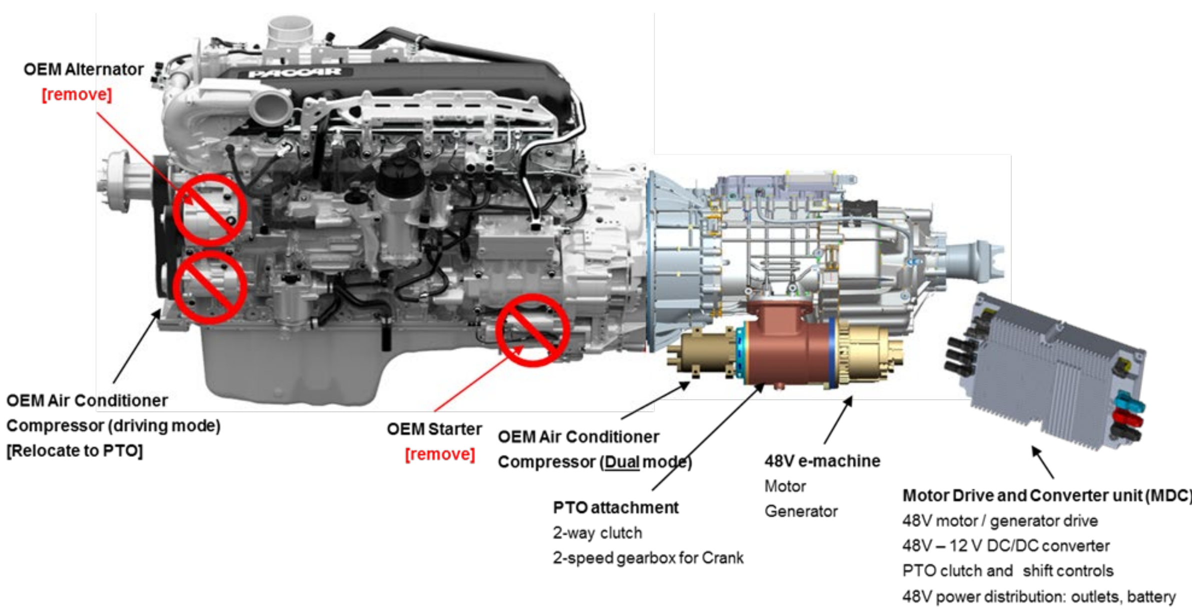


Figure 82 - Proposed Hybrid Architecture, Electric Regenerative Accessory Drive (eRAD)

The definition of vehicle states has been defined by an analysis of different scenarios and use cases. This ensures that the control system is robust to all conditions that the vehicle will encounter, whether they are part or SuperTruck II objectives (such as freight fuel efficiency assessment) or fall outside of the objectives (such as driveability and low speed maneuverability). The focus of development is on systems that will influence the objectives and additional testing will be performed to ensure robustness of the controls under real-world conditions.

The eRAD system has the capability of transmitting power through the transmission and PTO gearbox to facilitate the following functions:

1. CRUISE – During Cruise mode, the engine is powering the vehicle and electrical motor/generator as it would with conventional architectures, although the power path is through the transmission. In the SuperTruck II project, motor/generator power draw from the engine is modulated by a predictive cruise control and the energy management algorithm based on expected regeneration opportunities, electrical load, and battery state-of-charge (SOC). Electrically driven accessories are powered by the batteries or the motor/generator via the electrical bus.

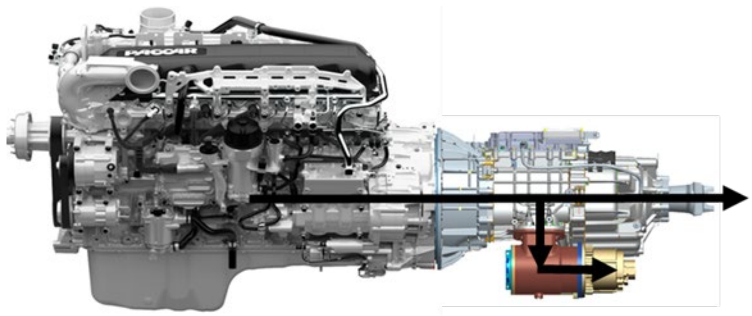


Figure 83: eRAD in "Cruise" mode

2. COAST – During Coast mode, the wheels (vehicle kinetic energy) are driving the motor/generator through the transmission; allowing the engine to be turned-off or idled to save fuel. Electrically driven accessories draw their energy from the motor/generator. It is during the Coast mode of operation that regeneration via electrical battery storage occurs, increasing the battery SOC to be used later when in Cruise mode. This kinetic energy is typically lost as heat during braking, but is captured as regenerative energy in the eRAD concept.

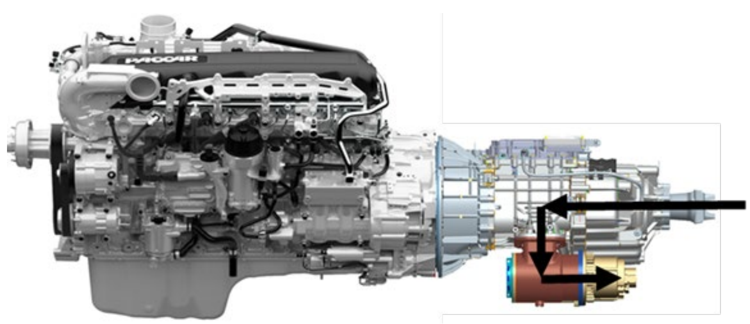


Figure 84: eRAD in "Coast" mode

3. CRANK – During crank mode, the gearbox shifts into a deep gear reduction with the transmission in neutral, and the motor/generator is powered off the 48V batteries to crank up the engine to starting speed. Following the starting of the engine, the gearbox is returned to normal gear reduction and the eRAD system can recharge batteries through the motor/generator when needed. The elimination of the flywheel-based starter reduces weight and cost of the system. The 48V system also provides a powerful and robust starter that enables Stop/Start. The presence of a clutch between the starter and the engine helps reduce starter motor wear.

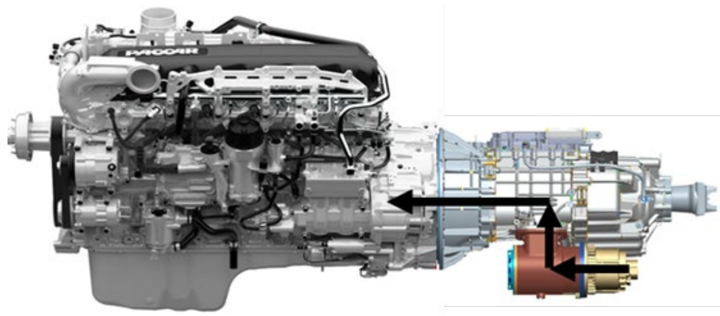


Figure 85: eRAD in “Crank” mode

For first level simulations, constant electrical loads were used in analyses. Now, detailed examination of electrical loads are underway by Kenworth and PTC engineers to drive energy storage and system power requirements. Typical electrical loads during normal driving conditions are well known. As the SuperTruck II vehicle will be electrifying more accessories than normal, additional electrical loads are being examined – including hotel-specific loads. Typical power consumption from the electrical system during cruise is estimated to be around 10kW with electrically driven cooling fans, power steering, and cooling pump(s). Peak power is strongly affected by the electrical power steering loads and the high battery charging load immediately after hoteling all night. This is expected to load the electrical machine to its full continuous rating and may still draw transient power from batteries for short timeframes.

E-motor

The major considerations for the selected e-motor have been the torque available for cranking the engine, the efficiency of the motor while driving the HVAC compressor during a hoteling event, peak voltage generated during regeneneration, base speed, and physical size. The motor will be liquid cooled with a cooling circuit separate from the engine coolant, must be robust to the environment of the application, and must be available for characterization within the constraints of the SuperTruck II schedule.

The primary requirements of the motor/generator are high power regeneration (30kW power extracted from coasting vehicle kinetic energy), and the engine cranking function to start the engine (1000Nm for cold crank torque at the flywheel). The motor/generator suppliers under consideration have provided torque and efficiency predictions for their machines, an example of which is shown in Figure 86.

Cruise				Coast			
Gear	Engine [RPM]	Speed [MPH]	Motor [RPM]	Gear	Speed [MPH]	Engine [RPM]	Motor [RPM]
11	1000-1250	43-54	2250-2812	11	70-80	1628-1860	3663-4186
12	1000-1250	56-70	3079-3848	12	70-80	1246-1424	3837-4386

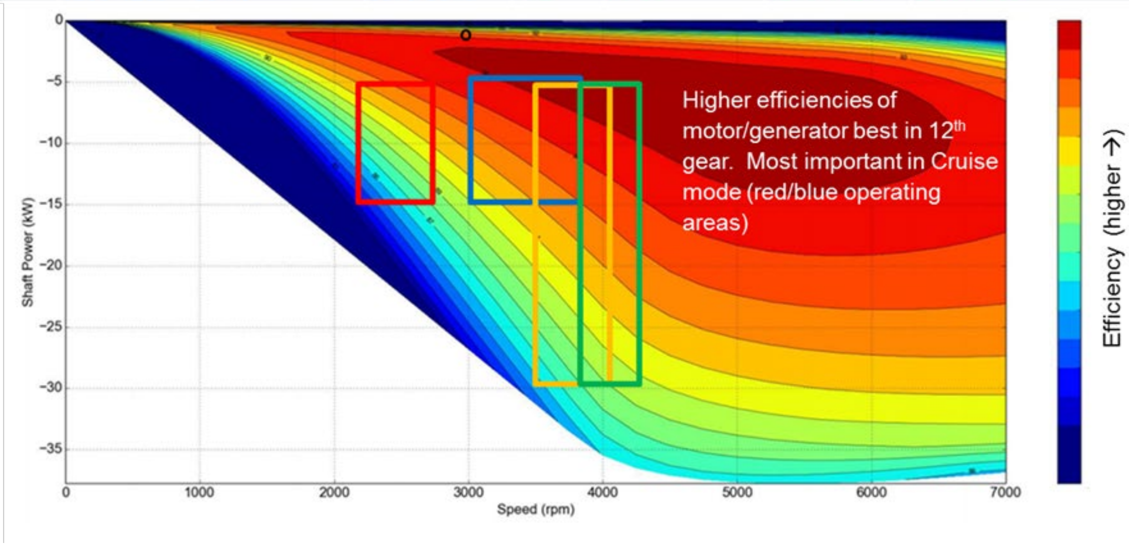


Figure 86 Predicted Generator Efficiency in Top Two Gears, Cruise and Coast Modes

Gearbox

One interesting result of the Power Take Off (PTO) gearbox design analysis is the quantification of how the choice of transmission gear impacts the efficiency of the generators. During a downhill coast, the efficiency during intermittent electrical regeneration is not nearly as important as when the generator will be operating in continuous cruise mode (alternator mode). Cruise mode might be needed, for instance, after an overnight hoteling battery drain with the HVAC system, where the battery state of charge needs to be replenished up to a minimum level. During such situations, the generator may be operating continuously for some time and keeping the motor in its most efficient operating speed range is important. Due to the design of the PACCAR 12 speed AMT, operation in overdrive versus direct gearing for the powertrain results in more efficient charging during Cruise (alternator) mode.

Other analyses were completed looking at the cold engine friction predictions on the PACCAR MX-11 engine and overlaying those on the torque capability of the electric machines when cranking the engine, as shown in Figure 87. Both prototypes (two suppliers) will meet the 1000Nm torque at stall and should be able to spin the engine well above the minimum start speed. This results in a smooth start but will require good fuel control by the engine controller to minimize engine speed excursions above idle when first starting.

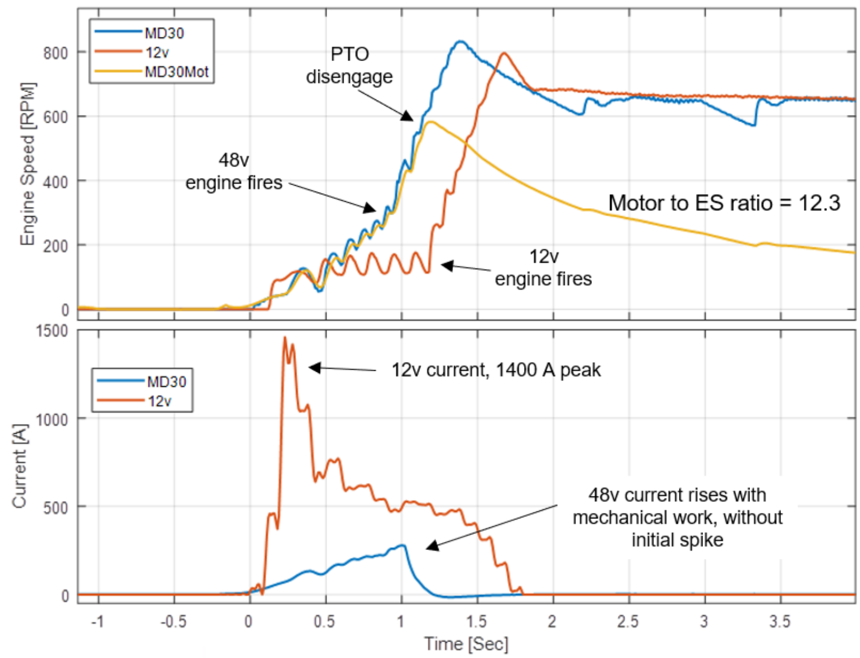


Figure 87: Engine cranking example: conventional 12V versus MD30

The PTO will have two speeds, actuated by the same actuator used in the commercial eRAD PTO prototype. One gear ratio is used for normal driving and regeneration, and a deeper reduction gear ratio will be used for increasing the torque available for cranking the engine. The end of the PTO previously used for interfacing to the air conditioning compressor will not be used because the SuperTruck II demonstration vehicle will have an electric compressor located in the vehicle cab. Efficiency of the PTO has been improved through the elimination of one seal and reduction of churning losses within the PTO. The total mechanical losses of the PTO are a fraction of those of the transmission, as predicted by Eaton models of gearbox losses, as shown in Figure 88.

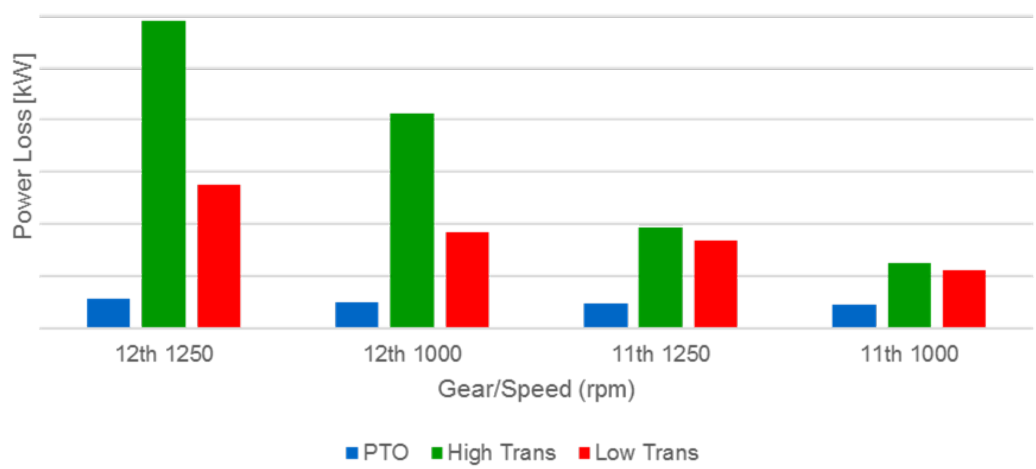


Figure 88: Eaton Model Predictions for PTO and Transmission Losses

The motor drive converter (MDC) architecture has been reevaluated from the original eRAD prototype and has new microprocessors including an ultra-low quiescent supervisory controller that manages the multiple internal and external power supplies within the controller and manages the low power communication to the lead acid battery sensors for management of those components. A diagram of this architecture is shown in Figure 89. The motor drive inverter is sized to handle the larger motor/generator power, and the integrated 12-48VDC bidirectional voltage converter will be based on prior proven designs.

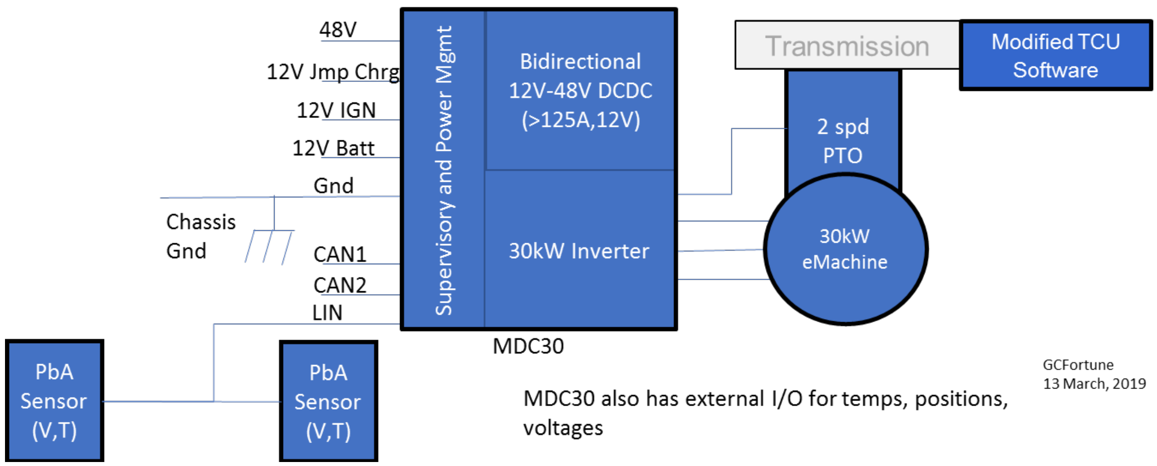


Figure 89: SuperTruck II 30 kW PTO Mounted Mild Hybrid Architecture

Dynamometer Test-rig

A dedicated 30kW motor-generator-inverter test dynamometer was built at Eaton to test the higher power levels of the SuperTruck II e-machine and inverter with the 48V Li-Ion battery pack integrated. E-motor data was collected to facilitate motor mapping and generation of torque loss and efficiency maps (Figure 91), which confirmed the earlier received performance data from the supplier.

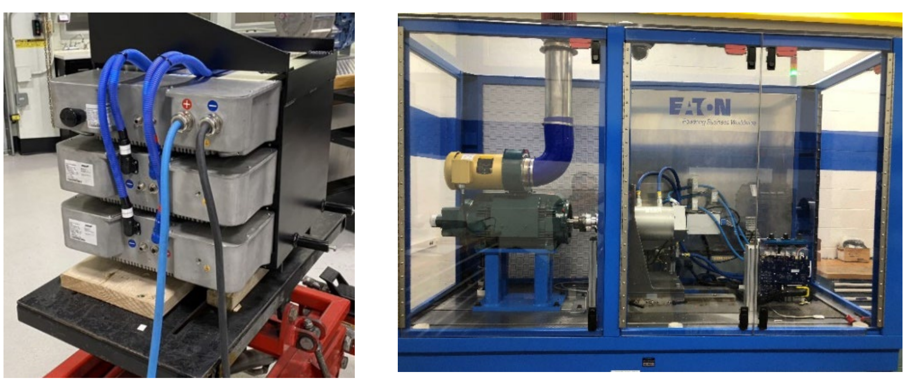


Figure 90: Battery system assembled into truck battery tray (left), and Integrated into Dyno (right)

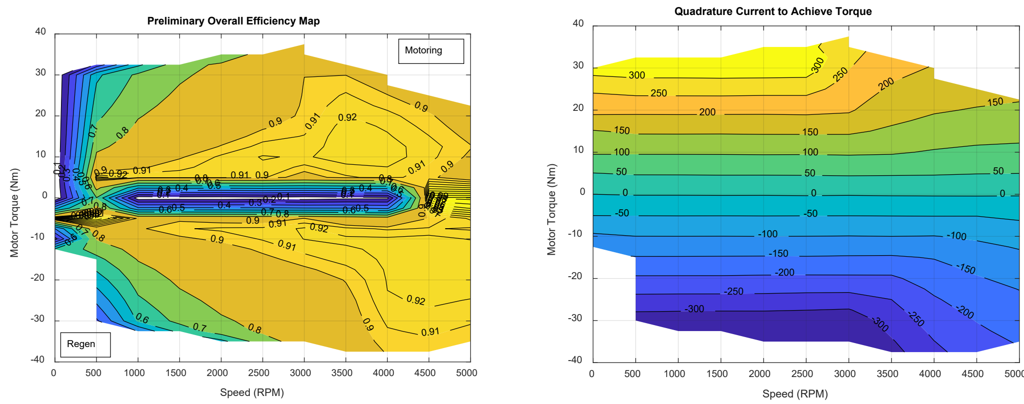


Figure 91: Measured e-motor efficiency (left) and torque loss maps (right) from the mild-hybrid test dyno

Figure 92: PTO Gearbox Test Fit on PACCAR 12 Speed Transmission

48-12V DC to DC Converter

Air-cooled, 1.5kW (3kW peak) 48-12Volt DCDC converter prototypes were developed for SuperTruck II use. Design items addressed included: enclosure design, and circuit development. Requirements were captured at the vehicle, system (powertrain) and sub-system (controller) levels. This enabled visibility of control iterations as the control strategy develops, including increasing complexity of testing from static testing through, Hardware in the Loop environment, Mule, and final demonstration on the SuperTruck II vehicle respectively.

The next generation DC/DC converter design is a bi-directional, air cooled, 1.5kW (3kW peak) 48V-12V DC/DC converter and has a tested efficiency of 95% at rated power and temperature. Thermal measurements on first prototypes indicated that the air-cooling capacity is less than what is required for commercial success (horizontal mounting, convection only), so the team worked on improving the DC/DC thermal dissipation ability in addition for the final hardware versions.

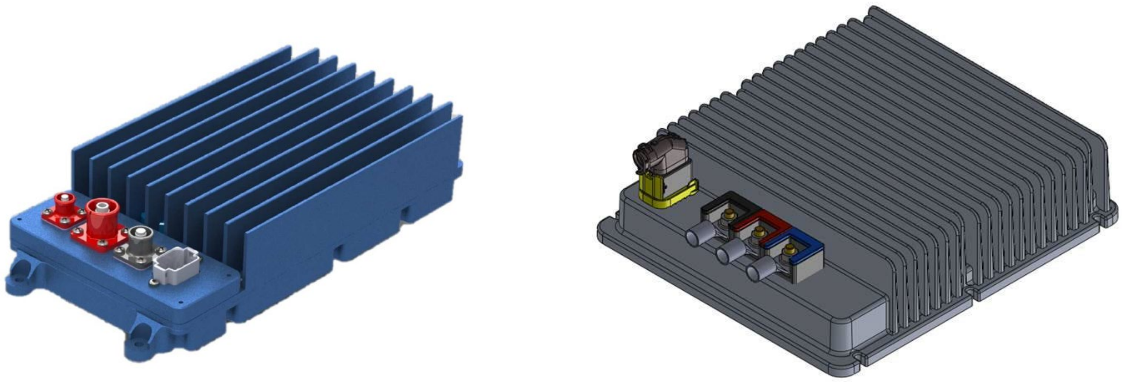


Figure 93: 48-12V DCDC Converter Designs. VTI Prototype (left), used for earlier development work and the final selected program prototype hardware (right).

Motor Inverter with PTO shifting control

Inverter prototypes include assembled printed circuit boards, cold plates, custom connectors, current carrying assemblies, and enclosure parts.

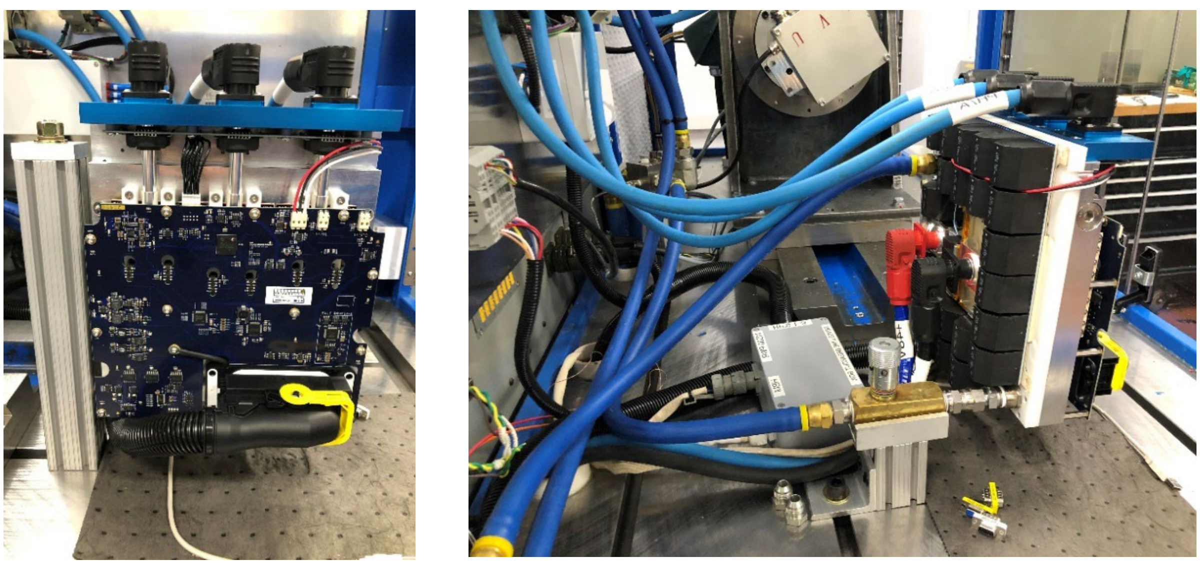


Figure 94: 30kW Inverter mounted on cold plate in dyno fixture (less enclosure)

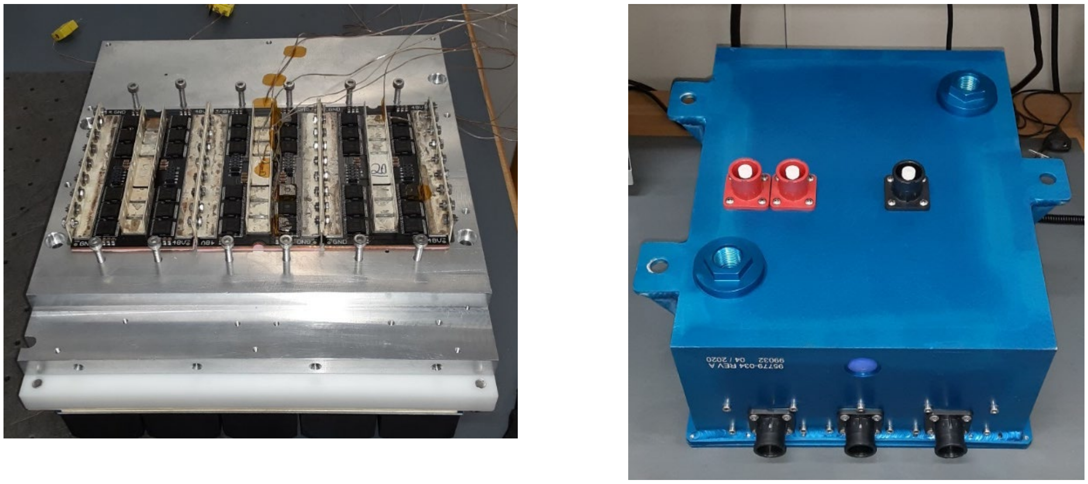


Figure 95: (Left) Power board instrumented with thermocouples to assess cold plate performance. (Right) 30kW Motor Drive Inverter Assembly

Throughout the program the design of the 30kW inverter continued with more detailed work on the enclosure, the cooling methods, and the electronics circuit design. A mockup of the primary components of the power flow conduits and thermal flux paths was created. Besides challenges encountered of running up to 1000A of phase current within the 30kW inverter, managing the temperature of critical components was also significant for the 48V 30kW inverter for the SuperTruck II program. The flow and thermal performance of the semiconductor power devices where modeled when mounted to a cold plate. Results suggest that for both continuous and peak load situations, the power semiconductors were running at least 25˚C below their maximum allowable temperature. Modeling experiments allowed simplification the cold plate design without impacting relative thermal performance. Following thermal characterization on the 30kW dynamometer, temperature and power data was collected to validate the simulations.

Figure 96: Mechanical Mockup of Inverter Components

Mild Hybrid Powertrain - Vehicle Commissioning

A Kenworth T-680 tractor was delivered to Eaton for upfit with a mild-hybrid powertrain. The "controls truck" was used by the SuperTruck-II team for software and control development/validation as it was mentioned in the Controls development section. Function and controls development of basic charging, regeneration, engine crank, engine off coasting, regen gear selection, and 48V Akasol Li-ion battery (see battery section) integration were all developed on this vehicle, minimizing the validation time of the powertrain on the final demonstrator vehicle. Additionally, Eaton put some vehicle miles on the system to resolve early integration issues and build confidence in its robustness.

Figure 97: Kenworth T680 equipped with a (temporary) 48V Lead-Acid battery system

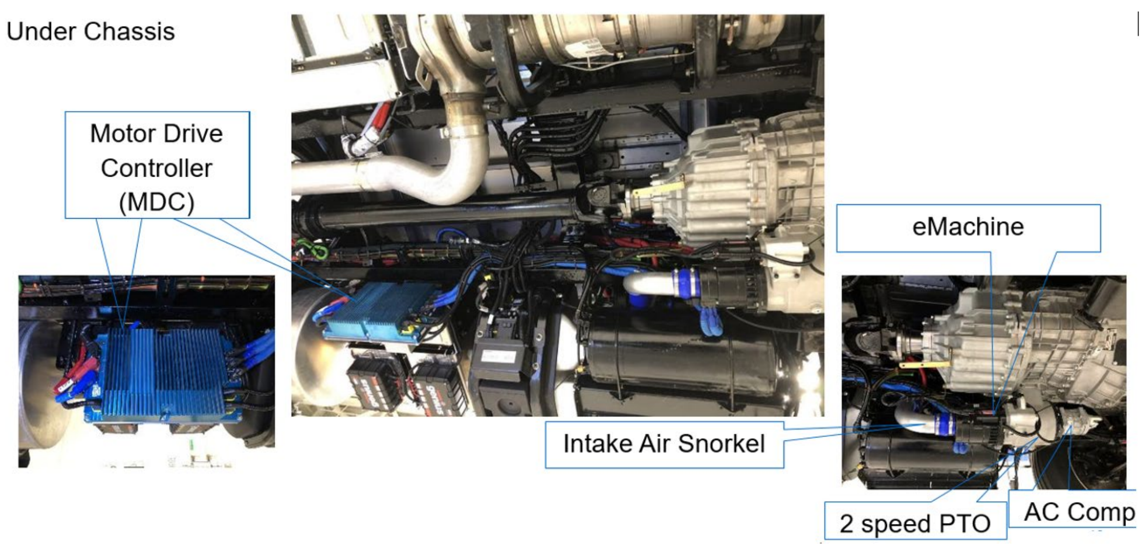


Figure 98: Under chassis view of the Controls truck that was upgraded to a 30kW mild-hybrid SuperTruck II Powertrain

Chassis

The concept development under SuperTruck II can be summarized as follows:

- Mass Centralized
  - Wider Frame Rail Spacing
  - All Major Masses to be Mounted Between the Rails
  - Reduce Force Nodes
- Narrow Front Track
  - Enabling Full Wheel Closeouts for Aerodynamics
  - Require Reverse Splay at the Front
  - Reduced Scrub Radius
- Low Mid-Ship Engine Position
  - Set Forward Front Axle
  - Leading Link Front Air Suspension
- Designed for Modular Assembly
  - Simplify Load Paths
  - Enable Supplier Pre-Assembly
  - Inverted Rails
- Taller Rail Section
  - Improve Structural Rigidity in Bending
  - Larger Footprint for Crossmembers

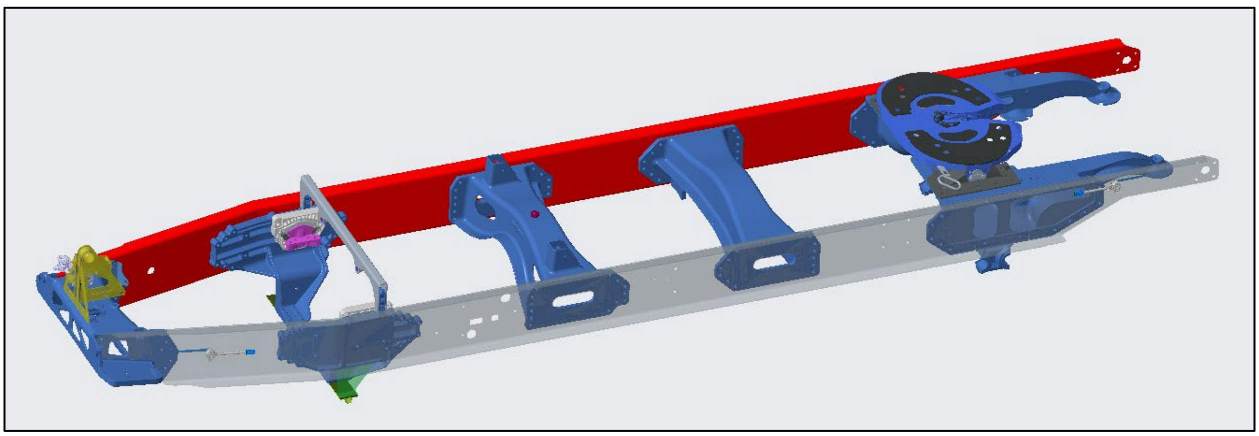


Figure 99: SuperTruck II vehicle chassis

Most modern truck chassis can trace their lineage to vehicles produced a century ago. The historical approach is to treat the chassis as if it was a monument within the architecture that could not be changed. Our team extended our systematic approach to the chassis development efforts, which resulted in a development process that was heavily iterative. Every interaction that could be affected by a change in one area was evaluated against its impacts elsewhere.

Efforts began with evaluating the capabilities of our supplier base to see how much freedom was available. A variety of materials and manufacturing methods were considered. Overall value and

development timeline led us to a traditional material (140ksi Steel) with new frame rail sections and processes.
The optimized frame rail geometry enabled unique chassis structures only previously possible with multi-piece castings and complex joining methods, all of which traditionally added cost and weight. By reducing cantilevered masses, right-sizing fuel capacity and driving loads into centralized points, chassis structures are significantly simplified. A wider frame compared to current product also enables additional packaging space and flexibility for mounting fuel tanks, batteries, suspension cross-members and the fifth wheel, all leading to lower mass.

An updated study of tractor weight and component mass was conducted to refine the SuperTruck center of gravity estimate and resulting axle loading. As an example, the engine position, which was placed entirely behind the front axle, offset the front axle loading otherwise incurred by the cab-over-engine cabin placement. We found that managing our masses while retaining the desired 5^{th} wheel location, relative to the back-of-cab plane, would be challenging. A weight management model was created that would fix the 5^{th} wheel location at the desired distance from the back of cab for optimum aerodynamics and calculate where the rear axle would need to reside to allow us to use a 4x2 configuration at the legal limit of 20k lbs. The resulting wheelbase was 222", still much shorter than current 6x4 T680 tractors given the front axle is closer to the nose and there is no rear-rear axle (wheelbase is measured to the center of the tandem).

With the adjustment to modeled trailer gap, mentioned above, the 5th wheel is now much further ahead of the primary rear suspension crossmember. A new concept was generated, showing promise of handling the required stresses while achieving weight targets. A key feature of this concept is that the suspension loads are cantilevered off the back of the rear crossmember. This feature requires additional mass at said crossmember, however this is significantly less than would otherwise be needed for a tandem axle configuration.

A variety of modeling and analysis processes were utilized, which leveraged advanced simulation techniques. Initially lightweight models were developed and incorporated into the development process for quick concept evaluation and faster iterative loops as the remainder of the vehicle came together. This process analyzes the chassis as a whole, incorporating system-level performance requirements and specific component designs through an iterative feedback loop.

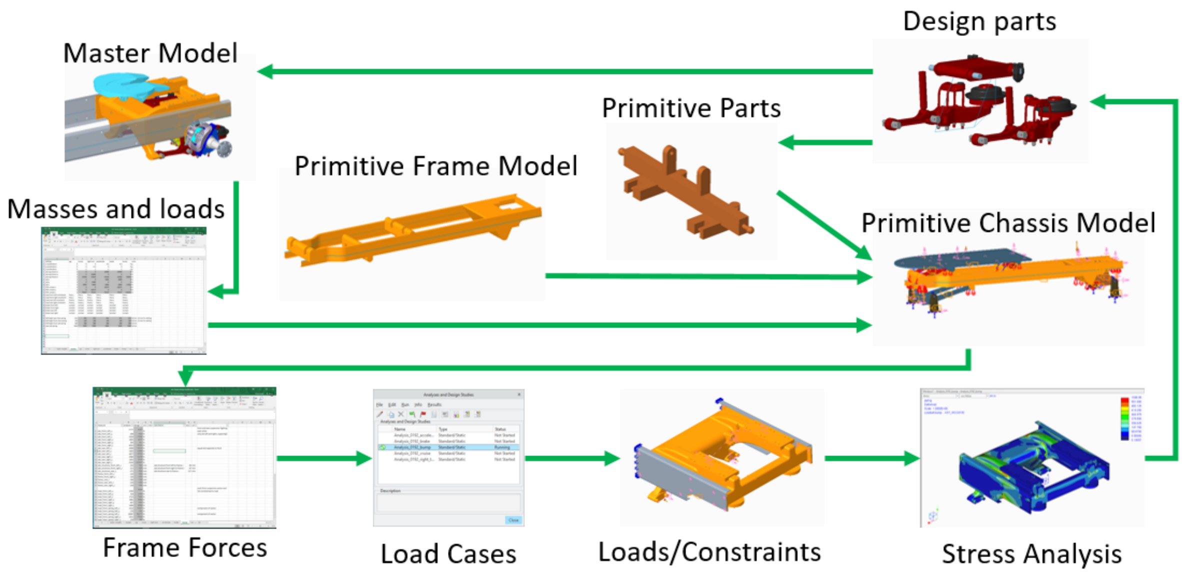


Figure 100 – Iterative Chassis Design Process

From these full-chassis analysis and simulation tools developed in the earlier parts of this program, the team translated system requirements to primary component design requirements. Detailed designs were released, procured, and assembled onto the mule vehicle chassis. The subsequent data that was collected during our validation exercises was fed back into the iterative chassis simulation tools, where further refinement was conducted, now with more detailed models.

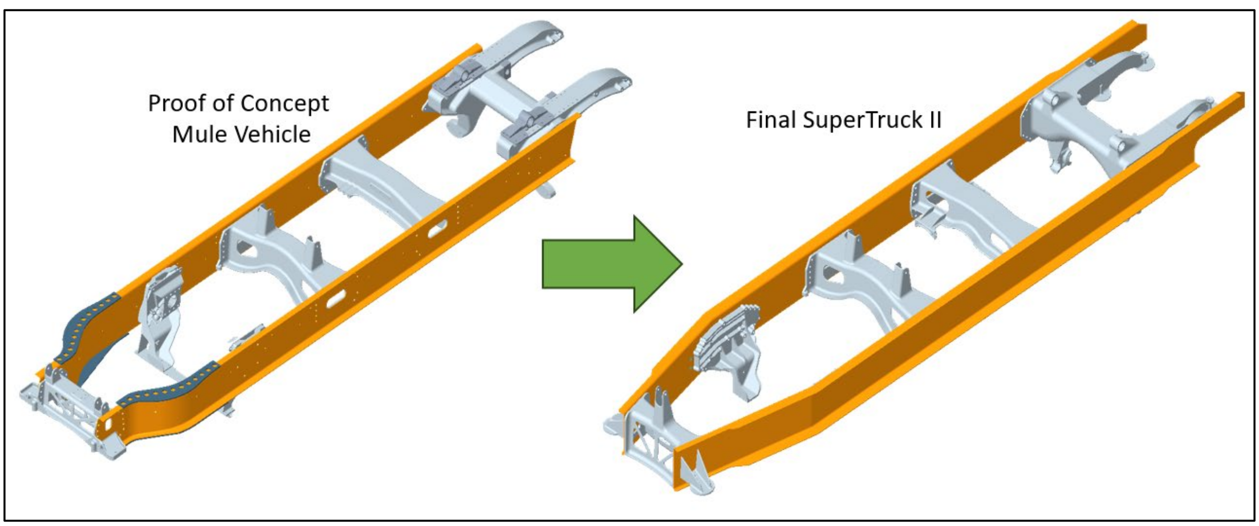


Figure 101 – Comparison of Mule and Final SuperTruck Chassis

Notable changes from the Mule chassis and the Final ST2 chassis include the deletion of the double bend and reinforcements for a single 12° bend that is reinforced by the second crossmember assembly. The remaining crossmembers have been mass optimized and reinforced where needed. The cab suspension supports have been inverted, where we originally had 2 points in the front and one in the rear, we now have two in the rear and one in the front. This change was made to address feedback received from the test drivers that the cab rolled excessively with front suspension articulation.

Front Suspension/Narrow Track/EPS

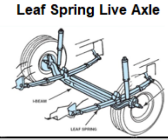
The front suspension of the SuperTruck posed several challenges. The COE layout required a cooling module and engine placement that compromised the packaging of a traditional leaf spring suspension style. A decision matrix was used to weight several attributes of the various front suspension styles against each other to identify the best option for our project. An air sprung leading-link live axle was selected, incorporated into our layout to see how it would best be applied. To reduce the load nodes the radius control arms were extended to the second crossmember. This had the added benefit of reducing the caster change through the suspension travel. It's worth mentioning that one advantage the leading link has over a trailing link is that the caster gain from lowering the vehicle goes in the direction that is most useful with the leading link layout. The net result is that at the lowered ride height, which will be used at highway speeds, there is more caster than at low speeds. This gives you more self-centering and improves lane keeping, reducing driver fatigue. At low speeds the caster is reduced, making the steering effort lighter and reducing the tire wear from cornering.

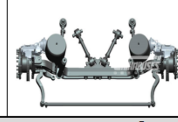
PACCAR Inc

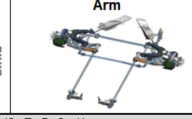
Concept Decision for ST2 Front Suspensions

Kenworth Supertruck - Front Suspensions　**Rev. Date: 05/22/2018**

Participants:　**Leader:** Adrian Murias







Criteria	Weighting (From Pairwise)	Current Leaf Spring Live Axle	Concept 1 - Leading Link Air Sprung Live Axle	Concept 2 - IFS Torsion Sprung - Double A-Arm	Concept 3 - IFS Air Sprung - Double A-Arm
		Score (9, 7, 5, 3, 1)			
Braking (G's)	**0.278**	5	7	7	7
Vehicle Stability & Predictability/Safety Aspect due to Path Accuracy	**0.205**	5	7	9	9
Durability (miles)	**0.132**	5	5	5	5
Weight (lbs)	**0.115**	5	7	3	3
Cost ($)	**0.063**	5	7	1	3
Maintenance Cost / Component Wear ($)	**0.050**	5	7	1	3
Lane Keeping / Bump Steer (°/mm of travel)	**0.043**	5	9	7	7
Tire Wear (miles)	**0.039**	5	5	3	3
NVH (db)	**0.033**	5	7	9	9
Cornering Performance (G's)	**0.022**	5	7	9	9
Packaging	**0.010**	5	7	3	1
Unsprung Weight (lbs)	**0.010**	5	5	9	9
		5	**6.73**	**5.94**	**6.15**

Figure 102 – Comparison of Mule and Final SuperTruck Chassis

One key element of our design is the Narrow Track. Our Aerodynamic studies had highlighted that the front wheel openings were incurring unwanted drag. A variety of technologies were considered for wheel closeouts at the expense of mass and complexity. As we evaluated the steering assist required to meet our existing dry-park steer requirements, we found that a reduction in scrub radius would prove beneficial, particularly when the service brakes were applied. Reducing the scrub radius also reduced the road feedback that is translated back to the steering system and driver, again reducing driver fatigue. Several iterations later we found that when a reduced scrub radius was combined with the correct tire size and amount of track reduction, we could achieve our targeted steer wheel angle (55°) fully within the body work. Thereby not requiring fender closeouts, a reduction in complexity and mass.

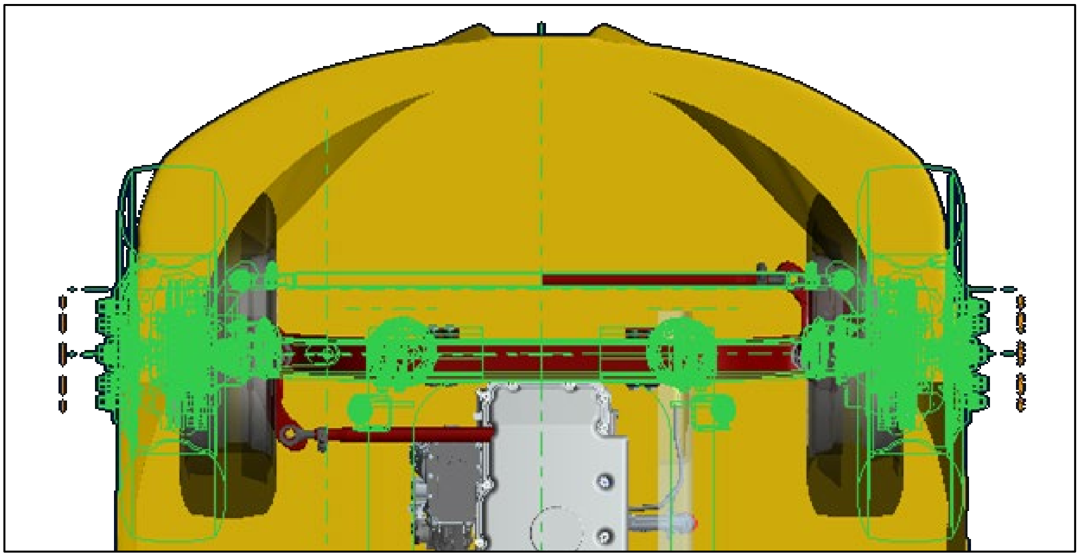


Figure 103 – Comparison Hendrickson Narrow Track on ST2 vs Traditional Class 8 track width

In order to accommodate the center driver position and comply with our packaging requirements a center mounted steering rack was selected over a typical steering gear. The electric power steering assembly is a safety critical and a high-risk component on our truck, so a significant amount of attention has been paid to this system. To ensure we mitigate against these risks our supplier Cascade Drive conducted a variety of subsystem and assembly level tests on both the electrical and mechanical elements of the prototype unit. This unit will help Paccar evaluate this technology and determine production feasibility.

Hendrickson was selected as the supplier for the front axle and suspension for SuperTruck. Their support and expertise helped us meet our goal of 200mm of suspension travel to enable our variable ride height targets. When combined with the set far forward axle location and short overhang the Supertruck II vehicle has an approach angle that exceeds any of our existing aerodynamic trucks.

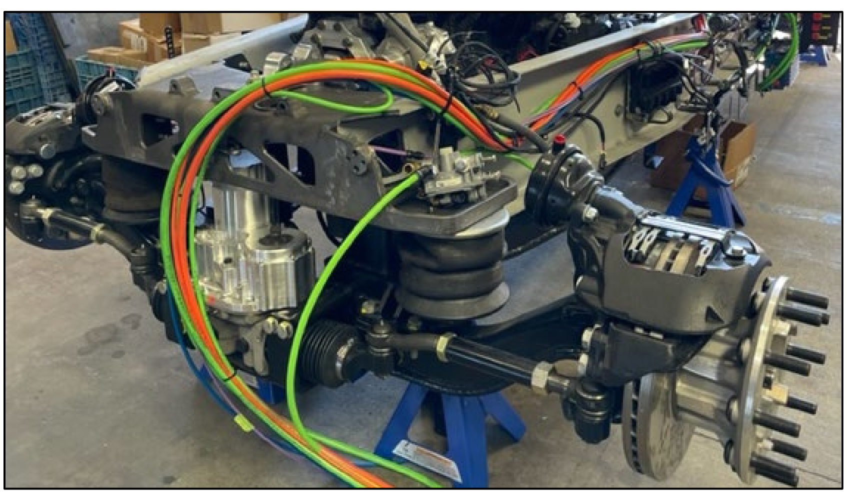


Figure 104 – Cascade TruEPS as fitted on the SuperTruck II vehicle

Rear Axle/Suspension/Driveline

As part of our powertrain optimization effort, the team down-selected six axle ratios for comparison in our full vehicle drive-cycle simulations. These axles were paired with the PACCAR 12-speed transmission in either direct-drive or over-drive configurations. A 2.28 rear axle ratio was selected and subsequently implemented into a new lightweight Meritor rear axle housing that incorporates features for the new Kenworth AG5000 rear suspension.

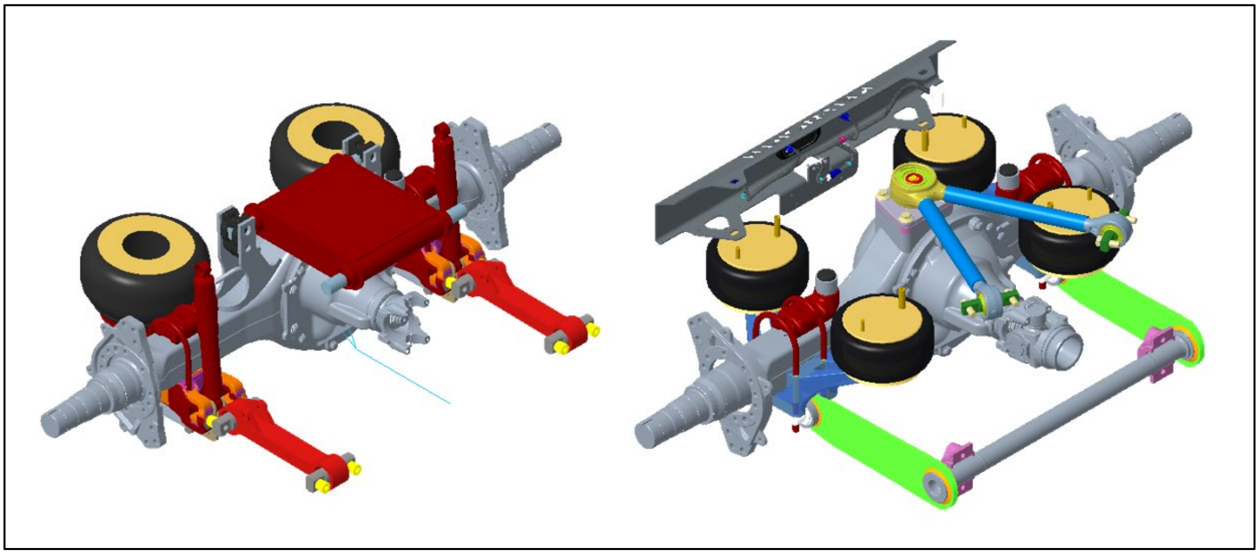


Figure 105 – Comparison of the Hendrison HTB fitted to the Mule vs the Kenworth AG5000 Suspension

The mule vehicle was fitted a Hendrickson HTB rear suspension. This suspension allowed us to validate the effects of raising the roll center in full vehicle dynamics. The results we positive, which led us to develop a new rear suspension based on our AG-Series. This new suspension was 20 lbs lighter, supported 200mm of suspension travel, allowed for a lower ride height, and transferred the loads to the crossmembers more efficiently. These changes allowed us to remove over 50lbs from the rear crossmember due to a reduction in moment arm from the air springs to the 5th wheel in our vehicle.

Meritor's innovative integrated slip-yoke in the differential carrier enabled the use of a 1-piece driveshaft. The length we required would normally require a 2-piece construction to properly address the forces incurred from the harmonic vibrations. Meritor proposed a carbon fiber reinforced driveline for additional weight savings. Interestingly, a carbon fiber driveshaft becomes more cost competitive in applications like ours that would normally use a 2-piece, where the stiffer material can be leveraged to increase the allowable length that can be accommodated in a 1-piece.

Wheels/Tires

Tire selection for the SuperTruck was a subject of multi-variable optimization. All available options were thoroughly evaluated to refine possible sizes for improved rolling resistance, weight reduction, compliance with state and federal laws, and account for the many vehicle packaging impacts. These included available wheel cut, drag area, required wheel width, brake clearance, and possible scrub radius reduction. A compilation of commercially available tire sizes, both in North American and European markets were analyzed and evaluated against the SuperTruck program targets. A balance point has been found that retains vehicle maneuverability by limiting tire diameter, enabling maximum targeted wheel steering angle and resulting turning radius while offering required load ratings. The down-selected offering also enables using the same tire carcass size between the front and rear axles, which offers additional value to vehicle owners. Continental was selected as the supplier and they provided with the tires outlined in the table below (Figure 106). This combination yielded an average CRR of 4.2 with an overall weight savings of over 800 lbs over baseline including aluminum wheels.

Continental Supertruck II Tire Sizes														
Position	Tires per Axle	Number of Axles	# of Tires/	Wheel	Width (mm)	Section	Diameter (mm)	Diameter (in)	Resulting Size	Resulting Lbs/Inch	Projected CRR	Tire Weight (lbs)	Wheel Weight (Lbs)	SUMMARY
Front Axle	2	1	2	22.5	255	70	928.5	36.6	255/70R22.5	597.6	4.58	93	52	CRR = 4.194 Weight = 1698
Drive Axle	4	1	4	22.5	255	70	928.5	36.6	255/70R22.5	498.0	4.14	88	39	
Trailer Axle	2	2	4	19.5	445	45	895.8	35.3	445/45R19.5	485.2	4.09	170	55.001	

Figure 106: Supertruck Final Tire Selection

The wheels were provided by Alcoa. The drive and trailer wheels were commercially available, however the steer wheels were made to Paccar dimensions. These were designed to increase the offset from the fastening plane to the centerline of the wheel. This enabled us to reduce the scrub radius while still maintaining the required brake clearance in order to use disk brakes.

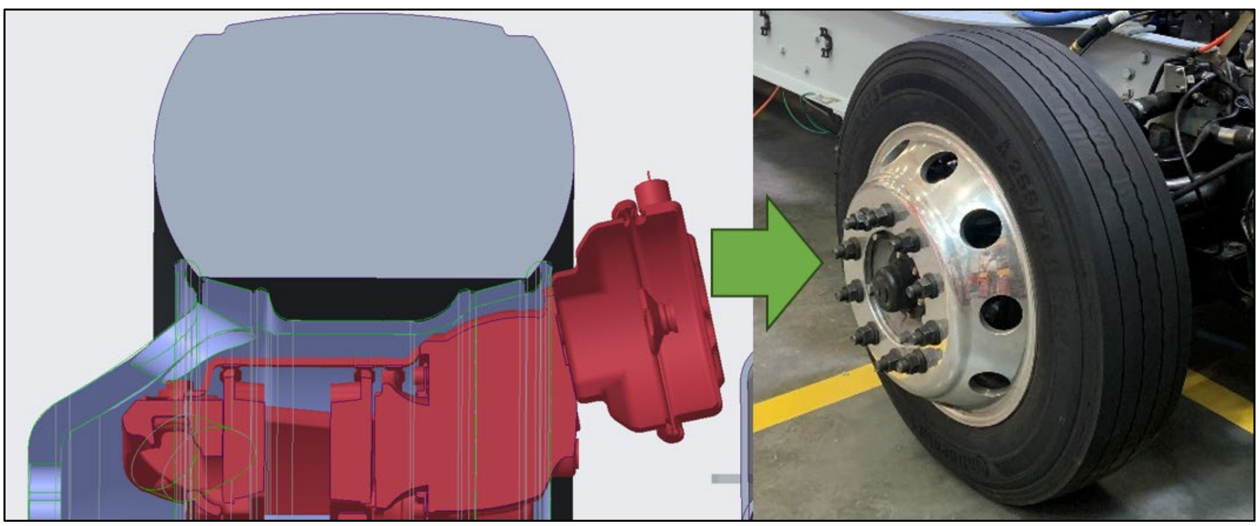


Figure 107: Supertruck High Offset Front Wheels

Battery Development

As part of the mild-hybrid powertrain development activities under the SuperTruck II program, detailed battery pack requirements were developed and compared to feedback from different potential suppliers, shown in Table 9. Both battery requirements and BMS development were guided by the commercial viability of the technologies.

Supplier	1	2	3	4
Module Voltage	44 V	48 V	42 V	48 V
Module Capacity	4.15 kWh	6.1 kWh	2.7 kWh	7.1 kWh
Module Dimension	173mmx125mmx45mm	800mmx500mmx45mm	413mmx330mmx90mm	610mmx300mmx150mm
Module Volume [dm3]	0.973125	18	12.2661	27.45
Configuration	1S4P	1S6P	1S5P	1S2P
Nominal Energy	16.6 kWh	12.2 kWh	13.5 kWh	14.2 kWh
Sum Module Mass	~130 kg	110 kg	90 kg	110 kg
Sum Module Volume	~280 L	~110 L	~61 L	~55 L
Gravimetric Density	~130 Wh/kg	~115 Wh/kg	~150 Wh/kg	130 Wh/kg
Volumetric Density	~60 Wh/L	~110 Wh/L	~220 Wh/L	~260 Wh/L
Availability	+	+	++	++
Power Capability	-	-	+	+
Cost	-TBD-	-TBD-	+	+
Useful Life	+	+	-TBD-	-TBD-

Table 9 Summary of Supplier Feedback on SuperTruck II Battery Specifications

Akasol was selected as the battery supplier for the SuperTruck II program, using a Li-Ion LFP based chemistry for the 48V battery packs. Three individual battery systems from Akasol were ordered for SuperTruck II. Each system was used for different powertrain integration tasks throughout the powertrain development project, summarized as;

- Performance and control concept testing
- Vehicle electric and cooling system integration at Kenworth
- Integrated power electronics testing and integration at Eaton

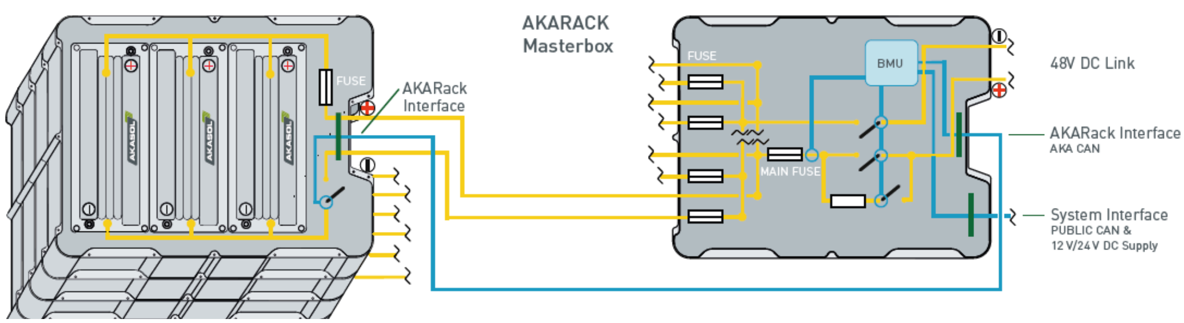


Figure 108: Akasol Battery Rack and BMS system Lay-out

A detailed analysis together with the battery supplier was performed to define an appropriated error handling of the Battery Management System (BMS). Objectives of this study were to ensure battery safety (avoid thermal overload) and lifetime (current over load & minimum State of Charge), while ensuring that all other power electronics of the mild hybrid powertrain are protected in the case of an error fault on the batteries resulting in opening the contactors of the BMS. The calibration settings for each error handling code are now defined and implemented in the BMS software release.

The team worked on the selection of the optimum battery architecture using detailed simulations. The optimum number of battery packs being either 2 or 3, having a battery power capacity of 18 versus 27 kW respectively, needed to be defined for best freight efficiency Finding the optimum freight efficiency is based on determining the trade-off between the maximum battery charging capacity and the mass of the battery pack itself. The battery supplier provided the required data and associated limits and capabilities of the two potential battery systems. Next, the battery models were implemented in the plant model and connected to the latest control strategy for battery State of Charge (SOC) management. The battery management ensures optimum efficiency while maintaining sufficient SOC for hoteling at the end of the drive cycle. Both configurations were simulated over the selected 24-hour drive cycle for SuperTruck-II, covering 3 return trips between New Salem, ND and Beach, ND, followed by a 10hr Hotel event. The simulation results demonstrated that the smaller pack provides a higher freight efficiency of ~1%, which is our primary goal in the SuperTruck-II program. The higher functional capability of the larger pack (higher max. in/out current) leading to lower fuel consumption, was offset by the lower mass of the smaller battery system. The lighter 2-pack configuration was therefore selected for the SuperTruck-II vehicle.

Vehicle Cab Design

The exterior shape of the vehicle determines its aerodynamic efficiency but to be practical it must also take into consideration the requirements of manufacturing, operation, and maintenance over the life of the vehicle. Kenworth is applied its experience in the industry to design a tractor that balances these requirements to extend the performance of the vehicle, while ensuring there are no barriers that prevent its adoption in the marketplace.

The frontal area and the coefficient of drag, resulting from the shape of the vehicle, determine a vehicle's aerodynamic performance. For the Super Truck II program, PACCAR is optimizing the tractor design around the industry standard van trailer dimensions. We are investigating updates to the trailer to optimize performance but are assuming the basic shape and configuration will not change. This approach recognizes that there is a large trailer fleet in operation across the United States and the adoption of new tractor designs will be greatly enhanced if they can be used with moderate changes to the existing trailer fleet. Because this approach limits the opportunities for frontal area reduction, we have identified active suspension lowering as a technology that enables improved aerodynamics while remaining compatible with existing trailers. By lowering the entire vehicle while in over-the-road steady state operation we can reduce the aerodynamic drag without compromise to approach angles and snow clearance in secondary road operation where we operate at a higher ride height.

Three primary influences on the exterior shape of the tractor are powertrain packaging, operator interior environment, and ensuring there is smooth airflow transition around the vehicle and down the surfaces of the trailer.   The current Kenworth T680 model is the most aerodynamic tractor in the industry, but the traditional vehicle architecture used on this model limits the opportunities for significant aerodynamic improvement; the large vertical radiator located in the front of the engine significantly limits vehicle shape optimization and allows more underhood airflow than ideal.

Our Aerodynamic efforts began with multiple cab profiles, which gave us an understanding of what the ideal shape would look like. These early CFD simulations validated the improvement potential of the streamlined vehicle shape and provided the data that was later used in a sensitivity analysis to determine optimal frontal profile. The resulting shapes were then used as the boundaries for packaging exercises.

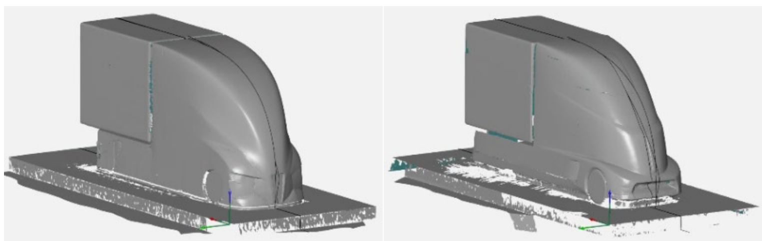


Figure 104: Early Design Examples for Aerodynamic Analysis

From there the Aerodynamic development proceeded with a multi-parameter design-of-experiments based study to optimize and refine the shape of the tractor and trailer aero devices. This was compiled by changing features of each system component and evaluating the results with CFD, then engaging in iterative loops with the powertrain, chassis, cooling, and interior engineers to find the best compromises where necessary. The conclusions from these efforts netted a shape that is capable of near ideal shape aerodynamic performance when combined with an optimized trailer but can also yield significant drag reductions when pulling a current production trailer with aerodynamic devices that are either commercially available today or could be in the next couple years. This was a key program objective since it is well known that the existing trailer population is likely to remain in service for years to come and that behooves us to ensure that our vehicle provides excellent performance with them as well.

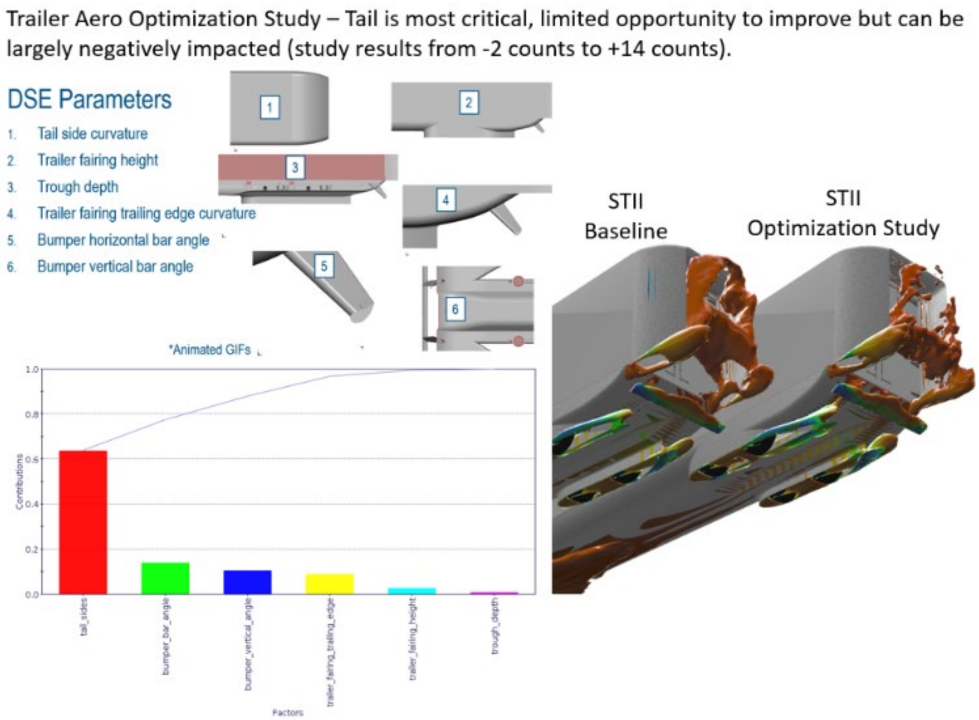


Figure 1050: Trailer Aerodynamics Optimization Study

Using similar methods to the one outlined above, the trailer, tractor underbody, and side-extenders have undergone characterization studies to refine the overall vehicle shape and determine which components or locations are most critical. In each case the ROI, impacts, and implications were evaluated to determine if the change would be included in the demonstration vehicle.

As an example, while the full tractor underbody does show a performance benefit, challenges exist for assembly, service, and routine inspection which may prevent adaptation of the technology. Another example, the team also investigating the benefit of shields to prevent impingement on trailer chassis and suspension components to yield similar performance gains with lower impact to customer and business operations.

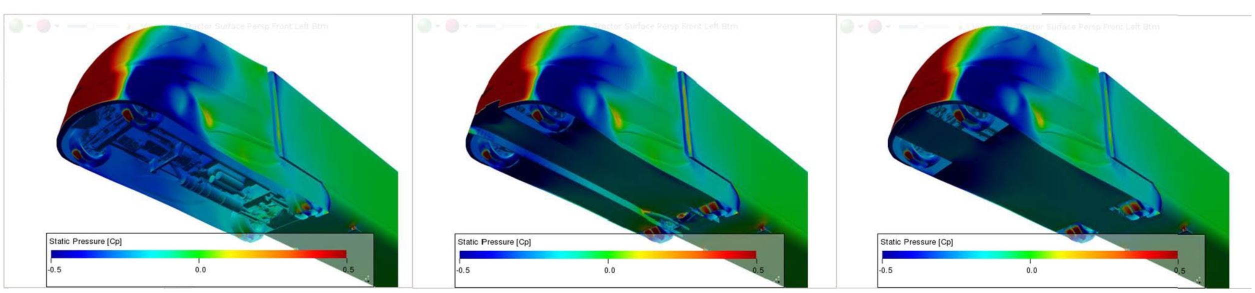


Figure 111: Static Pressure Results for Underbody and Side Extenders Aerodynamic Studies

Similarly, analysis of tractor side-extenders to close the trailer gap indicates that the performance of a sealed coupling can be effectively replicated with gap distance between 0-4”, beyond this point the drag is significantly increased. Creating such as short trailer gap with the side extender was a challenge for the team. A variety of “gap treatment measures” were evaluated, primarily focused on passive, movable panels. The risk of damage and poor ROI of these systems led us to investigate and optimize a fixed side extender to compare against. The resulting design yielded a compromise that would give our customers the best possible value at this point.

Styling efforts were making excellent progress during this period. We selected a primary styling language and theme for further development. A defining feature is the covered front wheels, which was a request from the aerodynamic team and supported with the chassis architecture, particularly the narrow track front axle and reduced scrub radius in our front suspension. This feature creates unique opportunities and challenges for the look and feel of this new vehicle.

Following many iterations, the styling and design teams refined the shape with a variety of clay models that helped the team share their vision with the organization and kick off the cascading activities that would be required.

Figure 112: 50% Clay Model – Completed milling, roof assembled and ready for use by styling/design/engineering teams

The activity of finalizing the Body in White (BiW) surfaces followed. Developing a good strategy for part breakup and details like headlamps, radar, and openings for cooling was a challenge. As the styling was refined and the related technologies defined, the 50% clay model was updated as well in order to best understand the full implications. As an example, the headlights needed to comply with legal requirements as well as packaging within the exceptionally small space claim that was available for them behind the A-surface and in front of the tire space claim. The team engaged possible suppliers who generated quotes, and a leading supplier quickly emerged. Leveraging existing technology, Depo Auto Lamp was able to achieve the target shape, features and stay within budget.

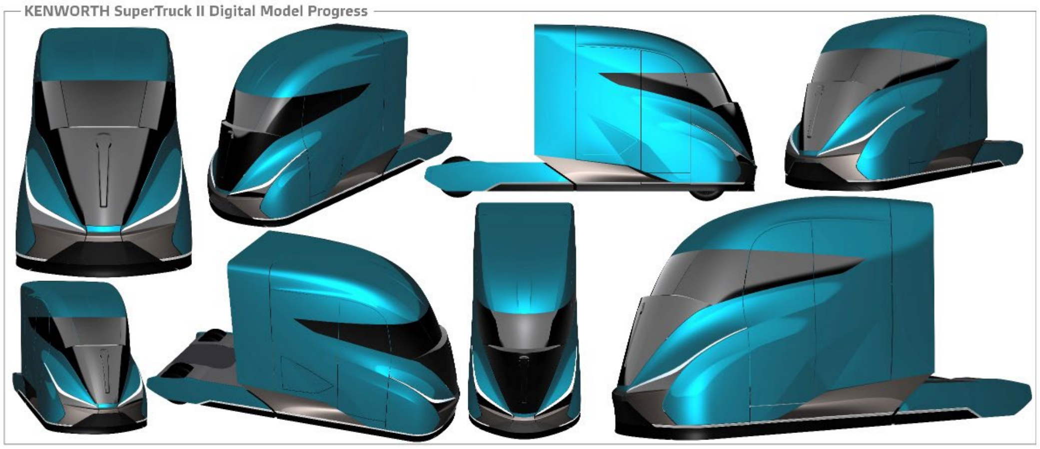


Figure 113: SuperTruck II styling as presented at the Kenworth styling review meeting

Figure 114: The 50% clay model before implementing the latest design updates

Figure 115: Progression from decided body split lines to clay model and headlight proposal.

Figure 116: Final Headlight – Daylight running light illuminated

Once all the exterior surfaces were fully defined and split lines defined the BIW structure and outer-body panel development was initiated. The team generated a BIW concept model that factors in supports and packaging for all vehicle systems such as electrical, HVAC, air systems, interior designs, improved visibility, and cab suspension. High content areas where multiple vehicle systems will interface became the focal points for development. Provisions for HVAC ducting, wire routing paths, pockets for mounting wiper motors and space-claims for brake valves are included within the new BIW concept model pictured below. The series of images below, from left to right, depict the current state of BIW model development, packaging for all vehicle systems surrounding or attaching to the BIW, the door aperture and steps, and storage space for typical items.

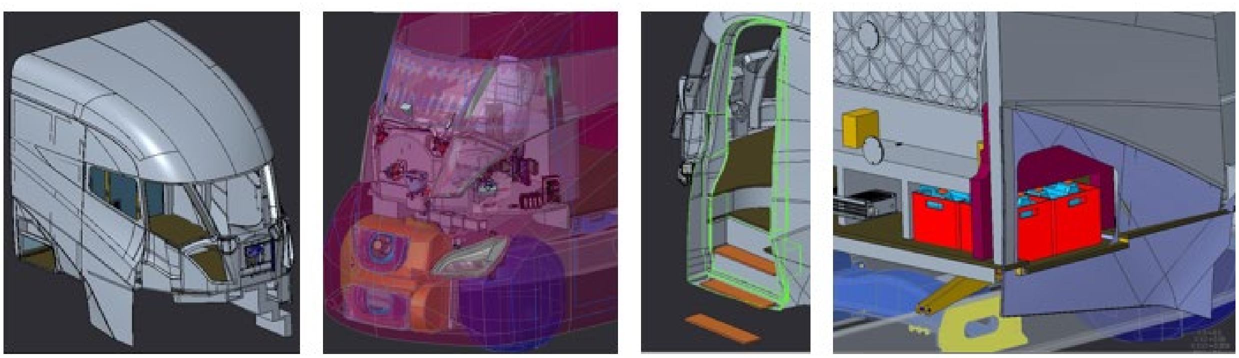


Figure 117: BIW structure development, packaging BIW with support systems, door & step interfaces, storage evaluation.

Creating this higher-level definition BIW structure led the team to encounter two major challenges, the door mechanism with BIW interfaces and the covered front wheels. With previous assessments, concluding a generic panning motion and pop-out step mechanisms would be utilized, detailed motion paths and assembly packaging began. Several solutions including 4-bar rotation, combinations of rotation with sliding, and purely sliding mechanisms were developed and evaluated. Packaging and complexity of sliding mechanisms as used on mini-vans were evaluated but ultimately ruled out for a number of reasons including complexity, packaging depth into the vehicle and assembly cost.

Complexity and cost helped rule out rotating combination mechanisms, leaving rotation only solutions for further consideration. This conclusion was the result of a decision analysis that considered these and other factors such as ergonomics, safety and reliability. Based on design requirements for opening angle, safety features, and desired mechanism, a study of existing products yielded a suitable solution for mockup on the treehouse scale model. This door mechanism, leveraged from the bus market, is now in hand and ready for modification into the scale treehouse model. Once operational, the team will test function and safety of the system, develop specific controls for integration into the overall vehicle architecture, and develop latching solutions to be included in the BIW structure.

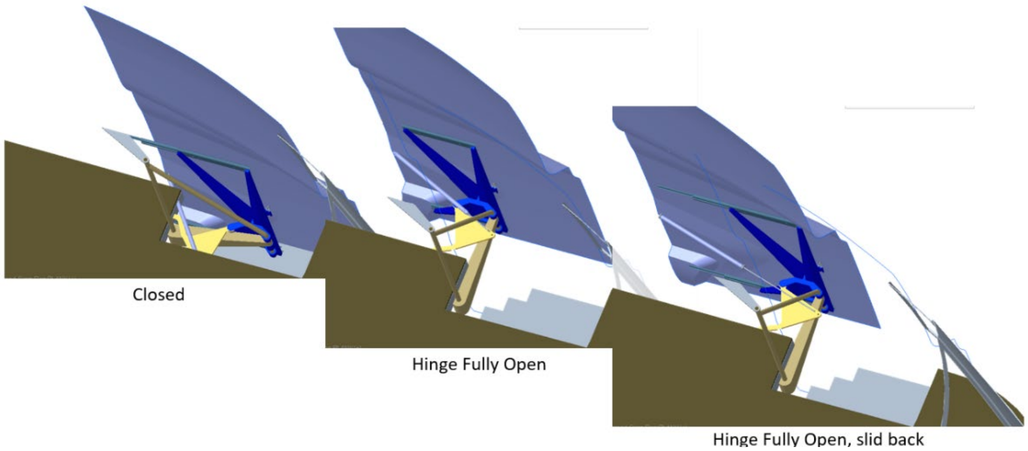


Figure 118: Illustration of possible door opening mechanism.

In order to evaluate this door mechanism as well as the related steps and user facing hardware, a full-scale mockup of the cab and sleeper platform was created. This enabled users to experience getting in and out of the cabin space, transition from driving to resting space, experience the central driving position, and evaluate exterior visibility. The mockup continued as a platform to trial interior concepts and placement of major components as we progressed towards a full interior design freeze.

Figure 119: Early cab mockup

To complement the mockup, a virtual visibility study comparing our current product T680 to the Super Truck shape was conducted. Due to the driver shifting to the center, there is a small loss in direct visibility on the left but a significant gain in visibility on the right. To understand the potential impact to the windows and structure we used the T680 ground strike array on the Super Truck model. To augment the revised visibility of the vehicle and support aerodynamic goals, an improved camera and mirror display system for is under development. The below images capture initial visibility assessments for front and side views.

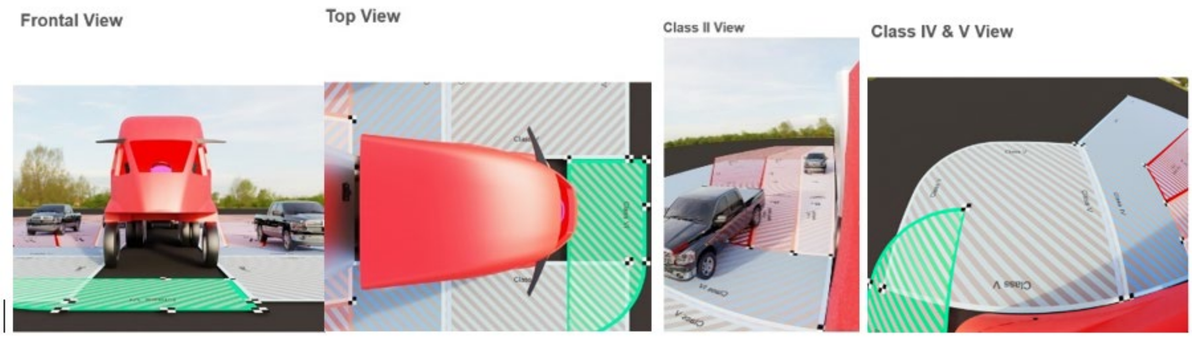


Figure 120: Visibility Evaluation

Three primary cab access ideas were investigated and mocked-up with main focus on cab access, safety and convenience. Some of the challenges considered were the large size of the door and the trade-off between sleeper platform floor area and stair rise/run. This has led us to investigate a lower step that moves outboard, outside the vehicle envelope, in order to improve the stair run and also enable a smaller door. Any step needs to be robust and predictable so simple mechanical connections to the door were targeted initially. Ultimately, the preferred solution was to add an additional 4th step which improved the ergonomics and allowed less severity to be assigned to lower step failure. The second, non-moveable, step is now sufficiently close to the ground to make the lowest step a convenience rather than a necessity. A key element to ensure access ease even if power is lost.

The center driving position benefits from rear view camera technologies. Our exterior camera mounting system consisted of a highly aerodynamic wing-shaped arm that houses rear view and top-down cameras, as well as IR/LED modules for night vision. Extensive CFD simulation was used to create a design with minimal drag and optimal air flow characteristics. An early prototype version was 3D printed internally at Paccar Technical Center which was then used for fit and function testing.

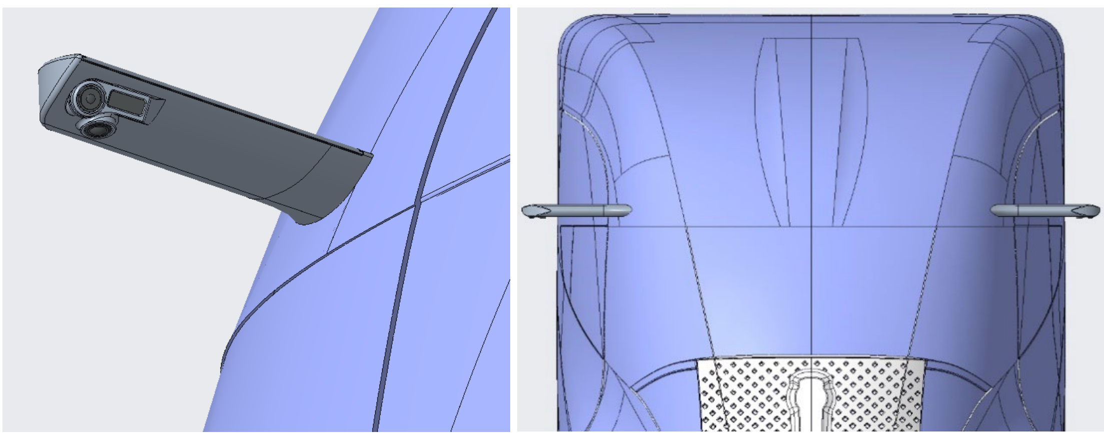


Figure 121: Exterior Rear-View Camera Mirrors

Interior

Moving to the inside of the cab and sleeper, the engineering and styling teams have developed and finalized the driver package theme. Establishing this set direction for content, layout and overall shapes enables the engineering team to progress ideas into functional concepts, incorporating requirements for ducting, routings, electronic modules, ergonomics and visibility as examples. Increasing visibility has been a unique challenge with this product, a complication born from the center driving position and ideal location for displays. Lowered belt-lines on side windows improve side and rearward visibility while added front corner windows significantly improve forward and nose visibility. Creating a cohesive theme for the interior meant incorporating HVAC vent locations for defrost and occupant comfort, leveraging information from NREL simulation studies to define ideal locations, consideration of major structural requirements, and iterating concepts with support from Mahle for HVAC performance.

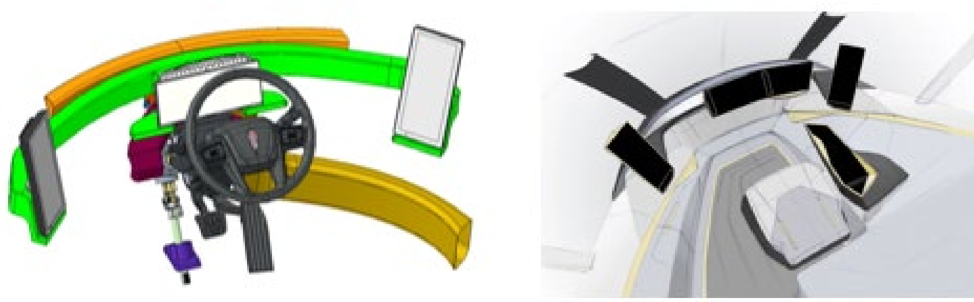


Figure 122: Engineering layout & analysis (left), and final driver package theme (right)

To expand understanding of a center driving experience and evaluate visibility changes, 3D models of the current product and SuperTruck II interiors were created for a virtual reality experience exercise. The team exported models to Technicon, assisting with rendering and VR demonstration. In parallel, models created to retrofit the mock-up cab, reflecting the latest theme. These changes enable the mock-up cab to continue serving as an evaluation and development tool. This continued to be the case through the development effort, culminating in a full-scale interior mockup using milled-clay and actual hardware, enabling internal buy-off before concepts were frozen and sent to suppliers.

Point Innovation was selected as the supplier responsible for interior design and integration. Point Innovation collaborated with KW studio and Interior engineers on design theme, material selections, and interior layout of center driver. Particular focus on instrument panel, 2-tier headliner, sleeper compartment, packaging of electronic controllers on left side, right side, and rear console.

The center driver position required an All-Belt-to-Seat driver seat with electric height adjuster. Our driver seat was developed by Gramag, a current KW seat supplier. To reduce R&D and manufacturing costs, Gramag incorporated existing motor/lift system, belt system, cushion/back frame, and sub-assemblies for our prototype seat. Seat covers were custom made to match our interior styling themes. Point went on to take the seat further by developing new foam and seat covers according to the chosen materials, colors & theme by styling.

Figure 123: Seat design details

Trailer Development

CFD (Computational Fluid Dynamics) Analysis has shown that we could provide an estimated 43% increase in drag performance over a standard Utility trailer with an optimized skirt + tail concept. These aero aids, however, would represent a significant departure from what is available in the aftermarket at this time or within the near future. Considering this and recognizing that our aero development goals had been exceeded by significant margins, it was decided to seek a path that would be more representative of what is possible with the existing population of trailers in the field. To that end, we partnered with Ridge and Stoughton to develop the auxiliary aero components and have them ready for testing by the time the tractor was completed.

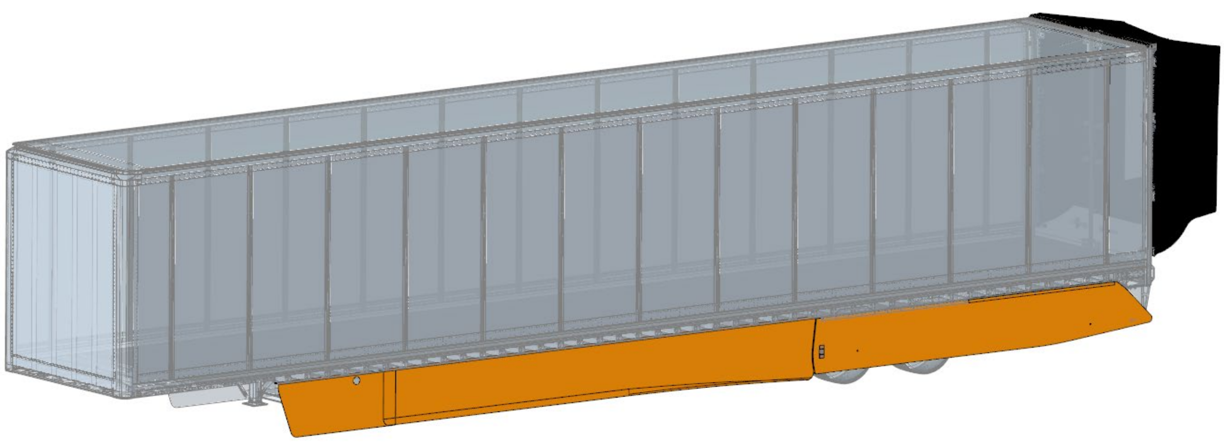


Figure 124: Trailer with aero skirt

Figure 125: Trailer skirt prop rods, trailer boat tail deployed, and stored

Cab Suspension

The cab suspension is a design element that was found to be an enabler to weight reduction by virtue of how it interacts with the cab, chassis, outer body, and driver. The driver's interactions were of particular importance, safe and comfortable operation of the vehicle being the highest priority. The design we chose provides the driver with a comfortable ride without the need of an air suspended seat, a significant weight reduction. This also means that the driver will not be moving in relation to the steering wheel and pedals while he operates the vehicle, leading to less driver fatigue from having to account for all those changes. Additionally, the suspension geometry was optimized to have anti-dive characteristics, leading to more confidence while braking. This was a complaint that has often been voiced by North American drivers after their first drive of one of our DAF European spec vehicles. Interestingly, the findings from our initial anti-dive cab-suspension evaluations were shared with our European divisions and folded into their current developments, proving the value of these projects to the advancement of technologies.

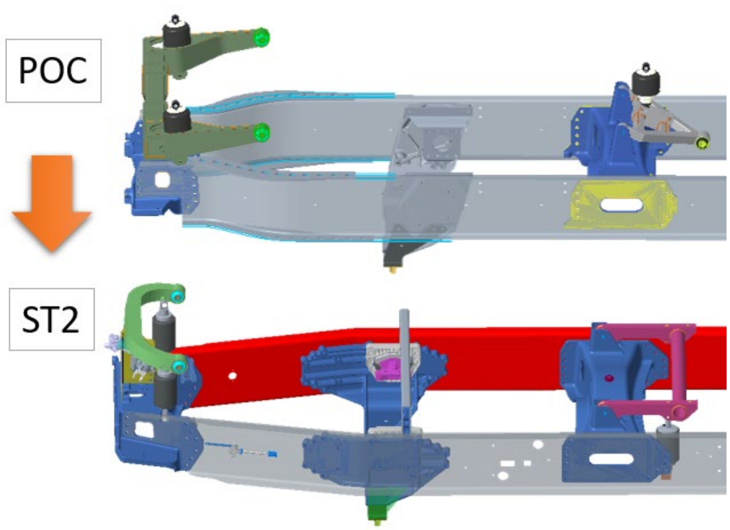


Figure 126: Illustration of the cab suspension, the POC "Mule" vs the SuperTruck 2

HVAC

As with the other technologies, a great variety of options were evaluated. The team developed plans for conditions, equipment and performance criteria that enable comparison to current product as reference, and correlation to simulation models. Working closely with NREL, we selected to partner with Mahle to develop a heat pump assembly. Incorporated in the design is a chiller plate for the refrigerator in the heat pump assembly, which will be placed into a Kenworth-developed refrigerator cabinet. By removing the additional compressor and evaporator from the refrigerator, weight and heat transfer into the cab are reduced. This also provided benefits in packaging, economy, and flexibility for future use. Electrical and controls engineers developed wiring harnesses and tuned control software to operate the refrigerant expansion valves, coolant pump, and coolant valves from the vehicle control head such that the operator would have a more seamless interaction.

Figure 127: Heat Pump Assembly as fitted to Mule

Analysis and simulation modeling were performed to optimize the diffuser locations with an emphasis on windshield demist and occupant comfort of the demonstrator vehicle. NREL and Dassault Systems were developed proposals for duct locations based on time to clear the glass and human comfort models. Evaluations are conducted at 5th percentile -6C ambient conditions for heating, deemed to be the worst case condition. By applying the Berkeley Comfort and Predicted Mean Vote Sensation the team has evaluated iterative proposals and homed in on ideal diffuser locations, which were optimized in parallel to interior layout studies. This process enabled tuning interior surfaces and storage layouts to promote circulation around an occupant when in the driver seat, increasing the predicted comfort sensation while reducing the total conditioning work.

Shifting focus to defrost and demist performance, simulation studies were conducted at the 5th percentile condition to evaluate time to clear the windshield and side glass. The heat pump enables near instant heat at the diffuser, dramatically improving defrost time when compared to traditional engine coolant based systems. The heat pump is predicted to enable full visibility within 5 minutes without the supplemental heating elements placed within the windshield laminate layer.

Mahle is designed the hardware for the heat pump assembly and selected all of the sub-components. The system is comprised of an indirect coolant to refrigerant condenser with a traditional evaporator dedicated for cooling mode, 48 Volt scroll compressor and high pressure heat exchanger dedicated to heat mode. A common air handler is used to provide conditioned air to the forward and side windshields, driver occupant space, ventilated driver seat and rear sleeper occupant space.

PACCAR provided an extensive technical requirements document to define the performance characteristics and packaging constraints. The heat pump system has three modes: AC cooling; heating with dehumidification; heat mode without dehumidification. Each of these modes have a different set of hardware requirements. The AC evaporator is dedicated to cooling mode. Heating mode uses a traditional heater core while a thermal mass is available from the engine. The engine is not required to be running to take heat from this core. Alternately, for hoteling, heat mode uses a high-pressure refrigerant to air heater core.

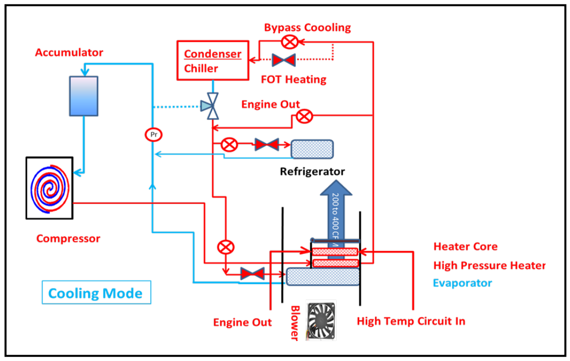


Figure 128: Cooling Mode

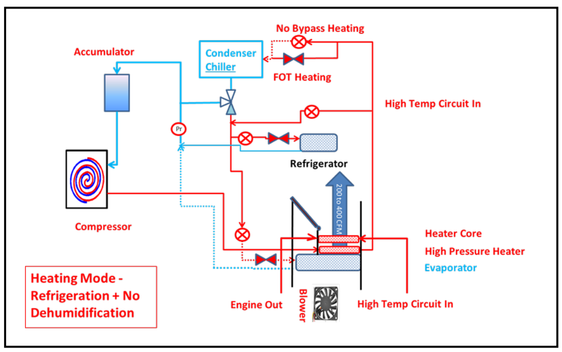


Figure 129: Heat Mode

The heat pump assembly is designed for ease of manufacture and assembly. The assembly would be received from the supplier pre-charged with refrigerant. This eliminates the need for a refrigerant charging station at the truck plant or the need to deliver and store refrigerant. Elimination of this Green House Gas and related process reduces the risk of releasing CFCs into our atmosphere. The unit is intended to be a drop in unit which is capable of supporting a vision for the capability to build any truck model at any build plant.

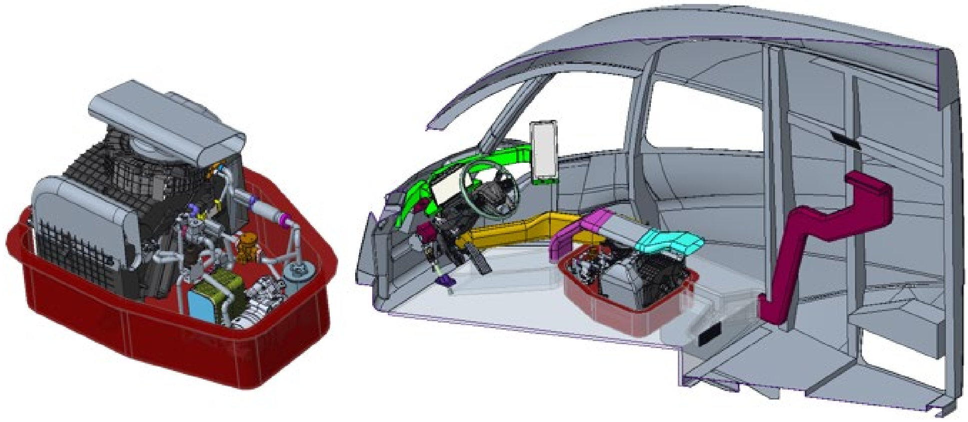


Figure 130: Left: Heat pump assembly, modular and pre-charged. Right: Heat pump assembly within STII demonstrator, including ducting for cab and sleeper.

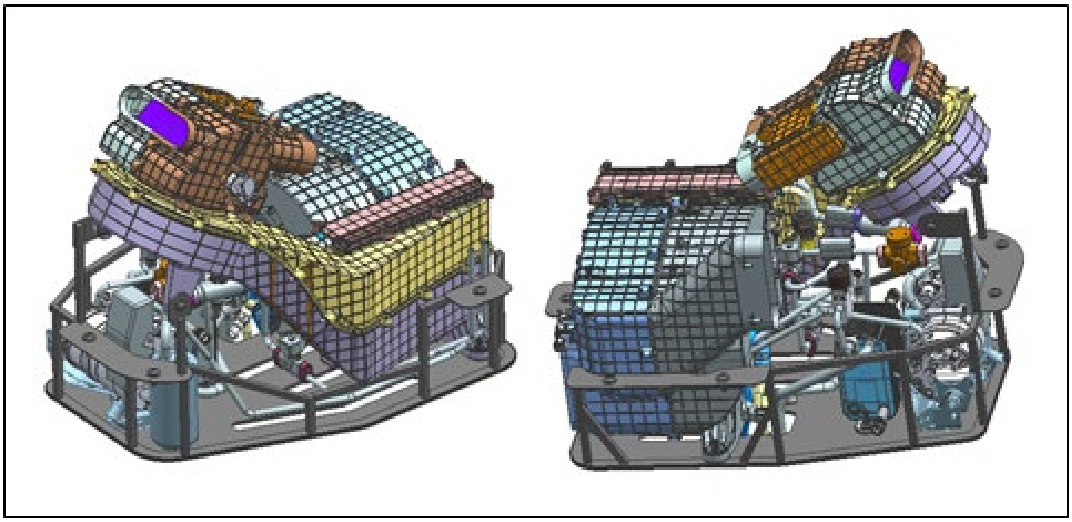


Figure 131: Completed Heat Pump Assembly

Engine Cooling Module:

The traditional approach on trucks is the use of large radiators placed in front of the engine to take advantage of the ram air through the radiator while at cruise and utilize an engine powered fan when under increased loads. The improved thermal efficiency of the SuperTruck II powertrain lowers demands on the cooling system and allowed the use of smaller radiators. The use of electric fans allowed mounting the fans on the positive side of the cooling modules to better optimize the needed grill area.

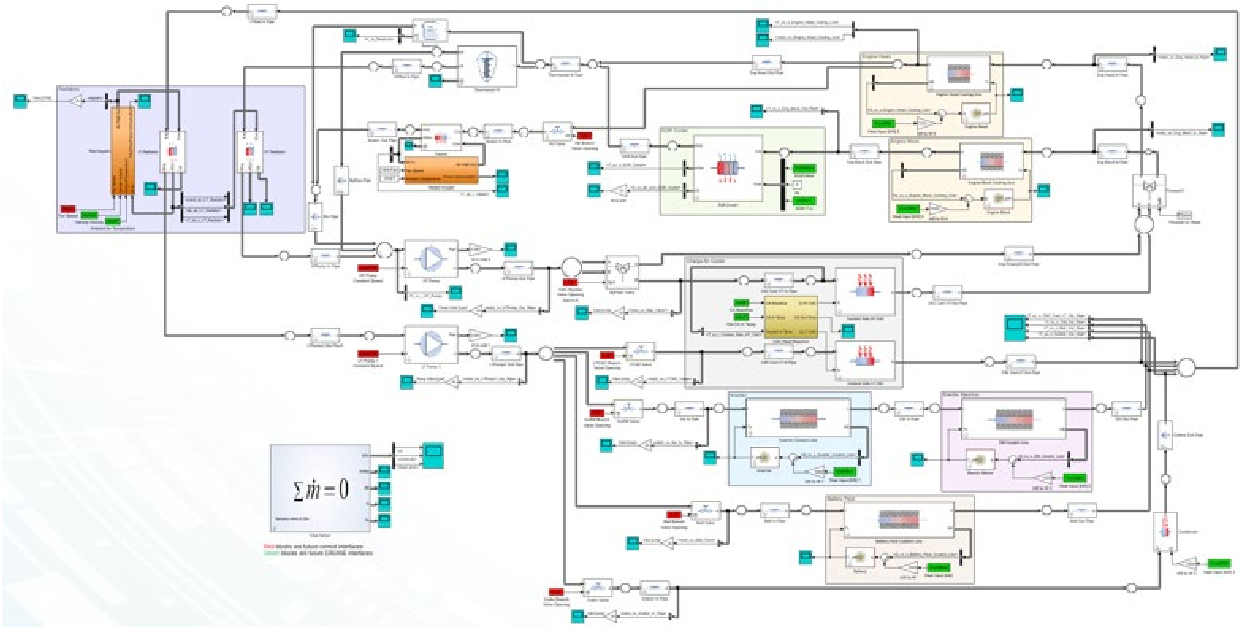


Figure 132 – Cooling System Plant Model

A cooling plant model was created to evaluate the circuits needs. The model includes the high temperature (HT) and low temperature (LT) cooling loops with the pumps, valves, heat sources and radiators that are in each of the systems. This model has been tested to verify it operates as anticipated when connected to the AVL-CRUISE™ vehicle model. The model is being updated with data provided from testing and simulation. The requirements for the control system have been captured and the control architecture is being confirmed prior to developing the control strategy.

With the cooling diagram solidifying, packaging studies evaluated a variety of layouts. A key objective has been to minimize the exposed frontal area for aerodynamic performance. This normally incurs pressure drop increases due to a reduction in core size, which challenges for airflow with high system restriction. To resolve the restriction challenge, centrifugal fans were selected and optimized using CFD. Through several iterations of radiator size, flow direction, fan wheel and motors, a feasible cooling system solution was defined.

The final solution reduced our core size by nearly 40% and our grill area by 60%. Most importantly, the total power draw allocated to cooling fans was reduced to 6kW, nearly an order of magnitude lower than our baseline vehicle's 32" on/off fan.

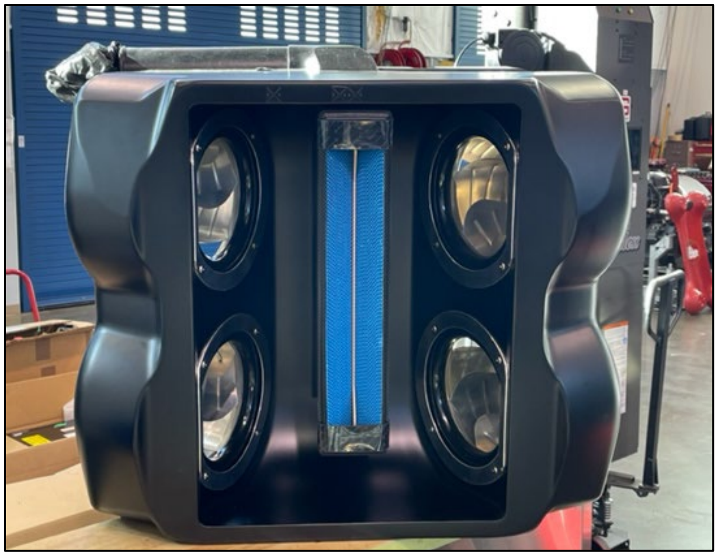


Figure 133 – Cooling Module with 4 Centrifugal Fans

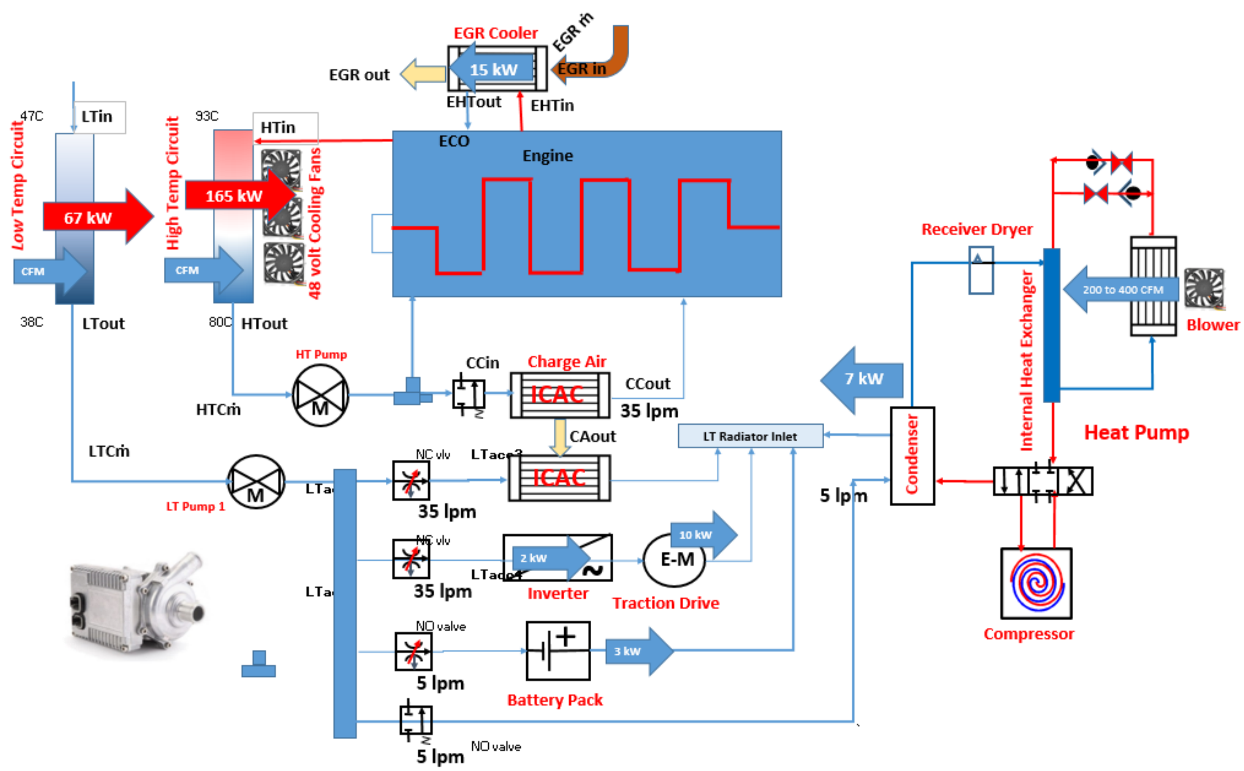


Figure 134106 - Cooling system schematic, two-circuit, two-stage iCAC

Summary: Mahle 1D Analysis & Simulia 3D CFD

Low Temp Radiator (coolant: 50/50 water glycol)

system	core dimension	fin density	tube number	heat transfer	air inlet flow	air inlet velocity	air inlet flow	air inlet temp	air outlet temp	air inlet pressure	air DP	coolant inlet flow	coolant inlet flow	coolant inlet temp	coolant outlet temp	coolant DP est.
-	mm3	fins/dm	-	kW	kg/s	m/s	m³/s	°C	°C	kPa	Pa	LPM	kg/s	°C	°C	kPa
342. 55.	771.0*808.3*42.0	55.0	89	26.5	4.8	6.7	4.2	38.0	43.5	102.0	309.7	42.3	0.74	53.0	42.7	8.6

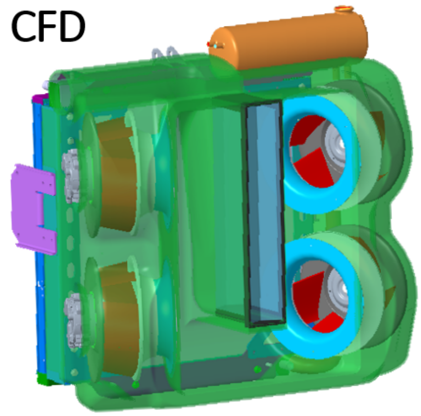
High Temp Radiator (coolant: 50/50 water glycol)

system	core dimension	fin density	tube number	heat transfer	air inlet flow	air inlet velocity	air outlet flow	air inlet temp	air outlet temp	air DP	coolant inlet flow	coolant inlet flow	coolant inlet temp	coolant outlet temp	coolant DP est.
-	mm3	fins/dm	-	kW	kg/s	m/s	m³/s	°C	°C	Pa	LPM	kg/s	°C	°C	kPa
342. 59. 4	771.0*808.3*42.0	62.0~	89	226.6	4.8	6.9	4.9	43.5	90.2	415.5	300	5.08	106.0	93.7	46.2

Fan Performance

Code	Airflow (kg/s)	Airflow (m^3/s)	Air Temp (°C)	Hex Air DP (Pa)	Veh Air DP est. (Pa) *	Fan Static Pressure (Pa)	Fan Air Power (kW)	Total Fan Electrical Power	Fan Qty	Fan Diameter (mm)	Per Fan Airflow (m^3/s)	Per Fan Airflow (m^3/hr)	Per Fan Electrical Power (kW)
LTR+HTR	4.80	4.20	38.0	725	150	875	3.679	9.198	4		1.05	3783.46	2.3

			New package (031)
Fan Speed		[RPM]	4000
LT RAD	BTT	[°C]	42.4
	TTT	[°C]	52.7
	Air on temp	[°C]	38.0
	Air mass flow	[kg/s]	4.82
HT RAD	BTT	[°C]	94.2
	TTT	[°C]	106.4
	Air on temp	[°C]	43.2
	Air mass flow	[kg/s]	4.81



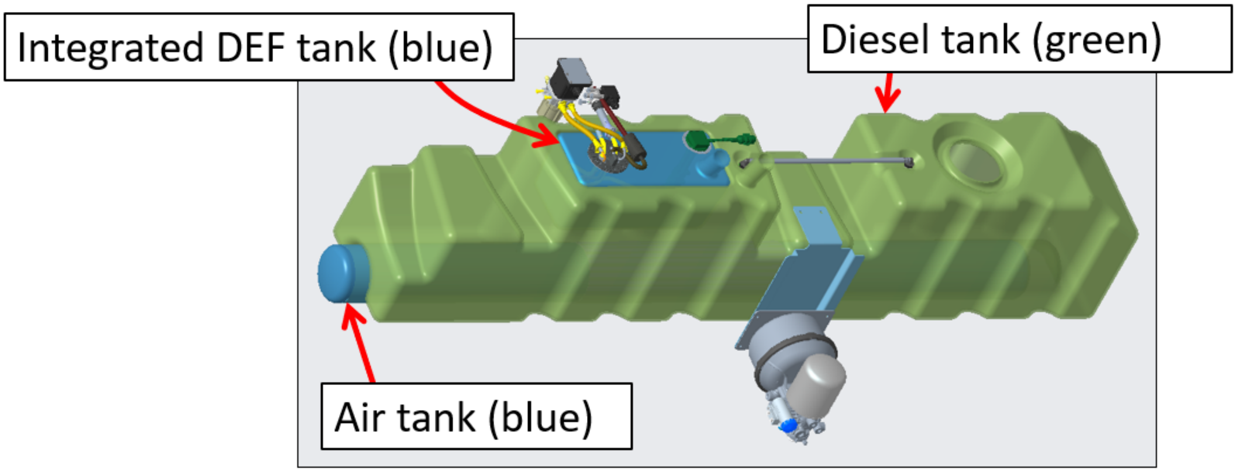
Minimum Hydraulic Areas		
At the 4 Fan Inlets	0.142	M^2
At the Seal to the Grill B-Side	0.413	
At the Grill, not including mesh	0.383	

- Final grill area is under 0.4M^2.
- Cooling targets met with license plate and radar in lower grill

Figure 135107 - Cooling module summary, two-circuit, two-stage iCAC

Fuel, DEF, and Air Tank Assembly

The Kenworth team completed a concept for a sub assembly of the diesel, DEF, and air tanks that can be assembled to the chassis. The assembly also includes related components, including pumps and air dryers. The assembly will efficiently use packaging space between the frame rails, while improving quality, manufacturability, and cost. Packaging in unused space between the frame rails frees space outboard of the rails for other components. Quality will improve because many fuel, DEF, and air connections can be made in a more controlled environment, before the assembly is installed on the truck. Manufacturability also improves by moving tasks from the main line to a separate build cell. Cost will improve due to the use of a plastic diesel tank (rotomolded in this case, but blow molding is a long-term direction) in place of a welded and polished tank.



Integrated Tank Assembly

Figure 136108 – Integrated Fuel/Def/air tank

Final Vehicle Demonstrator Freight Efficiency Testing

The PACCAR SuperTruck II freight efficiency demonstration activities followed a stepped approach from functional testing, performance calibrations, to final road demonstration. First, the team gained over 250 miles on the Kenworth SuperTruck II demonstration vehicle while confirming the functionality of several integrated systems. Systems such as the mild-hybrid powertrain, cooling, e-power steering, HVAC, and braking were all thoroughly verified to confirm functionality and safety following PACCAR's standardized prototype vehicle practices. The first functionality tests included validation of the installed fuel economy measurement systems on both the baseline and SuperTruck II demonstration trucks to ensure integrated time resolved measurements equaled the total weight based measured fuel consumption after each test.

After the initial functionality testing, the team completed an initial fuel efficiency test on the test track. This test gave more insights on the necessary improvements to calibrate the systems for the most efficient truck performance. Functional testing concluded with the successful completion of a road worthiness evaluation and DOT inspection.

As a next step, the team completed the truck tuning and verified truck performance during approximately 28 individual test drives on the PTC test track and a section of Interstate 5. While completing the final performance tuning of engine, transmission, electric power steering, cooling and the mild-hybrid system, the truck gained over 750 accumulated miles of testing.

The final step in the program was to perform the freight efficiency test using the HDCC demonstration route. Prior to the test the Kenworth SuperTruck II and baseline vehicles were prepared for testing and the official test was initiated by completing the following standardized steps:

- Both trucks were loaded to 65,000 lbs.
- The tire pressures were set to manufacturer specifications
- Both trucks completed a warmup process of 60 minutes of highway operation
- Vehicles idle for 1 minute
- Lead vehicle starts the route
- Following vehicle starts the route 1 minute later
- Both vehicles accelerated to target speed and the cruise control is engaged as quickly as possible
- Both vehicles complete their road testing sections
- When both vehicles arrive at the end of the test they idle for 1 minute

Figure 137: Freight Efficiency testing showing the Kenworth SuperTruck II and Kenworth T660 baseline vehicle

The final result of the test is a 105% freight efficiency improvement over the entire composite cycle using traditional weighting factors developed internally at PACCAR to represent the US national average class-8 vehicle usage. The figures below show the fuel consumption of the SuperTruck II truck and baseline truck. The upper portion of the plot visualizes the instantaneous fuel consumption, while the lower portion of the plot visualizes the accumulated fuel used.

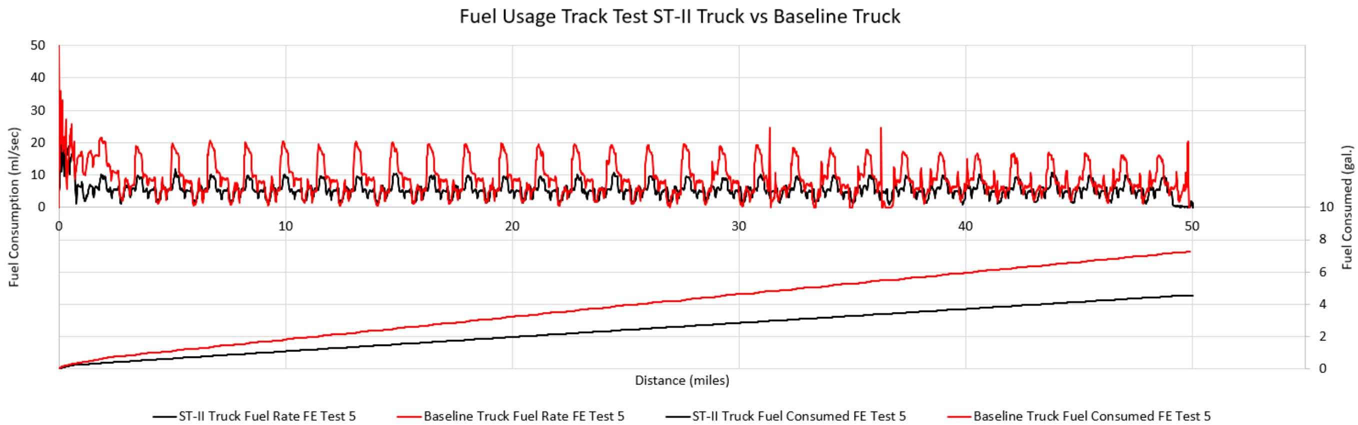


Figure 1098: Time resolved and integrated fuel consumption of the Kenworth SuperTruck II and Baseline vehicle on the PACCAR Technical Center test track

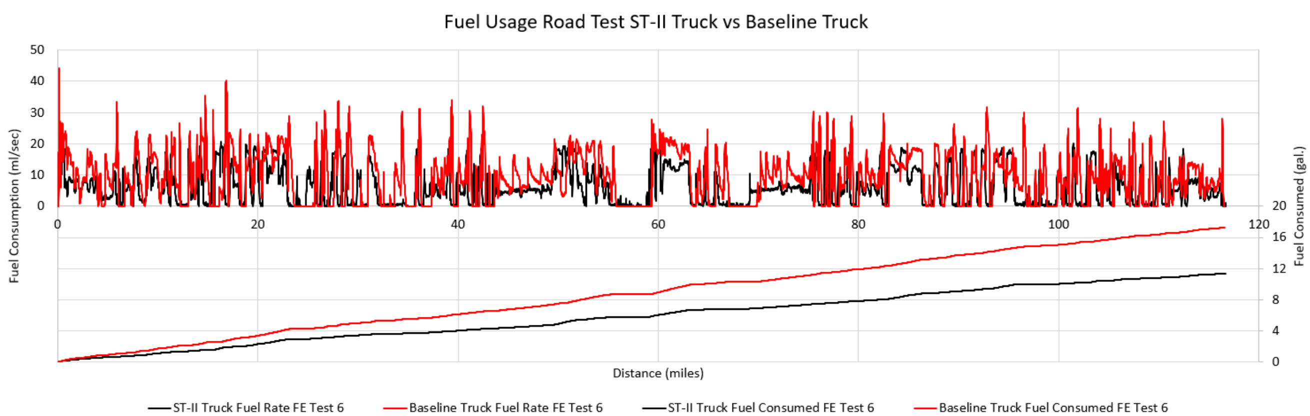


Figure 139: Time resolved and integrated fuel consumption of the Kenworth SuperTruck II and Baseline vehicle on Interstate 5

The plot below shows the fuel consumption during steady-state operation at highway cruising speed. The upper portion of the plot visualizes the instantaneous fuel consumption, while the lower portion of the plot visualizes the accumulated fuel over a 5 mile section of the demonstration route. During typical highway cruising operation a freight efficiency improvement of 136% over the baseline was measured, with the Kenworth SuperTruck II vehicle showing an actual fuel consumption 12.8 mpg.

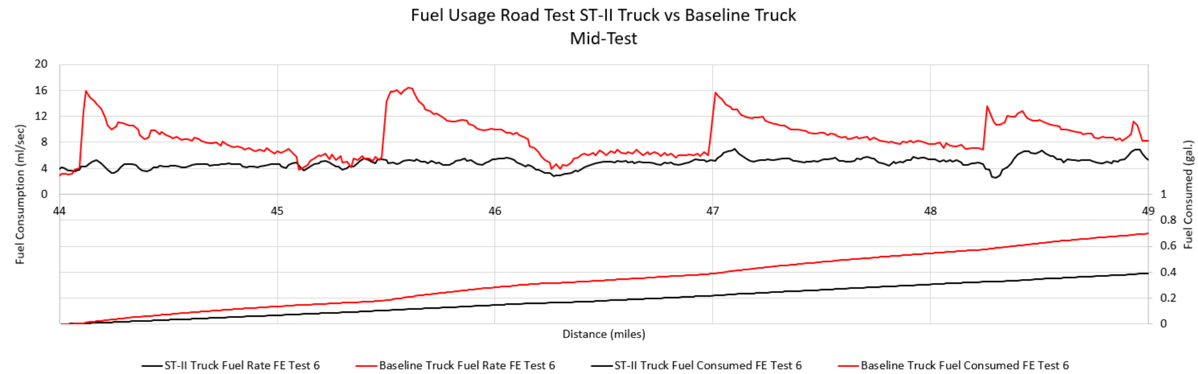


Figure 140: Time resolved and integrated fuel consumption of the Kenworth SuperTruck II and Baseline vehicle on Interstate 5

A more detailed performance analysis, supporting the 12.8 mpg fuel consumption reveals that the Kenworth SuperTruck II operated steadily in gear 12 resulting in an optimal engine speed of ~ 970 rpm, right in the sweet spot of the engine efficiency map. In addition, during the steady state test sections the truck maintained a consistent state of charge of the battery and the inverter provided power for all the accessory loads meaning no stored electric energy was used to achieve the efficiency numbers measured (i.e. it was based on diesel fuel consumption only). The plot below shows a summary of the efficient operation of the engine and the mild-hybrid system components during this steady state operation.

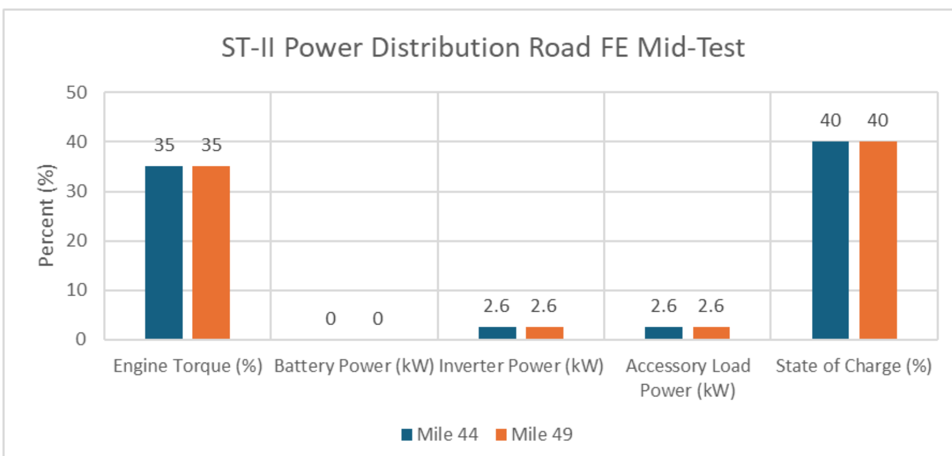


Figure 141: Energy consumption break-down of the Kenworth SuperTruck II powertrain during highway cruising operation

During on road operation of the Kenworth SuperTruck II vehicle, the mild hybrid powertrain uses several strategies to maintain and charge the 48V Li-Ion battery packs to improve overall fuel consumption.

To understand the SOC charge sustaining strategy, the Kenworth SuperTruck II vehicle demonstrated that the inverter can supply the needed power for the accessory loads and charge the Akasol battery packs to maintain a consistent state of charge. During these times, the engine power draw is only for the engine load, there is no power draw from the battery.

Looking at one of the steeper grades of the test route from mile 50 to mile 69, the SuperTruck II vehicle accelerates just before beginning the ascent. At the top of the hill, as it starts the descent, the powertrain shifts into neutral coast where the engine runs at idle speed to reduce fuel consumption. Once a steeper downward grade is detected, regenerative braking is utilized to charge the batteries.

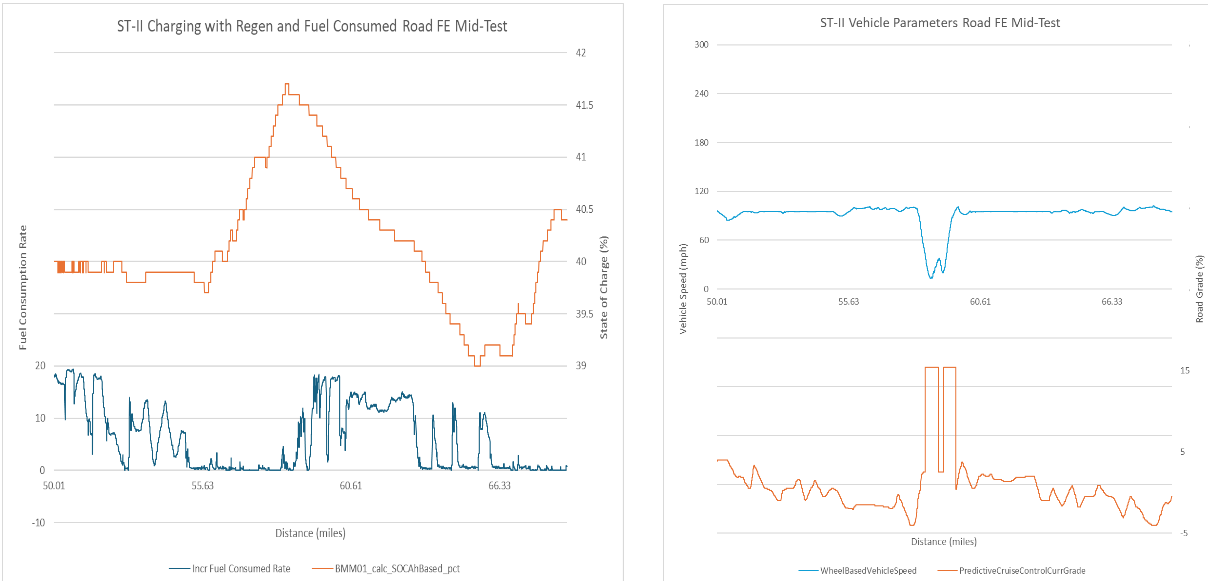


Figure 142: SuperTruck II Vehicle SOC strategy

The portion of the test route from mile 70 to mile 81 is after descending a hill. It shows that, during the downhill event engine load and accessory load increased slightly, lowering the battery state of charge. The powertrain inverter provides the power to the accessory loads. It is also noted that for some instances, battery power was also utilized by the inverter to provide the necessary accessory load. Once, the accessory power draw is stabilized, the inverter charges the battery and maintains the SOC, with its charge sustaining strategy.

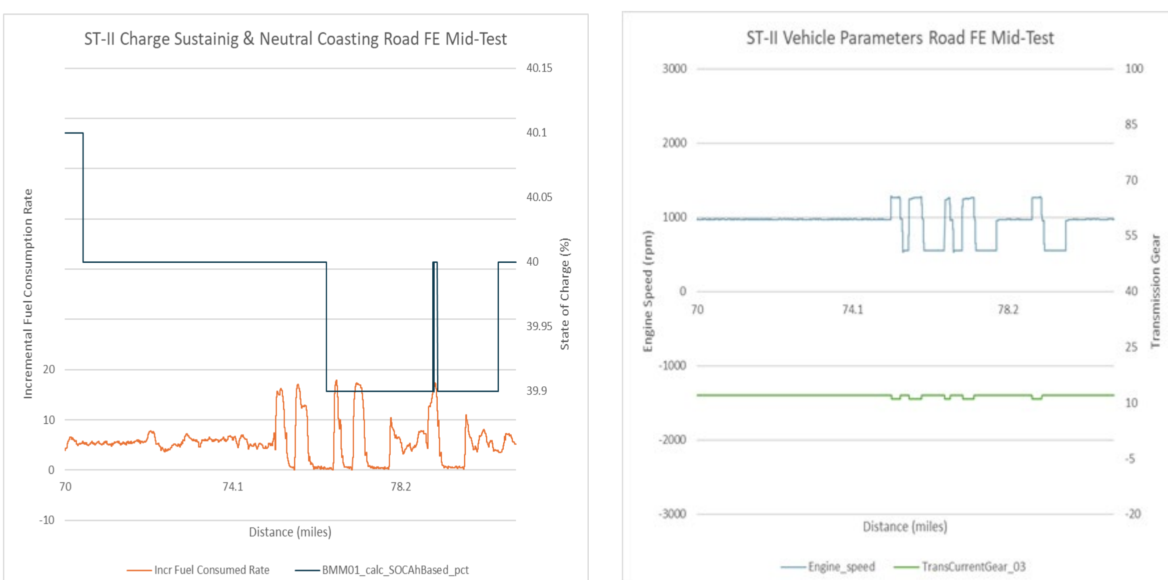


Figure 143: SuperTruck II Vehicle SOC strategy using regen mode

The powertrain strategy also improves fuel economy when possible, using neutral coast, where engine speed drops to idle speed, and the transmission shifts to neutral. These strategies together manage to maintain a steady state of charge of the battery. The plot below shows the SOC gain from regenerative braking and neutral coasting on a portion of I-5 against the power consumption from engine, battery, inverter and the accessory loads.

Based on the data collected from freight efficiency testing, it is concluded that the Kenworth SuperTruck-II vehicle robustly meets the 100% freight efficiency improvement target compared to the 2009 class-8 baseline truck. Measured improvements span a range from 105% improvement for the implemented national average representing drive-cycle (not including the most favorable elevation sections as originally proposed under the program by NREL), up-to 136% improvement under steady highway conditions at 60 mph while maintaining a battery energy neutral charging strategy.

9. Technology Transfer

The technologies developed under the PACCAR SuperTruck II program have contributed to the body of knowledge on freight improvement, impacting ideas on the basic elements of future commercial vehicles, not just diesel trucks. Key elements are:

- Approach the truck/trailer/driver/payload as a complete system
- Aerodynamics on tractor and trailer and as a combined system
- Tire rolling resistance
- Lightweighting based on design, advanced materials, and system integration
- Speed, configuration, and route management for efficiency
- Mild hybridization and electrification of accessories
- Overnight hoteling using vehicle batteries
- Driver ergonomics
- Power electronics
- Advanced Diesel engine technologies
- Aftertreatment Systems

One of the most insightful aspects of the PACCAR SuperTruck program is the electrification of accessories through mild hybridization. Direct fuel efficiency gains from removing load from the engine and related design opportunities they enable such as regenerative braking, overnight hoteling, are key to future truck designs. Hybridization of diesel powertrains has been investigated for some years, but The PACCAR SuperTruck II vehicle demonstrated the opportunities of a carefully selected architecture evaluated in large detail minimizing cost and complexity will providing maximum fuel efficiency benefits.

Lightweighting has always been fundamental to trucking. Less weight has always been pursued as a method to increase the proportion of customer freight, always with the goal of reducing cost. Unfortunately, lighter materials such as advanced steels, aluminum, magnesium, carbon fiber and other composites generally come with an increased raw material and/or processing cost, sometimes overwhelming the potential savings.

The design of the PACCAR SuperTruck II cab really sets the baseline for future vehicle programs of what can be achieved in-terms of aerodynamic drag reduction. This understanding directly feeds into our electrification programs where vehicle drag reduction is considered even more important due to the limited range and high costs of batteries for Battery Electric Vehicles. In addition, it helped PACCAR to understand which GEM category for GHG credits is technically feasible for diesel powered vehicles and how trade-offs work compared to other technologies such as WHR which comes with direct fuel efficiency savings but prevent the vehicle to be as aerodynamic as the Kenworth SuperTruck II vehicle.

Next to the aerodynamic impact of the WHR on the vehicle, detailed evaluation of the WHR revealed that the added weight, complexity, and systems cost unfortunately don’t outweigh the significant fuel savings the technology brings. It is recognized that the WHR system developed under the PACCAR SuperTruck II program captures every engine waste heat stream available to maximize its BTE improvement contribution, but a similar conclusion was drawn from less complex systems that would only capture exhaust streams as the highest “energy quality” waste stream of the engine system.

The PACCAR SuperTruck II engine demonstrator is a great example of technologies developed under the program that now directly supports production intended engine programs within the 2024-2030 time frame of start of production. Lessons learned in auxiliary electrification resulted in adoption of various 48V auxiliaries for power generation and emission control. The developed long-stroke concept demonstrated a significant impact on thermal efficiency when combined with the right valve timing, in- and exhaust port design and high efficiency turbocharger. Lessons learned are included in future engine architectures currently under development.

In addition, the combustion system developed by using a combined approach of state-of-the-art simulation tools and testing facilities, has been carried over directly into the PACCAR product process. Finally, the benefits of friction and parasitics reduction measured are now much better understood and ready to be implemented when GHG requirements or fuel prices justify the hardware changes to be implemented.

Overall, the value of the lessons learned from the SuperTruck II program are important to PACCAR and the trucking industry in general. It provided the right platform to evaluate new, sometimes out of the box, technologies in an early stage in a structured manner utilizing the Technology Readiness Level (TRL) process. The technologies that made it to production, but also insights obtained of technologies evaluated that will not have commercialization potential in the foreseen future, fully justify the recourses spent over the time-period of the program.