A HYBRID HEAVY DUTY DIESEL POWER SYSTEM FOR OFF-ROAD APPLICATIONS – CONCEPT VALIDATION

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ABSTRACT

A multi-year power system R&D program was completed with the objective of developing an off-road hybrid heavy duty diesel engine with front end accessory drive-integrated energy storage. This system was validated to deliver 10.5 – 25.6% reduction in fuel consumption over current Tier 4 Final-based 18L diesel engines, over various off-road machine application cycles. The power system consisted of a downsized heavy-duty diesel 13L engine containing advanced combustion technologies, capable of elevated peak cylinder pressures and thermal efficiencies, thermal barrier coatings, exhaust waste heat recovery via SuperTurbo™ turbocompounding, and hybrid energy assisting and recovery through both mechanical and electrical systems. Following the concept definition, design, and analysis phases of the program, the final phase focused on building and validating the performance and efficiency in laboratory tests. While aspects of the system such as start/stop and reduced off-road cooling package energy losses were only analytically evaluated, the main 13L concept engine with full hybrid system was successfully built and tested in steady-state and in transient certification and real-world application cycles. Extensive simulations in Caterpillar's DYNASTY™ software environment utilized the validation test data to assess performance more fully and confidently over varied cycles and strategies. An average fuel consumption reduction of 17.9% was realized, and the majority (~13%) of the benefit stemmed from the core concept 13L engine. To conclude, a total cost of ownership analysis provides context to commercial viability and where adoption focus should be placed.

Keywords: Engine, Diesel, Hybrid, Flywheel, Motor Generator, Transient Response, SuperTurbo, Thermal Barrier Coatings, Efficiency, Simulation, Validation, Technoeconomic

NOMENCLATURE

BEV	battery electric vehicle
BSFC	brake specific fuel consumption
CSLA	constant speed load acceptance test
CVT	continuously variable transmission
FARC	fuel-air ratio control
FEAD	front engine accessory drive
GHG	greenhouse gas
H2D2	hybrid heavy duty diesel power system
HAPTS	hybrid assist power trim strategy
HSFW	high speed flywheel
IMAP	intake manifold absolute pressure
MAF	mass air flow
MGU	motor generator unit
MMA	modular model architecture
NRTC	nonroad transient cycle
PTO	power take off
PMEP	pumping mean effective pressure
SOC	state of charge
STMEP	SuperTurbo mean effective pressure
TBC	thermal barrier coatings
TCO	total cost of ownership
TSFC	total specific fluid consumption
VEI	virtual engine installation
VGT	variable geometry turbocharger

1. INTRODUCTION

With the ever present need to provide off-road power systems that increase efficiency and power density, a hybrid concept diesel power system was developed to demonstrate the potential to achieve much higher efficiency than Caterpillar's current production diesel engines. The drive for efficiency is motivated by economics - since fuel consumption for heavy-duty applications dominates the lifecycle cost of ownership - and by greenhouse gas (GHG) reductions toward the sustainability goals of heavy-duty customers and manufacturers. A program was structured around Caterpillar's industrial interests in alignment with the U.S. Department of Energy's funding opportunity announcement DE-FOA-0001919. The high-level goals of the program were to develop an off-road hybrid diesel power system which could achieve the following:

- 17 (±2) % more fuel efficient than current Tier 4 diesel engine
- Equivalent transient response vs. baseline diesel engine
- Tier 4-Final exhaust emissions levels

If the stated goals could be achieved and implemented into Caterpillar's product range, over twenty-five million barrels of petroleum would be saved in a ten year period. And equally important, critical improvements in total cost of ownership (TCO) and sustainability could be delivered to off-road machine and power system customers.

The result of the R&D effort was the design of a high-efficiency power system with a 13L concept engine – downsized from 18L – for use in off-road machine applications. The 18L to 13L downsizing was selected due to a wide application opportunity and for understanding of how hybridization may help with this difficult ~30% downsizing challenge. For example, maintenance of power system transient response is critical in off-road applications, as it is a key driver of machine productivity. The power system will have hybrid elements and a front-end accessory drive (FEAD) that incorporates:

- High-Speed Flywheel (HSFW)
- Mechanical-drive Turbocharger (SuperTurbo)
- Motor-Generator Unit (MGU)

The hybrid power system concept configuration seeks to capture the fuel consumption reduction potential of engine downsizing, higher peak cylinder pressure capability, energy recovery, anti-idle and real-time performance optimization via advanced controls. Additionally, the SuperTurbo device, conceptually, has the potential of contributing to both overall fuel efficiency and transient response improvement. However, to quantify the benefits of the various efficiency and productivity building blocks, over varied off-road machine cycles, a significant amount of 1D system simulation was performed. Some of this simulation, and the initial hybrid concept work was previously reported [1], and this core concept proceeded mainly unchanged through the build, implementation and testing validation phases. Further refinement of the system simulation was performed as engine-only and hybrid-only subsystem test data became available. This work is described in Section 2.

The availability of three unique hybrid devices (HSFW, MGU and SuperTurbo), all of which are able to either assist the engine during transient events or recover otherwise wasted energy, enable multiple different arrangements to be made depending on how these devices are connected to each other and/or to the engine (Figure 1). These arrangements, or concepts, can result in different levels of overall power system performance, and one of the main tasks for the team during the concept definition phase of the project was to evaluate the feasibility and performance implications of the different hybrid power system concepts over a range of relevant machine application cycles [1]. The assessment was done for three key applications, namely: 745AT (articulated truck), 988K/XE (wheel loader), and 390F (excavator).

Ultimately, Concept 2a (Figure 1) was selected based on both performance and design considerations. Concept 2a incorporates:

- E-Drive HSFW connected directly to the engine crank
- 48V battery and MGU connected directly to the engine crank
- SuperTurbo connected directly to the engine crank via a mechanical CVT transmission. Due to packaging concerns and multiple other connections with the front of the engine, the SuperTurbo connection was moved to the rear PTO of the engine instead of the FEAD

All the power system simulation models, simulation results and test results presented and discussed in this paper will be based on Concept 2a unless otherwise specified.

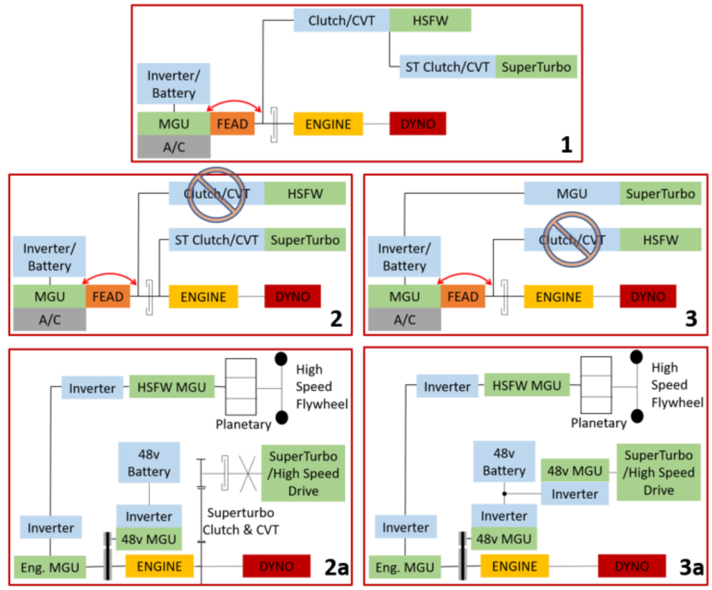


FIGURE 1: HYBRID POWER SYSTEM CONCEPTS-OVERVIEW

2. SYSTEM MODELING

Caterpillar has significant experience in evaluating the efficiency and productivity benefits associated with various levels of hybridization. Due to the extremely wide range of off-road machine applications, duty cycles and hardware configurations, 1D simulation tools are used extensively in the development process – from concept definition all the way to prototype and production performance validation.

2.1 DYNASTY H2D2 System Level Model Overview

The H2D2 system level model (Figure 2) is one of the main performance assessment and development tools employed by the H2D2 team throughout the duration of the project.

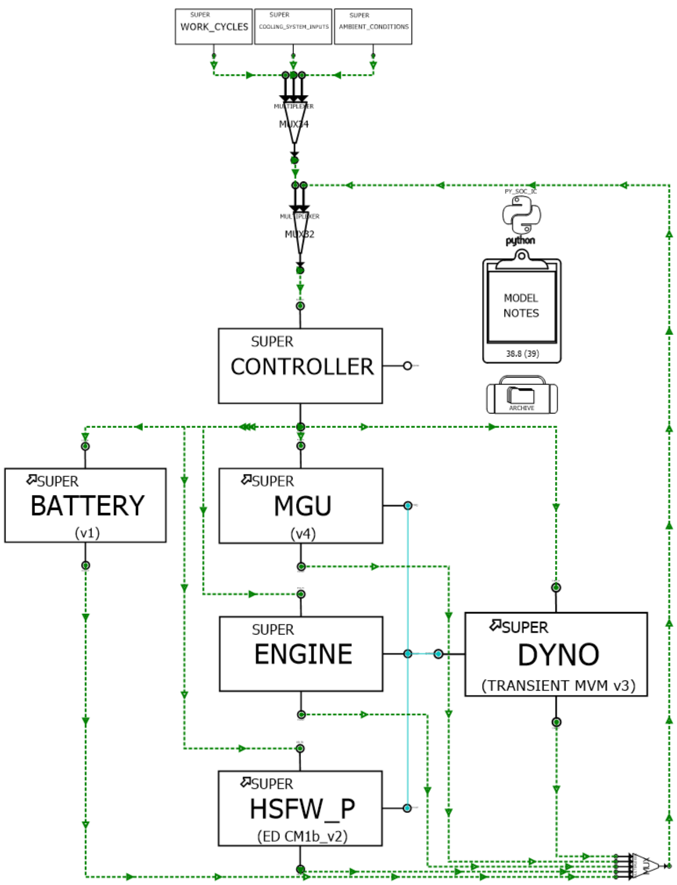


FIGURE 2: H2D2 SYSTEM LEVEL MODEL – TOP LEVEL VIEW

The H2D2 system-level model employs a Modular Model Architecture (MMA) which enables the simulation of multiple power system hardware configurations with varying levels of fidelity. The MMA provides the necessary flexibility and robustness to meet the evolving project requirements using a single 1D simulation model. Some of the main model features are summarized below:

- Transient and steady-state capability.
- Capability to simulate a wide range of transient cycles.
- Ability to simulate the CAT® C18 baseline power system.
- Ability to simulate various 13L hardware and hybrid configurations.
- L4 (sub-system level) and L5 (power system level) controls fully integrated with the plant models coupled with the ability to utilize both compiled code and co-simulation for controls development.

More information about the model structure and individual subsystems can be found in [1].

2.2 L5 Supervisory Controls

The L5 supervisory controller was integrated in the H2D2 system level model along with the L4 system plant models and lower-level controls. Full system base calibration, which accounts for hardware limits and operational constraints, was developed and implemented. This base calibration is responsible for generating the target signals for the intake manifold absolute pressure (IMAP) which is then sent to the L4 level SuperTurbo controller. The base calibration is also responsible for generating the target power signals for the HSFW and MGU L4 controllers. In addition, Design of Experiment (DoE) simulations were executed to explore the calibration space for the best fuel consumption (and/or system level TSFC) while meeting the transient response capability.

The L5 supervisory controller logic is responsible for limiting the HSFW and MGU target power requests. The decision about the direction (regen or assist) and magnitude is based on the states of the individual L4 subsystems, as well as the cycle demands. For example, the HSFW will assist the engine if the engine is close to being fuel-air ratio control (FARC) limited and will switch to regen mode when there is either enough recoverable energy available or when it needs to maintain its minimum speed.

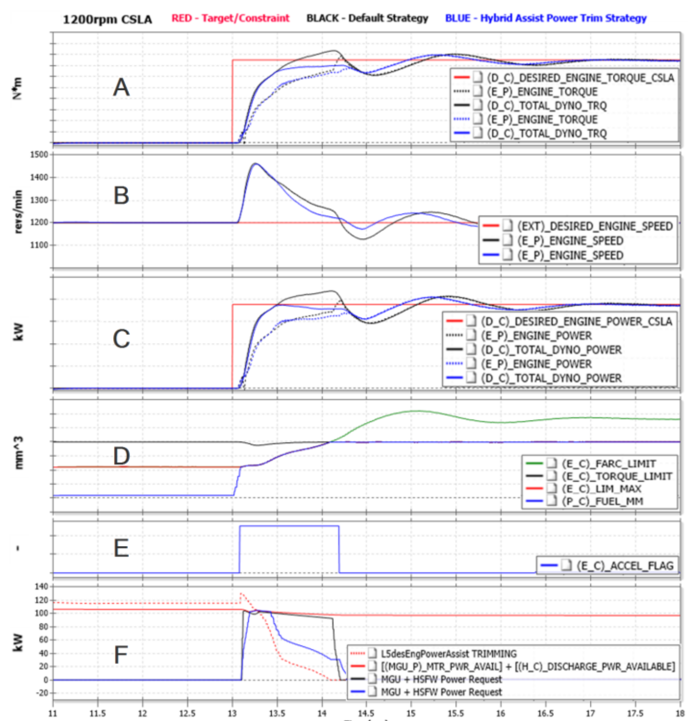


FIGURE 3: HYBRID ASSIST POWER TRIM STRATEGY - CSLA

The H2D2 hybrid power system operation over highly transient cycles with very limited opportunities for energy recovery is especially challenging. The H2D2 system level model has been used to evaluate numerous alternative control strategies which may be better suited for such highly transient operating conditions. One of the investigated and implemented strategies is the Hybrid Assist Power Trim Strategy (HAPTS). This strategy gradually reduces the assist power during the transient event so that the hybrid power system does not overshoot the target power value. Overshoots are generally undesirable as they use more energy to increase the hybrid power system power to high levels which exceed the needs of the cycle.

Figure 3 illustrates the intention behind the HAPTS during an idealized engine acceleration event over a Constant Speed Load Acceptance (CSLA) cycle.

During this cycle the target engine speed (Figure 3B) is kept constant (at 1200rpm in this example) while the target torque/power is stepped up from 0 to the maximum torque/power which the engine is capable of at the given engine speed (Figure 3A, C)). During the transient event the engine and the hybrid devices aim to increase the power system torque/power as quickly as possible until the target torque/power is reached. During this time the engine would normally be FARC limited (Figure 3D). The transient event duration is also indicated by the Accel Flag (Figure 3E) and with the **Default Strategy** the Accel Flag drives the MGU + HSFW Power Request (Figure 3F). In addition, this Power Request is only limited by the maximum assist power available from the MGU and HSFW (Figure 3F). This often results in the power system torque/power overshooting the desired torque/power towards the end of the transient event (Figure 3A, C)

In contrast, the HAPTS aims to gradually reduce the desired MGU and HSFW target Assist Power after the initial part of the transient event as seen in (Figure 3F). At the beginning of the transient event (between 13s and 13.5s - Figure 3) the assist power is kept at the maximum assist power available from the MGU and HSFW (Figure 3F) to assist the engine as much as possible until the target torque/power is reached. However, the gradual reduction of the desired MGU and HSFW target Assist Power results in the elimination of the power system torque/power overshoot towards the end of the transient event (Figure 3A, C) In conclusion – both strategies enable the target power to be reached at the same time (13.5s, Figure 3) which in turn indicates that the transient performance should remain the same while the overall power system efficiency will be increased as less hybrid assist energy will be required for each transient event.

2.3 Subsystem Validation – HSFW Plant Model

The HSFW plant and controller models have been under continuous development throughout the project. However, to ensure the HSFW plant model performance is still representative of the HSFW hardware, an additional validation exercise was performed when test data became available from the full H2D2 hybrid power system testing.

Figure 4 shows the results from the HSFW validation exercise using the final H2D2 hybrid power system test data set:

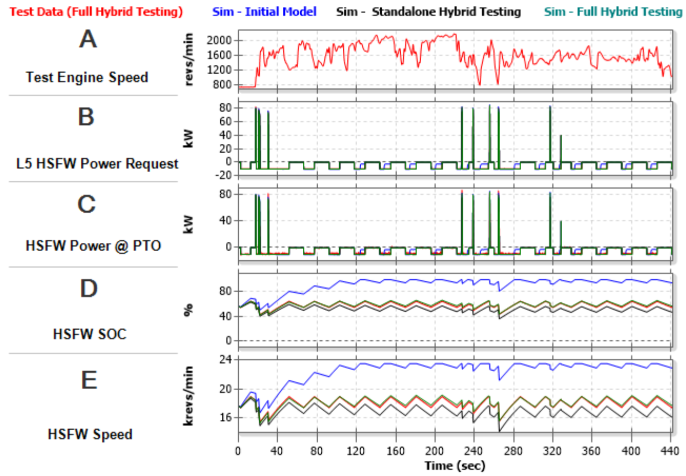


FIGURE 4: HSFW PLANT MODEL VALIDATION

To isolate the HSFW performance from the rest of the H2D2 power system, a boundary diagram was established around the HSFW plant and controller. This was achieved by using the test engine speed (Figure 4A) as a boundary condition at the HSFW PTO and the test L5 HSFW Power Request as an input to the HSFW L4 controller (Figure 4B). The HSFW power at the PTO (Figure 4C), HSFW SOC (Figure 4D) and HSFW speed (Figure 4E) were then used as the performance parameters on which the model vs. test match quality can be assessed.

As seen from the results in Figure 4, the initial model (based on supplier-provided performance and efficiency data) predicted much higher HSFW SOC (Figure 4D) and HSFW Speed (Figure 4E), which indicates lower losses compared to the test data. The HSFW model validated against standalone hybrid testing test data showed much improved performance compared to the initial model and was able to get much closer to the test data. However, the HSFW losses set in this model revision were a bit higher than necessary for achieving a good match with the test data and as a result – the predicted HSFW SOC (Figure 4D) and HSFW Speed (Figure 4E) were lower than their respective test values. Finally, the HSFW losses in the HSFW plant model were reduced slightly compared to the HSFW model validated against standalone hybrid test data, which resulted in a good match with the test data for the HSFW SOC (Figure 4D) and HSFW Speed (Figure 4E). The HSFW model validated against full hybrid test data set was used in the H2D2 system level model when generating the simulation results used for the final H2D2 power system validation.

2.4 H2D2 System Level Model Validation

The H2D2 system level model was validated using two sets of validation data: engine-only and full hybrid system.

2.4.1 Engine-Only Validation

The first validation exercise, also referred to as engine-only, was conducted using the Engine, SuperTurbo and Aftertreatment (no HSFW or MGU) and was aimed at evaluating the performance of the 13L engine, SuperTurbo and aftertreatment systems over a range of transient application cycles.

Figure 5 shows a comparison between the H2D2 system level model predictions and test data over the 745AT NRTC cycle:

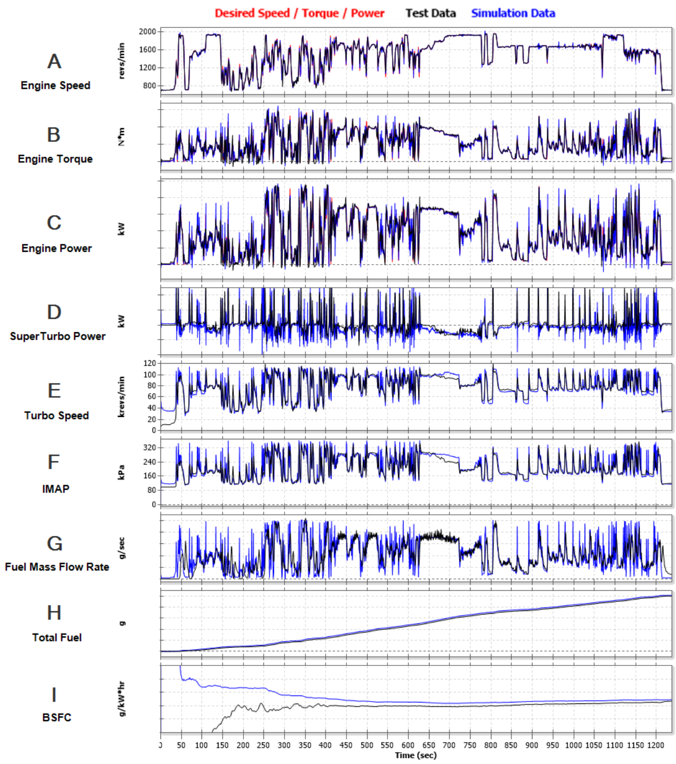


FIGURE 5: ENGINE-ONLY VALIDATION – 745AT NRTC

In general, the model demonstrates a very good correlation with the test data throughout the duration of the cycle. The Engine Speed (Figure 5A), Engine Torque (Figure 5B) and Engine Power (Figure 5C) all match well, which indicates that both the H2D2 system level model and the Test Cell Engine are tracking the desired cycle well. The Turbo Speed (Figure 5E) and IMAP (Figure 5F) are also in a good agreement throughout most of the cycle. The two exceptions are the idle period at the beginning of the cycle (0-30s) and a short section between 670s and 730s which are likely due to slight differences in the advanced controls map lookup resulting in slightly higher IMAP targets for the H2D2 system level model during parts of the cycle.

Finally, when it comes to the total fuel (Figure 5H) and Cycle BSFC (Figure 5I), the model is able to predict the performance of the test engine within 1% difference.

2.4.2 Full Hybrid System Validation

The full hybrid system validation exercise reflects Concept 2a (Figure 1) and includes the HSFW and MGU. It was aimed at evaluating the performance of the full hybrid system over a range of transient application cycles. However, due to differences between the desired L5 strategy used in simulation and the L5 strategy used during testing, no direct comparison between the two could be made. In the test data, the L5 strategy was not optimal, and did not switch to regen mode when it was expected to do so, therefore it did not take full advantage of any recoverable energy during some of the transient application cycles.

To overcome the issue and validate the H2D2 system level model against full hybrid system test data, it was decided to use the H2D2 system level model to replicate the sub-optimal L5 strategy used during testing. This was achieved by taking the HSFW and MGU L5 commands from the test data and using them directly in simulation, thus ensuring consistency in how the hybrid system operates.

Figure 6 shows a comparison between the H2D2 system level model predictions and test data over the 745AT NRTC cycle.

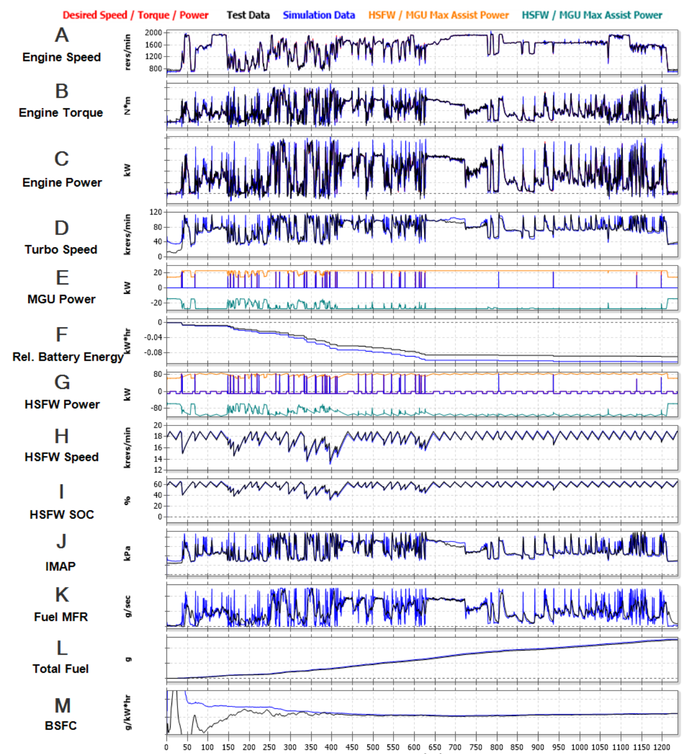


FIGURE 6: FULL HYBRID SYSTEM VALIDATION – 745AT NRTC

Just as with the engine-only validation, the Engine Speed (Figure 6A), Engine Torque (Figure 6B) and Engine Power (Figure 6C) all match well, which indicates that both the H2D2 system level model and the test cell engine are tracking the desired cycle well. The Turbo Speed (Figure 6D) and IMAP (Figure 6J) are also in a good agreement with the same few exceptions observed in the engine-only validation.

When it comes to the hybrid device performance, the MGU Power (Figure 6E) is only being used for assisting during transient events due to the sub-optimal L5 strategy as explained earlier. As a result, the Relative Battery Energy (Figure 6F) is negative throughout the duration of the cycle. For this parameter the H2D2 system level model predicts slightly higher energy used over the duration of the cycle, however, that difference is only about 0.014kWh, which compared to the total cycle positive work of about 50kWh, is a negligeable difference.

The HSFW Power (Figure 6G) is also being used for assisting during transient events (at a much higher power), but unlike the MGU, the HSFW SOC (Figure 6I) is being kept at around 60% by employing relatively frequent micro-charge events. This enables the HSFW to always have enough charge to assist the engine with sufficiently high power, but also keep the HSFW spin down losses down to more manageable levels. In addition, as a result of the HSFW plant validation exercise described in section 2.3 of this paper, the HSFW Speed (Figure 6H) is matching very well between test and simulation, indicating the HSFW plant model provides accurate prediction for the HSFW spin down losses.

Finally, when it comes to the Total Fuel (Figure 6L) and Cycle BSFC (Figure 6M), the model is able to predict the performance of the test engine very well (~ 0.13 % difference). The tabulated validation results for all application cycles can be found in Figure 7 with the full hybrid system correlation between test and simulation summarized in row E.

Despite the reduced number of transient cycles used for the full hybrid system validation, the correlation between the H2D2 system level model and the test data is consistently good. This gives the team confidence that the H2D2 Hybrid power system efficiency benefits vs. the C18 Baseline, predicted using the validated H2D2 system level model (Figure 7B), accurately and objectively represent the performance advantage of the H2D2 power system.

2.4.3 Validation and BSFC Benefit Summary

The transient H2D2 power system validation results from the engine-only and the full hybrid system validation exercises, along with the predicted BSFC benefit vs. the C18 baseline, are summarized in Figure 7:

Figure 7A Shows the % BSFC benefit of the H2D2 power system (Engine + SuperTurbo + Aftertreatment) vs. the C18 Baseline.

Figure 7B Shows the % BSFC benefit of the H2D2 power system (full hybrid system) vs. the C18 Baseline.

Note that the Optimal L5 strategy, which takes advantage of recoverable energy during some of the application cycles, was used by the H2D2 system level model to generate this data.

Figure 7C Shows the % BSFC associated with the addition of the HSFW and MGU (7C = 7B - 7A).

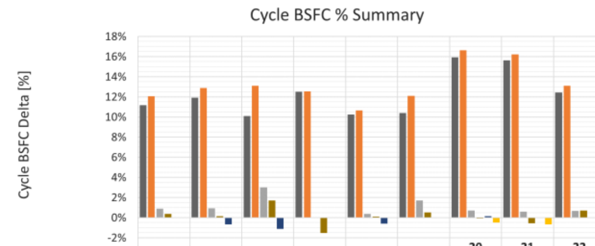
The additional benefit from the hybrid devices (HSFW & MGU) is higher for cycles which have more significant recoverable energy available.

Figure 7D H2D2 system level model (Engine + Aftertreatment) validation results vs. H2D2 Test data.

Figure 7E H2D2 system level model (full hybrid system) validation results vs. H2D2 Test data.

Please, note that sub-optimal L5 strategy, which does not take advantage of recoverable energy, was used to generate this data.

Figure 7F H2D2 system level model (C18 Baseline configuration) validation results vs. C18 Test data.



		01 745	05 745	06 745	10 390X	11 988K	19 988XE	20 NRTC 745AT	21 NRTC 390F	22 NRTC 988K XE
A	C18 SIM vs H2D2 SIM (ENG+AT)	11.16%	11.93%	10.10%	12.52%	10.25%	10.41%	15.94%	15.64%	12.43%
B	C18 SIM vs H2D2 SIM (HYBRID)	12.05%	12.86%	13.11%	12.53%	10.64%	12.10%	16.63%	16.22%	13.10%
C	HYBRID BENEFIT	0.89%	0.93%	3.01%	0.01%	0.39%	1.69%	0.69%	0.58%	0.67%
D	H2D2 TEST vs SIM (ENG+AT)	0.36%	0.15%	1.71%	-1.52%	0.10%	0.52%	-0.07%	-0.55%	0.71%
E	H2D2 TEST vs SIM (HYBRID*)		-0.68%	-1.10%		-0.58%		0.13%		
F	C18 TEST vs SIM							-0.49%	-0.65%	0.04%

FIGURE 7: VALIDATION AND BSFC SUMMARY TABLE

3. PROCURMENT AND SYSTEM BUILD

3.1 Testbed Design Overview

Test cell facility virtual installations were completed for the hybrid system only validation testing phase as well as the concept engine-only and full system level installation. Several benefits were derived from completing a virtual installation before a real-life build. Proper planning and layout contribute to a successful installation along with understanding system requirements such as cooling, electrical, exhaust, and mounting systems to reduce vibration.

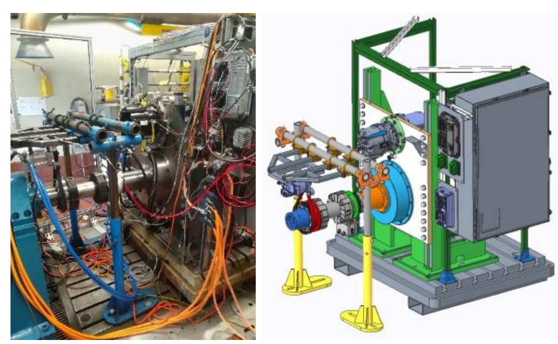


FIGURE 8: H2D2 HYBRID SYSTEM VALIDATION TEST INSTALLATION AND CAD COMPARISON

The non-engine shakedown testing effort planned before the engine-connected testing was to debug and validate the hybrid performance and controls. The partial build and shakedown were completed and included everything except the core engine and the SuperTurbo: HSFW, HSFW electric drive MGU's and invertors, 48V MGU and battery emulator.

The bedplate with hybrid components was modified to work with adapter tubing to secure the floating bedplate to the main test cell bedplate, reducing vibration and allowing for shaft component alignment between the front engine damper and high-speed flywheel system as shown in Figure 9.

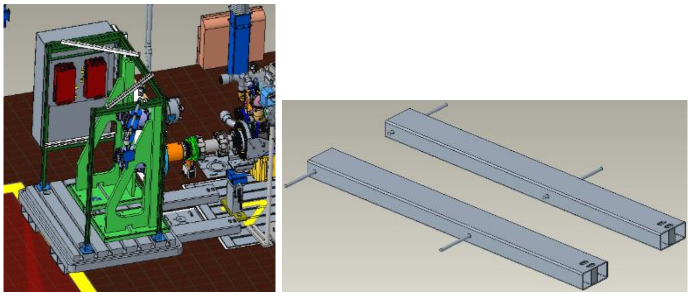


FIGURE 9: HYBRID COMPONENTS FLOATING BEDPLATE FIXTURED TO MAIN BEDPLATE USING ADAPTER TUBING

The front-mounted encoder setup that determines the crank angle was not feasible as the hybrid system had a driveshaft running off the front of the crankshaft. A rear encoder was added between the dyno and the engine to solve this issue.

The complete Virtual Engine Installation (VEI) with exhaust system, piping, filters, CEM, oil pumps, hybrid HSFW, cooling system, and guarding in place is shown in Figure 10. This final design proved difficult as the test cell has limited space when measurement instrumentation is present, and there has not been a floating bedplate with the heavy mass outside of the normal test bedplate tested before. Vibration was damped with rubber mats underneath the adapter plates and floating bedplate. Limited space did cause issues with maintenance and maneuverability in the cell. Overall, the design was robust and led to a successful installation.

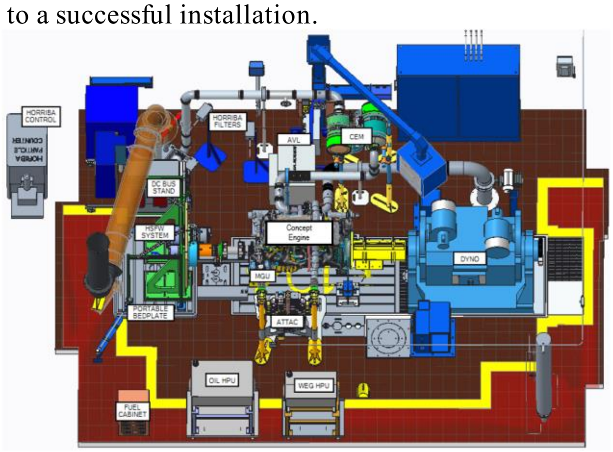


FIGURE 10: LAYOUT OF FULL H2D2 SYSTEM TEST CELL

Pictures of the final full H2D2 hybrid system test installation are presented in Figure 11 and document the complexity and successful capability of Caterpillar's power system facilities, resources, and personnel. The installation included the 13L concept engine, SuperTurbo, emissions aftertreatment catalysts, 48V MGU and battery emulator, HSFW electric drive, hybrid oil and coolant supply units, and numerous testing support systems. Functional system operation and data acquisition was achieved, and a high degree of coordination had to be implemented between the hardware, facility devices, and emissions measurement systems.

FIGURE 11: FINAL H2D2 FULL HYBRID SYSTEM VALIDATION TEST INSTALLATION

3.2 Hybrid Flywheel System

For the validation testbed, a flywheel module was designed by a supplier per requirements with an all-steel rotor and low-cost containment system while still meeting life targets, Figure 12. As part of the design process, analysis was done to evaluate the effect of a vacuum system on the performance of the module. The CFD results showed a significant drop (+50% lower) in aerodynamic drag between atmospheric pressure and partial vacuum configurations. To create partial vacuum, vacuum seals were added in the design. The partial vacuum system was further analyzed using several off-highway machine duty cycles to understand the effect of the flywheel parasitic losses on the system performance. The results showed that running the flywheel in partial vacuum significantly improves the system performance in various applications.

Parameter	Value
Diameter	210 mm
Width	121.5 mm
Inertia	0.177 kg.m^2
Mass	33.1 kg
Speed	23,500 rpm
Rated Energy	537 kJ

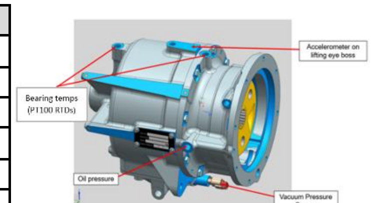


FIGURE 12: HIGH SPEED FLYWHEEL (HSFW) KEY PARAMETERS

The final design selected for the high-speed flywheel drive was an electrical CVT with an MGU connected to the engine and an MGU connected to the high-speed flywheel with power transferred through a common DC bus. The e-CVT advantages included fast response time and insensitivity to engine transients due to engine/HSFW mechanical decoupling. Power system sizing simulation and analysis resulted in requirements specifying a system that had +100kW of maximum power with

+500kJ energy storage capacity for as much energy recovery as possible from applications with heavy engine braking (745AT). The final hardware selection and configuration was made along with detailed component design, engine and test cell facility integration design, and torsional vibration analysis. The HSFW was connected to the MGU with a planetary gearset and adapter shaft and housing, Figure 13 (Top). Power was transferred from this MGU to the engine-connected MGU though 2 inverters and common DC bus. The engine-connected MGU connected to the engine through another planetary gearset, torque meter, and driveshaft assembly, Figure 13 (Bottom).

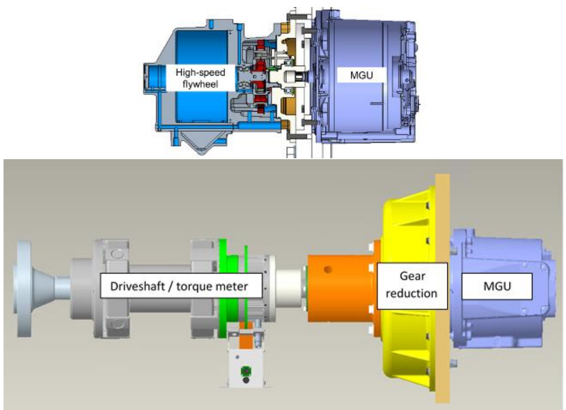


FIGURE 13: TOP – HSFW-MGU MODULE BOTTOM – HSFW-ENGINE FRONT DRIVELINE

Durability analysis was performed on the high-speed flywheel drive system using histograms created from simulated machine cycles. The histograms were input into Romax analysis software and showed bearing life and stress to be acceptable to meet the 12,000-hour target. The histograms were also used to evaluate gear life in Caterpillar's gear analysis software. Gear life and stress were also found to be acceptable. Durability analysis and testing was also done by the high-speed flywheel supplier for the flywheel module itself. Extensive fatigue analysis was done to verify the target of 600,000 full charge and discharge cycles could be met. Bench tests were also done on a similar design to validate the inputs of the analysis including material testing and stress validation. Bearing life calculations were performed by the bearing supplier and predicted 20,000 + hours of life based on a generic off-road duty cycle and vibration levels.

3.3 48V MGU System

A 48V MGU supplier and model were identified and procured along with a compatible inverter. The 48V MGU had a continuous power rating of 15 kW and peak power of 23 kW. With data from the MGU supplier, a plant model was created for use in DYNASTY along with an initial battery model. An MGU controller was also created in Simulink and together with the plant model were integrated into the full H2D2 system model. Simulations were completed to develop the L5 and MGU controls and determine its most efficient use. For test cell integration reasons, a battery emulator was chosen instead of a battery and the battery parameters were simulated.

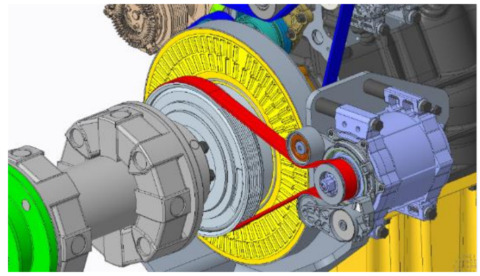


FIGURE 14: 48V MGU, BRACKET, DECOUPLER, AND TENSIONER

The 48V MGU was integrated on the front of the engine separate from the SuperTurbo, Figure 14. The design considered many factors including packaging, torsional vibration, and drive system. For easier implementation for the testbed, the MGU was integrated on a separate belt drive system whereas on a machine application the intention would be to replace the alternator and exist on the main belt drive. The available space claim and engine torsional data were provided to the belt drive supplier and a tensioner and crankshaft decoupler drive pulley for our application was designed. The belt tensioner was designed to provide correct belt tension with the MGU either motoring or generating power. The decoupler pulley was recommended to isolate the belt drive system from the torsional dynamics of the engine.

3.4 Engine and Aftertreatment

The high-level design requirements for the core engine were determined by using performance analysis so that the optimized designs could be developed to meet the performance and durability targets. The overall development path was chosen to be the stroked and upgraded 13L platform, and a path to meeting the program targets for fuel efficiency, power capability, and overall performance was laid out. This path to efficiency improvement in the engine system includes reduced friction, increased peak cylinder pressure (PCP), compression ratio change, reduced aftertreatment backpressure, improved combustion, better breathing, and improved air system. The H2D2 demonstrator engine was ultimately developed as a variant of Caterpillar's recently announced C13D engine [2]-[3] and was designed to achieve best-in-class power density, torque and fuel efficiency for optimizing the performance of heavy duty off-highway applications.

As indicated from the activities above, friction reduction is a consideration in much of the core engine design. Therefore, significant effort was put into outlining the potential friction reduction technologies and evaluating their cost-benefit trade-off so that the most effective technologies can be incorporated into the core design. The following list shows technologies that were considered to create the roadmap to the engine goal of 20% friction reduction.

- Oil thermostat for maintaining consistent oil viscosity
- Increased rod length to reduce rod angularity
- Reduced bore distortion through liner flange design and form honing
- Optimization of ring tension, height, and coating
- Optimized piston skirt profile
- Cooling jacket design to increase mid-stroke temperature and lower oil viscosity in that region
- Cooling and lube system flow optimization to minimize restrictions
- Crowned crank journals
- Bearing materials including polymer overlay
- Variable oil pump
- Clutched water pump
- Variable piston cooling jets
- Low viscosity oils
- Liner surface treatments such as thermal spray with mirror finish or honing stratification

FIGURE 15: TBC CYLINDER HEAD AND PISTONS SUCCESSFULLY MANUFACTURED

Thermal Barrier Coatings (TBC) were applied to the piston and cylinder head bottom deck, Figure 15, for the final validation testing of the engine. Throughout the project simulation and single cylinder experimentation were used to evaluate traditional ceramic thermal sprayed TBC materials, on the order of 0.1 to 0.5mm thick, for reduced in-cylinder heat transfer and efficiency improvements. The overall performance and efficiency impact was one of limited success and confounded results. Single cylinder engine indicated and brake measurements were used to compare efficiency and performance results, along with measurements of heat rejection from the oil and coolant systems. Comparisons between steel pistons and TBC coated pistons targeting the same geometry only found limited times of outright efficiency benefit. The program concluded that durable TBC components could be successfully produced for 500-1000 hour prototype testing and should be included in the full system validation for any possible heat transfer reduction benefit on turbocompounding, despite the difficulty of isolating direct in-cylinder efficiency benefits. Extended experimental and optical interrogation of the TBC intrusiveness on the combustion system can be found in the authors' recent conference publication [4].

A solenoid was installed in place of the mechanical priority valve in the main oil gallery. To accomplish the transition, machining had to be done to the block, but the threads remained the same. The solenoid allowed for full authority control of the oil flow to the piston cooling jets.

FIGURE 16: SOLENOID INSTALLED INTO ENGINE BLOCK

One of the major changes involving the design was the integration with the engine PTO. In an effort to reduce prototype cost and parts, a gearbox was internally designed to allow the use of the simpler "Industrial" PTO housing instead of the original, unnecessarily strong and longer lead-time, "Heavy Duty" PTO. The SuperTurbo was changed to be driven off the rear face of the industrial PTO instead of the front face of the Heavy Duty PTO which was originally planned. From the output of the gearbox, a stock-sized Gates pulley and belt were used to integrate with the SuperTurbo. This setup allowed for a ratio of 3.133 to the SuperTurbo and demonstrated to Caterpillar how the SuperTurbo drive may be highly configurable to different PTO arrangements through gearbox and pulley drive solutions.

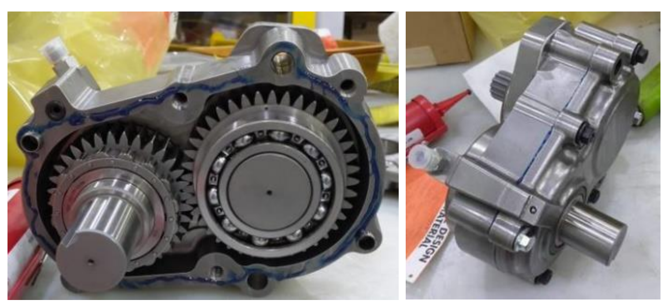


FIGURE 17: ASSEMBLED PTO GEARBOX

The SuperTurbo mounting scheme consisted of bolting a bracket group to the flywheel housing and two locations on the engine block. Each bracket supported a different section including the input shaft, clutch housing, and main housing.

After the drive configuration was finalized, brackets were designed and modified through several FEA simulations to reach the highest vibration levels within the time constraints. The mounting system for the SuperTurbo resulted in a lowest natural frequency of 228 Hz with the target greater than or equal to two times the maximum engine firing frequency, or 240 Hz.

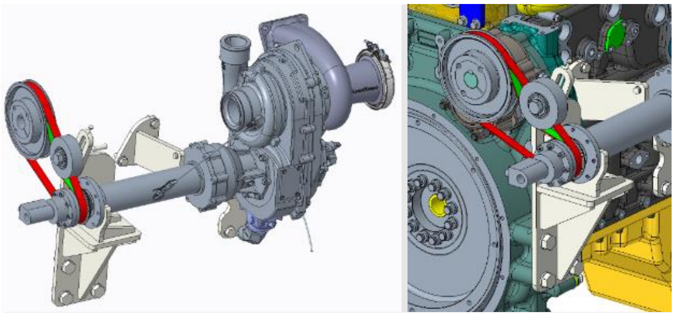


FIGURE 18: SUPERTURBO DESIGN INTEGRATION TO PTO

The aftertreatment system consisted of a switchback-flow design with a six-inch mixing tube and 13”x12” diesel oxidation catalyst filter (DOCF) on a hexagonal-cell substrate. The SCR and ammonia slip catalysts were split between two bricks: A 13”x5” copper zeolite SCR on a 400 cells per square inch (cpsi) substrate, and 13”x5” dual-zoned copper zeolite SCR and ammonia slip catalyst on a 400 cpsi substrate. Substrate geometries, mixing tube diameter, and overall system design were chosen based on simulation results to minimize backpressure for best efficiency with turbocompounding on the engine while still meeting Tier 4F emissions regulations.

3.5 SuperTurbo

A SuperTurbo is a mechanically driven turbocharger capable of providing precisely regulated intake manifold absolute pressure (IMAP) or intake mass air flow (MAF). For the H2D2 system it was the role of the SuperTurbo to 1) provide accurate IMAP regulation, 2) rapidly increase the IMAP during fast engine transients, and 3) recover excess exhaust energy at higher load conditions via turbocompounding.

The H2D2 SuperTurbo design was based on the generic SuperTurbo architecture capable of supporting commercial engine displacements ranging from 7L to 15L. Generally, when adapting a SuperTurbo for a given application, 90% of the parts are common with only the turbine, compressor, and mounting options changed based upon individual engine requirements [5]. A SuperTurbo can transmit power from engine-to-turbo (supercharging) or return power from turbo-to-engine (turbo compounding) as it maintains the necessary compressor speed to achieve the target IMAP. Doing this on engines with displacements of up to 15L requires that the mechanical linkage between the turbomachinery and the engine crank transmit 30kW of power continuously and up to 50kW of power intermittently. The continuous and intermittent power requirements are satisfied both during supercharging and turbo compounding. However, it is typical for the 30-kW continuous operation to occur during turbo compounding as the engine operates at a high-power output. The 50kW intermittent power typically occurs while supercharging to rapidly accelerate the compressor speed during aggressive transients.

A SuperTurbo can integrate into an engine platform mechanically at either the back of the engine through the PTO or at the front of the engine through the FEAD. For the H2D2 application, the SuperTurbo operated through the engine PTO. Directly after the mechanical integration point there is a controlled clutch. The clutch protects the turbo from excessive input torque during normal engine start and from engine speed reversals, i.e., the engine rocking, when the engine stops. In general, a clutch can either be electro-mechanically or pneumatically activated. Here, the clutch was the Logan R350 pneumatic clutch.

The core driveline components of the SuperTurbo that are common among all SuperTurbo designs reside after the driveshaft and clutch. They are cartridgelike subassemblies installed within a common housing. The first cartridge is the continuously variable transmission (CVT). The SuperTurbo CVT is a NuVinci Continuously Variable Transmission which uses a tilted axis ball configuration with traction drive input and output contacts along the orbit of the spinning balls. A small amount of force is applied via an electric actuator which tilts the balls in different directions, thus changing the relative position of the input and output races as they relate to the center of rotation [6]. This varies respective input and output speeds, thus providing overall controllability between the engine speed and the turbomachinery speed thereby enabling precise IMAP or MAF control.

The second cartridge is the high speed planetary, also referred to as the high-speed section (HSS) of the SuperTurbo. It is situated on the compressor shaft between the compressor and turbine. This planetary drive utilizes traction rollers that capture and position the turbo shaft both axially and radially. Thus, there is no requirement for fluid, ball, or thrust bearings on the main turbo shaft [5]. Such a traction drive system is necessary, instead of a gear-based system, due to the high surface velocity at the interface of the compressor shaft which can spin more than 100,000 RPM. Figure 19 illustrates the internal components of a SuperTurbo system.

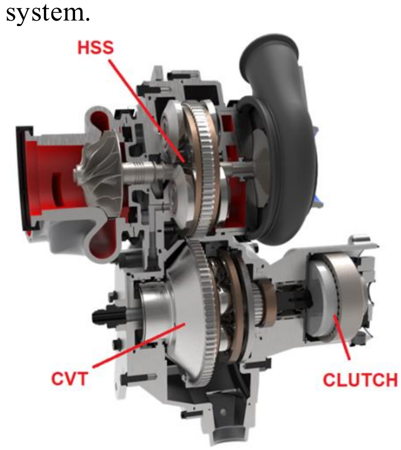


FIGURE 19: SUPERTURBO INTERNAL COMPONENT VIEW

The turbo machinery of the SuperTurbo for the H2D2 application consists of an off-the-shelf compressor used on Caterpillar applications (Figure 20) and a custom designed turbine (Figure 21). Using an off-the-shelf compressor design provided a “known quantity” starting point to the customization of the SuperTurbo, i.e. compressor map width, efficiency and performance were known to satisfy the requirements of the

H2D2 platform. For the turbine design, the ability of a SuperTurbo to return excess turbine power to the engine crankshaft instead of over-speeding the turbomachinery allows for a higher efficiency turbine design than is usable in non-driven turbochargers (both fixed and variable geometries). Therefore, a primary turbine design goal becomes the maximization of the turbine power output and the H2D2 custom turbine yielded a peak efficiency greater than 84%.

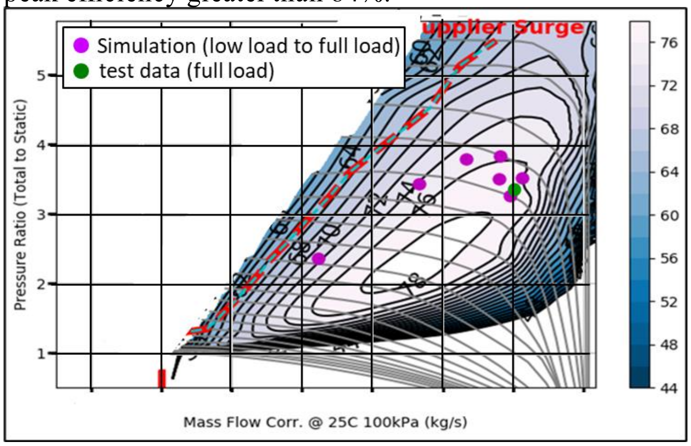


FIGURE 20: SUPERTURBO PERFORMANCE ON COMPRESSOR MAP

FIGURE 21: CUSTOM TURBINE DESIGN FOR SUPERTURBO

4. TEST RESULTS – ENGINE-ONLY

Engine-only (core 13L technology + SuperTurbo + Aftertreatment) testing of steady-state operation points occurred over the entire speed load region, which encompassed the specific lug curves of the three target machine applications. Example points and the composite maximum lug curve are shown in Figure 22 and highlight the high overlap between simulation and test point locations. This steady-state mapping provided useful performance validation data and enabled standard emissions certification automation setup for transient cycle data acquisition, such as the Nonroad Transient Cycle (NRTC) [7].

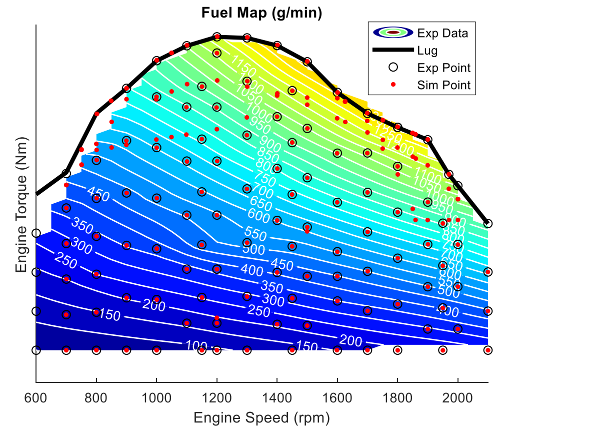


FIGURE 22: LUG CURVE ENVELOPE, EXPERIMENTAL AND SIMULATION DATA POINTS FOR STEADY STATE VALIDATION

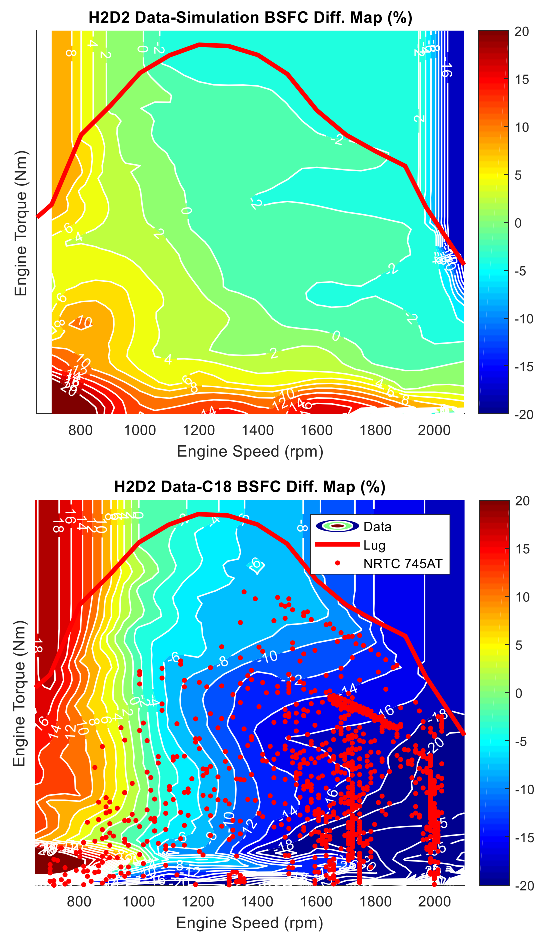


FIGURE 23: TOP – H2D2 DATA TO DYNASTY SIMULATION PERCENT DIFFERENCE IN BSFC. BOTTOM – H2D2 DATA TO ORIGINAL BASELINE C18 VALIDATED SIMULATION PERCENT DIFFERENCE IN BSFC.

To summarize the total combined efficiency assessment of the engine-only system, which includes all of the core engine, turbocompounding, and aftertreatment subsystems, percent difference comparisons to the reference C18 were made, Figure 23. The top contour plot shows the difference between the final validation test data and the DYNASTY Concept 13L engine-only simulation prediction prior to final correlation. There is a large region of ±2 difference, with the test data being more efficient (negative % diff) in the critical higher power and load regions. The areas of worst test efficiency and higher percent difference were found to be the very low load where the simulation typically struggles, and is attributed to differences in calibration, accuracy of combustion prediction, and accuracy of losses such as the SuperTurbo drive. The right contour plot illustrates where the total percent improvement in efficiency or BSFC (negative) predominantly was found to occur, and an overplot of the 745AT NRTC cycle highlights strong alignment. The correlated and validated C18 simulation was used for the reference and is consistent with how the reporting was conducted throughout the project. Substantial 10-20% improvement ranges were validated from this testing, and final cycle results will be discussed in the following sections.

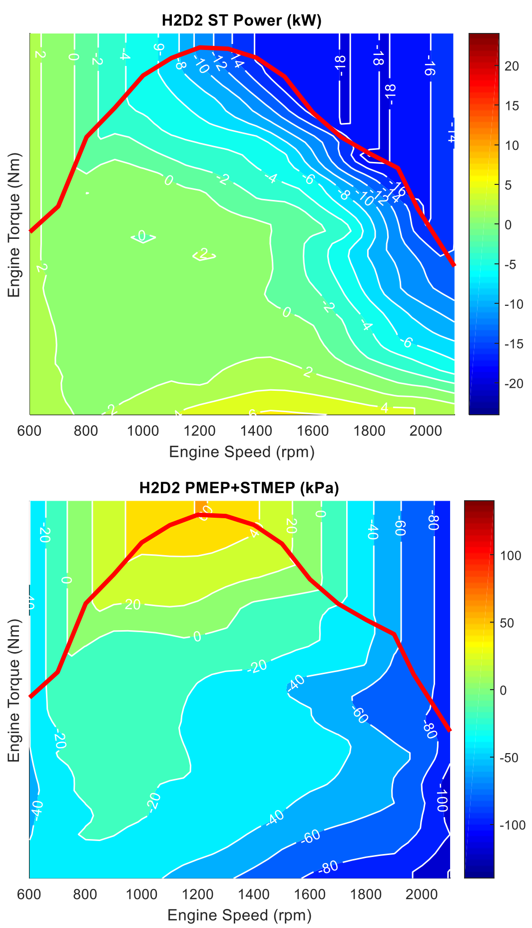


FIGURE 24: FINAL SUPERTURBO POWER AND COMBINED AIR SYSTEM MEP

The full operation region calibration and resulting behavior of the combined air system is presented in the contour plots of Figure 24. The measured net SuperTurbo power at the PTO interface connection shows more negative (energy recovery via turbo compounding) power in the higher load and power regions toward the lug curve. Continuous levels peaking around 20kW of net compounding were achieved with the nominal design point of the 1.05 factor turbine-flow nozzle utilizing the reduced backpressure aftertreatment system. Also, higher continuous levels were observed throughout testing with the other more restrictive nozzles, but the 1.05 nozzle was confirmed to be optimal for totally system efficiency. The combined PMEP+STMEP contour plot shows the resulting air system net gain or loss, and the region of positive benefit can be seen to be around peak torque and power. This combined MEP view is the result of the engine combustion, emissions, EGR, and air system optimization that was conducted in the simulation concept definition phase. Balancing PMEP, STMEP, EGR driving capability, combustion phasing and NOx for lowest fuel and DEF consumption was critical, as these all highly interact. For example, it was not always clear if maximizing compounding power should have been prioritized at the expense of PMEP, as there is no dedicated second compounding turbine for additional energy recovery. The validation testing confirmed good accuracy of the simulation and calibration modeling and sensitivities.

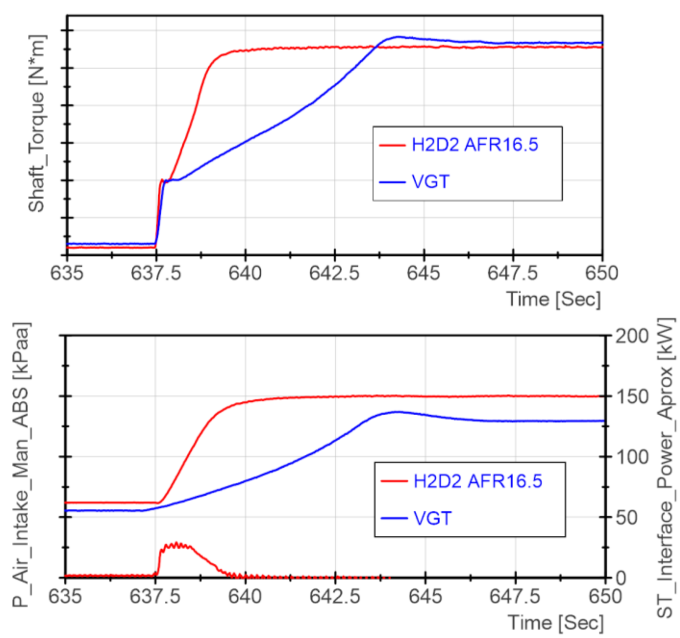


FIGURE 25: 1000RPM CSLA COMPARISON BETWEEN THE FINAL H2D2 CALIBRATION AND A VGT DATASET

Transient load response was measured through multiple tests and had an initial level of engine fuel and emissions calibration effort applied. Constant Speed Load Acceptance (CSLA) tests were conducted from low to high speed, with air-fuel ratio based fuel-limiting ranges, and with comparisons to other engine data with a variable geometry turbocharger (VGT).

A timeseries data example of the transient load response is plotted in Figure 25 and draws a drastic contrast between the H2D2 engine with the SuperTurbo and the VGT. Near 30kW of supercharging power was applied immediately after the snap torque and the IMAP rise was almost four times as fast as the VGT. Subsequently the torque rise mimics the IMAP rise improvement, and a more complete comparison of time to 85% torque over different engine speeds in Figure 26 clearly shows the benefit of the H2D2 engine at lower engine speeds. It should be noted that further improvement was possible (fueling was limited by other controls and not airflow/IMAP) at 1600 and 1800 rpm but not pursued due to time constraints. The ability to achieve superior transient load response without a sacrifice in efficiency is a unique and noteworthy capability of the developed engine and air system.

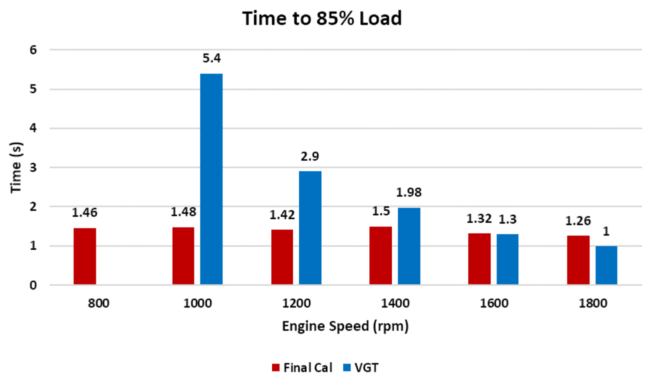


FIGURE 26: COMPARISON OF CSLA METRICS BETWEEN THE FINAL H2D2 CALIBRATION AND A VGT DATASET

A final aspect of the engine-only testing was validation of Tier 4 Final off-road regulated engine emissions. Certification-level testing was completed using the 745AT lug curve with cold and hot NRTC emissions cycles. The system passed Tier 4 Final emissions (0.17 g/kWh CO, 0.008 g/kWh HC, 0.349 g/kWh NOx, 0.01 g/kWh PM) with good engineering margin for a de-greened system at this stage of prototype development. Six key machine cycles and the three application NRTC cycles were evaluated for final fuel consumption validation, as shown earlier in Figure 7, and Table 1 in the later Final Efficiency Summary section documents engine-only improvement range (10.3-16.1%).

5. TEST RESULTS – FULL HYBRID

As discussed previously in section 2.4.2 of this paper, only a suboptimal L5 hybrid assisting strategy (best load response but with an expected efficiency loss) was able to be tested and conditions where engine braking occurred were not evaluated in test. This untested hybrid energy recovery impact on the system efficiency was quantified and captured with the highly correlated H2D2 system level model. Despite this testing shortcoming the project was able to acquire a large quantity of valuable validation data and system experience.

The BSFC percent difference to the engine-only cycle test results of 0.7% worse for NRTC and 2.4% worse for machine cycles indicate that the full hybrid system running only in the assisting mode is not more efficient. As discussed previously, this is expected as, if no braking energy is recovered, the additional hybrid drive losses and HSFW charging will act solely as an increase in parasitic loss. The main benefit of an assist-only strategy would be relative to machine productivity as opposed to power system efficiency.

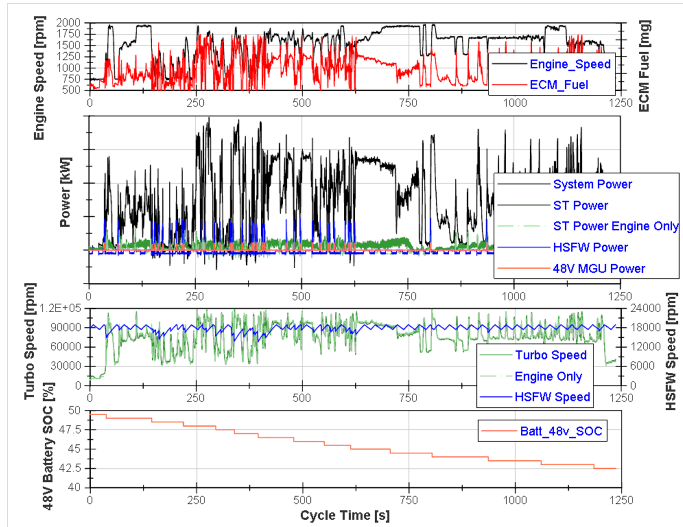


FIGURE 27: TRANSIENT PLOT OF HYBRID POWER FROM 745AT NRTC. ENGINE-ONLY DATA OVERLAYED AS DASH-DOT FOR COMPARISON.

The representative behavior of the full hybrid system when running over the transient cycles in the assisting mode are plotted in Figure 27 and Figure 28. The cycle is fairly transient with many accelerations and changes in load/power. The hybrid device (SuperTurbo, HSFW, 48V MGU) power can be seen to be active throughout the cycle with times of higher frequency associated with times of more frequent transient power. In the assisting only mode, the hybrid device power is mostly positive on the graph, which indicates power flowing in the assisting direction. The HSFW (blue) line plot dips negative regularly and indicates time when the HSFW is recharging (speeding up) by using engine power. The HSFW speed behavior is seen in the third from the top plot, and it generally is a saw tooth behavior centered at ~18 krpm (~60% SOC). This 18 krpm was set from the control strategy to enable high HSFW power availability for more powerful assists. Finally, the 48V battery state of charge (SOC) can be seen to only decrease throughout the cycle in the bottom plot due to the assist only mode and no attempt to recover engine braking energy (of which there is none in the NRTC). It should be noted that the SOC data in Figure 27 is using an emulated 2 kWh battery, while other full DYNASTY simulations used a 6 kWh battery. Also, the SuperTurbo torque transducer was failing or experiencing electrical interference during the hybrid runs and is the reason for times of nonphysical readings of high positive assisting power where there was turbocompounding negative power during the engine-only runs (e.g., time 625 to 750 seconds).

In Figure 28 the comparison to the engine-only NRTC cycle data can be compared by referencing the dash-dot lines, and the largest differences can be seen in the system total power. At time 339 seconds, a large load/power step is required and the full hybrid system triggers assisting by the HSFW, 48V MGU, and SuperTurbo. The system power is quickly brought up and greatly exceeds the "snap torque" behavior of the engine-only system. This is a good clear example of the desired behavior of the full hybrid system to enable a 30% engine downsizing. Changes in fueling, HSFW speed, and turbo speed can be seen in the corresponding line plots. The hill starting at ~332 seconds shows two distinct assisting events where the HSFW and 48V MGU are brought in and out in quick succession. This indicates an overshoot of the desired power and an undesired retrigger of the assisting. More refinement in the control strategy was found to be needed to avoid this type of behavior, and in Caterpillar's assessment this can be refined with more engineering time within the current L5 controls architecture. Nevertheless, the full hybrid system was built and engineered to a robust enough level to run full transients, interrogate performance, update models with test validation data, and put forth a demonstration assessment of the concept.

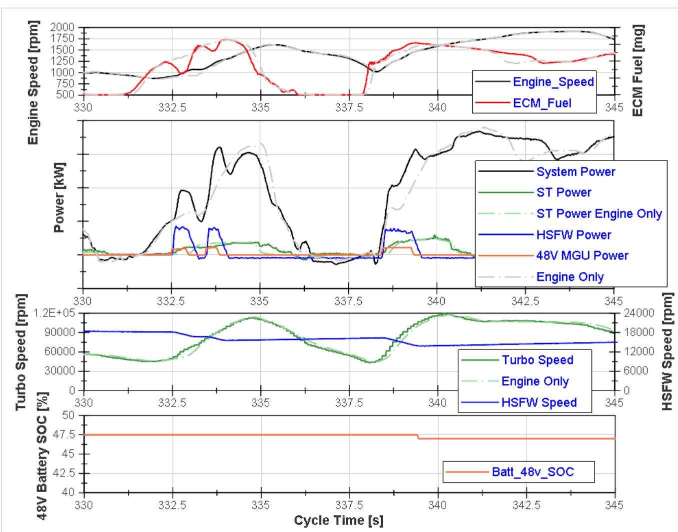


FIGURE 28: ZOOMED IN TRANSIENT PLOT OF HYBRID POWER FROM 745AT NRTC. ENGINE-ONLY DATA OVERLAYED AS DASH-DOT FOR COMPARISON.

5.1 Final Efficiency Summary

For the final assessment of the full hybrid system's capability and efficiency, when running with energy recovery and transient assisting, the validated DYNASTY system model was run for the six key machine cycles and three application NRTCs. This final efficiency accounting can be found in Figure 7, and includes corrections for SOC over the cycle. The NRTC application average benefit was found to increase by 0.7%, while the machine cycle application average benefit was found to increase by 0.6%, with a maximum benefit of 3.0% on Cycle 6 of the 745AT. The suboptimal assisting-only strategy is also listed and accounted for in the program total range.

Table 1 is the authoritative summary of the power system validation results for the hybrid R&D program. Final validation results in the second column can be compared to earlier simulation predictions and the associated component or power system aspect. The ranges of 10.3-13.8% and 11.8-16.1% for the machine and NRTC rows of Table 1 represent the machine application variance from the six machine validation cycles and three NRTC cycles. A program total efficiency improvement range of 10.5 to 25.6% with a midpoint of 17.9% met the program goal of 17±2%, with the 13L concept engine accounting for the majority at 13.1%. Start/stop, advanced engine controls, and reduced cooling parasitic loads were only evaluated in simulation and some extended discussion can be found in [1].

TABLE 1: FINAL QUANTIFICATION OF HIGH-LEVEL EFFICIENCY CONTRIBUTIONS OVER THE BASELINE 18L ENGINE POWER SYSTEM

Duty Cycle	Test or Final Validation Result	Sim. Prediction	Component
Machine Cycles	**10.3 – 13.8%**	9.0-12.6%	13L Concept engine
		0.3-1.1%	SuperTurbo Turbo-compounding
		0.5-2.0%	Thermal Barrier Coatings (TBC)
NRTC Cycles	**11.8 -16.1%**	12.4-15.9%	All Engine-Only Components*
Machine & NRTC Cycles	Assist Only Mode: **(-2.4) – (-0.7)%** With Regeneration: **0.0% – 3.0% ****	0.5-1.9%	High-Speed Flywheel (HSFW) & 48V MGU
Machine Cycles	**0.6% – 3.5% ****	0.6-3.5%	Start/stop implementation
Machine Cycles	**0% ****	0%	Advanced Engine Controls
Machine Cycles	**2.0% – 3.0% ****	2.0-3.0%	Reduced Cooling Parasitic
Machine & NRTC Cycles	***10.5 – 25.6%*** ***Midpoint: 17.9%***	*13.5 – 24.1%*	***Program Total***

***Includes 13L Concept Engine, SuperTurbo, and TBC**
****Simulation Only and Not Tested in Final Validation**

These final measurements and assessments of efficiency benefits for the full hybrid system and engine-only system were used as primary input to the technoeconomic analysis.

6. TECHNOECONOMIC ANALYSIS

Given the complexity of the power system and the significant improvements in fuel efficiency, a total cost of ownership (TCO) analysis, commonly adopted by industry and researchers [8], was completed to elucidate areas of high value and return on investment. The compilation of engine and power system cost, service, fuel consumption, and other relevant information was conducted for input into the analysis. Of course, this information is well known for the base diesel engine system, and this served as the normalizing reference.

The final TCO methodology contained the following input assumptions:

- 1500 hours/year machine application operation (745AT, 988 K/XE, 390F)
- Fuel and DEF consumption averages from the final test validated simulations for:
  - The NRTC for the three machine applications
  - And the six machine cycles validated in the testing
- Start/Stop fuel consumption reduction of 2% (middle of the analysis range of 0.6-3.5% in Table 1)
- 3.7% fuel reduction for Hybrid + Reduced Cooling (middle of the simulation prediction ranges in Table 1)
- 3% Inflation
- Low and High Fuel Prices
  - $4.50/gallon Diesel fuel, $7.00/gallon DEF
  - $6.00/gallon Diesel fuel, $9.00/gallon DEF
- Preventative Maintenance (PM) and overhaul costs
  - Lubrication, service events, service labor, major rebuilding parts, etc.
- Minimum and maximum CAPEX (Capital Expense)
  - Lower end of power system costs and profit margin
  - Upper end of power system costs and profit margin
- Residual power system value
- *No adjustment for the value of increased machine productivity due to a more responsive hybrid power system – this can be significant but is out of scope for the present project and analysis work given the power system's goals of matching the 30% larger baseline diesel engine*

The four data series in Figure 29 successively add the fuel consumption benefits and cost differences when starting with the C18, then the H2D2 13L concept engine-only, then adding start/stop to the H2D2 13L concept engine, then adding the remaining hybrid elements to get the full H2D2 hybrid power system. Differences in first cost, or CAPEX, and fuel consumption drive the TCO payback period with respect to the baseline C18. Only the H2D2 Full Hybrid system indicates it will not pay back quickly (where quickly is considered less than 1.5 years). Fuel cost dominates these applications TCO given the size and duty cycle, so the large H2D2 efficiency improvements can clearly pay for themselves. It should be noted that the lighter shaded areas around the mean trend line indicate the full range of the input assumptions, and therefore give a very strong indication of how confident a specific magnitude of benefit will be, even with the most favorable or unfavorable of state.

Table 2 summarizes the key metrics for the payback timeframe and TCO value for the three configurations of the power system from Figure 29 at two, five, and ten year life cycles. It can be seen that the H2D2 13L core engine, and therefore similarly the new C13D, has an immediate payback as it is a smaller, more compact, and a more cost-effective design than the older and larger C18 baseline. Adding start/stop to the power system quickly pays for itself in less than one year and on average pays back relative to just the core engine at 4.9 years (crossing point of yellow and blue curves in Figure 29). The full H2D2 Hybrid power system has the greatest TCO reduction at the end of ten years, but it takes nearly three years on average to pay back – which may be marginally acceptable. The full system takes 7.8 years to pay back relative to just the core 13L engine, and expanded TCO analysis can be found in the U.S. DOE final report when publicly released [9].

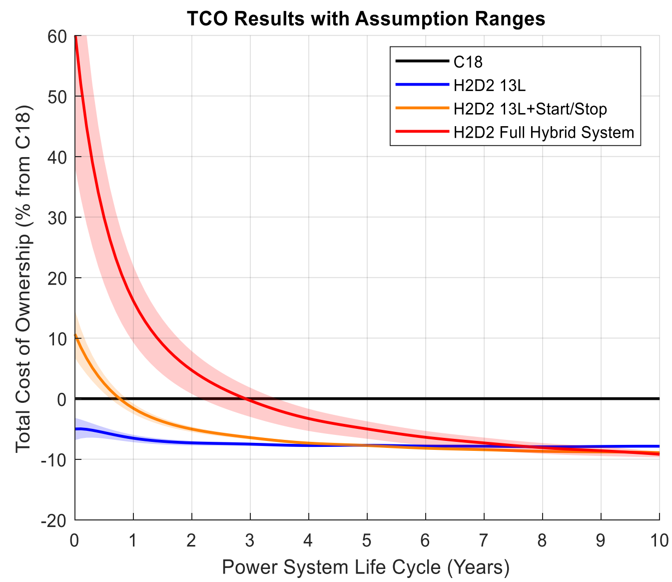


FIGURE 29: TCO RESULTS SHOWING REDUCTIONS (RELATIVE THE C18 BASELINE) FOR THE H2D2 13L, WITH START/STOP, AND WITH FULL HYBRID SYSTEM (HSFW AND 48V MGU AND START/STOP)

TABLE 2: TECHNOECONOMIC ANALYSIS KEY RESULTS – C18 BASELINE REFERENCE

Power system	Min. Pay-back (yrs.)	Ave. Pay-back (yrs.)	Max. Pay-back (yrs.)	Ave. 2 year TCO	Ave. 5 year TCO	Ave. 10 year TCO
H2D2 13L	0	0	0	-7.3%	-7.7%	-7.9%
H2D2 13L + Start/Stop	0.61	0.77	0.88	-5.0%	-7.7%	-8.9%
H2D2 full hybrid system + Start/Stop	2.16	2.92	3.45	+4.7%	-5.0%	-9.2%

7. CONCLUSION

The developed hybrid power system was validated through test and simulation activities to have a range of efficiency improvement of 10.5-25.6% with a midpoint of 17.9% – which met the program goal of 17%±2 efficiency improvement. The majority of the efficiency improvement (13.1%) came from the concept 13L engine, the system achieved superior transient load response, met T4F emissions levels, and a technoeconomic analysis showed that start/stop would pay back in less than one year and the full hybrid system in near three years.

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