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Final Scientific/Technical Report
Spark-Assisted HCCI Residential CHP

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Public Executive Summary

A transformative small spark ignition (SI) internal combustion (IC) engine fueled with natural gas was developed for combined heat and power (CHP) applications using a combination of cycle simulations, computational fluid dynamics (CFD) modeling, and engine experiments. The resulting 1 kW CHP engine was demonstrated to achieve 36.1% brake thermal efficiency (BTE) while meeting aggressive exhaust emissions targets using a low-cost three-way catalyst. Starting from a “clean-sheet” design, modeling tools were used to select the optimal engine parameters and operating characteristics. The disruptive technologies developed for the CHP system have a target lifetime of at least 10 years. The engine cost at volume of 10,000 units was estimated to be \$1,050. A combination of experimental and simulation results were used to identify a path to the program target of 38.6% BTE.

Base engine design parameters (i.e., displacement, speed, bore, stroke, valve timings, and compression ratio) were optimized to reduce friction, heat transfer, and exhaust losses. The base clean sheet prototype engine achieved 32% BTE. Optimization of the lube oil and coolant temperatures added 1.8% BTE. The addition of exhaust gas recirculation added 1.1% BTE. Novel thermal barrier coatings (TBCs) developed over the course of the project added 1.2% BTE. Further improvements in engine friction reduction (1.3% BTE) and improved combustion (1.2% BTE) are estimated to meet the program target 38.6% BTE, while also meeting the stringent criteria pollutant emissions targets.

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Accomplishments and Objectives

This award allowed Wisconsin Engine Research Consultants (WERC), LLC and project partners to demonstrate a number of key objectives. The focus of the project was using analysis led design to develop a clean sheet natural gas fueled spark ignition engine suitable for a high efficiency 1-kWe residential combined heat and power (CHP) system.

A number of tasks and milestones were laid out in Attachment 3, the Technical Milestones and Deliverables, at the beginning of the project. The actual performance against the stated milestones is summarized here:

Table 1. Key Milestones and Deliverables.

Tasks	Milestones and Deliverables
Task 1: Production Engine Build Zero for TBC Selection (Adiabatics) M1.1: Validate TBD durability through bench and engine tests M1.2: Engine durability testing of current TBCs	Q2: M1.1 Bench test 1,000 thermal shock cycle test on aluminum engine components provided by B&S and cast iron blocks. Build and validate baseline performance of TBC test engine demonstrating that the TBCs will survive in a natural gas engine combustion chamber. Results from both tests will determine if the lightweight TBC coatings can be used in the Prototype I engine build. The thermal shock test rig cycles TBC coated test sample through a NG flame (>1100C). It will hold the coated sample in the flame for a predetermined time duration and then pass it through a 55C water bath. The test rig measures both the front (flame) side temperature and the backside (behind the TBC) temperature. Failure occurs when the backside temperature increases 50F over the backside running average. Actual Performance: June 2016 Two TBCs, a mixed polysilazane-bonded TBC and a metal phosphate-bonded TBC, were subjected to and passed 1,000 cycles of thermal cycle bench testing on an accelerated failure laboratory thermal shock test rig. Q4: M1.2 Durability of current TBCs evaluated using OTS natural gas engine. Thermal cycle test to 400 hours. Inspect integrity of TBCs (different for valves and piston head). Provide input for next generation of TBCs and sets a baseline for heat loss reduction in engine models. Actual Performance: June 2016 The two TBCs that passed laboratory thermal shock testing were installed into the genset and durability tested for 400 hours. Some of the coated engine parts experienced minor failures towards the beginning of testing (within the first 60 hours), but after the initial coating deterioration, the coatings remained intact for the rest of the 400 hours.
Task 2: Baseline Development of SA-HCCI (UW-Madison) M2.1: Engine test stand is complete and baseline COTS engine is installed M2.2: Q6: SA-HCCI combustion is demonstrated	Q4: M2.1 Engine test stand is complete and baseline engine is installed. The test set-up includes an A/C motoring dynamometer, high-speed instrumentation for indicating measurements, torque sensor for brake measurements, and state-of-the-art emissions, fuel, and airflow measurements (fully flexible fuel and ignition system). Actual Performance: March 2016 An engine test stand was constructed including an A/C motoring dynamometer, high-speed instrumentation for indicating measurements, torque sensor for brake measurements, and state-of-the-art emissions, fuel, and airflow measurements (fully flexible fuel and ignition system). Q6: M2.2 SA-HCCI combustion is demonstrated at one operating condition with a COV of less than 5% using compressed air to approximate intake boosting device. CR, AFR, EGR, T intake, and spark timing required for successful SA-HCCI combustion will be determined. Actual Performance: June 2017

	SA-HCCI was demonstrated at 100 deg. C intake temperature with a COV of IMEP of ~1%.
Task 3: Base Engine Design: 1-D Modeling and Boosting System (WERC) M3.1: Q2 Design Review	Q2: M3.1 Design review of engine that meets FOA targets and is approved by expert panel. GT-Power design with engine design parameters selected such as bore, stroke, displacement, engine speed and load, identification of heat loss reduction required from TBCs, EGR (or residual mass) requirements, and intake temperature and pressure. Actual Performance: June 2016 A GT-Power model was validated to available test data. Validation was within program targets of 3% error on key parameters. This model was used to perform a wide-ranging DoE on design and calibration parameters. An optimum design and calibration were determined that showed ability to achieve program target BTE. Performance impact from sensitivity to parameter values was performed and used to guide ranking of importance. This ranking of importance was carried out by industry experts on the team.
Task 4: Provide Benchmark Current-State Engine Models and Performance Data (BASCO) M4.1: Delivery of benchmark models and test data	Q1: M4.1 Delivery of benchmark validated GT-Power model(s) and test data for the Briggs and Stratton 126cc single cylinder engine run on gasoline. Supply all required input data for initial CFD modeling of the same engine at engine speeds similar to those to be used on in the final engine (e.g., 1200, 1800 rpm). Actual Performance: March 2016 Benchmark models and test data was supplied for 2 Briggs & Stratton engines to allow initial CFD modeling. Additionally, test data and modeling information was acquired and provided for the Honda PCX125 engine, an engine closer in configuration to the proposed GENSETS engine.
Task 5: Initial CFD Model Validation (WERC) M5.1: CFD model capable of reproducing the measured emissions and performance characteristics of a base engine.	Q1: M5.1 Data from Task 4 will be used to validate the CFD model used to optimize the combustion system. The outcome of this task will be a validated combustion model that can be used to design the SA-HCCI combustion needed to achieve 40% electrical conversion efficiency. Prediction of peak cylinder pressure within 10%, IMEP within 5%, combustion efficiency within 3%, NOx emissions within 20%. Actual Performance: March 2016 A CFD model of the Briggs & Stratton W14 engine was generated and validated to 3 key operating conditions. The maximum prediction error for all conditions was peak pressure=4.48%, IMEP=2.76%, combustion efficiency=0.85% and NOx emissions=1.6%.
Task 6: Conceptual Evaluation of Multiple Engine Concepts (BASCO) M6.1: At least one engine concept designed to sufficient fidelity to allow continued analysis via 1-D and	Q4: M6.1 At least one engine concept designed to sufficient fidelity to allow continued analysis via 1-D and 3-D analytical methods with natural gas as a fuel. Engine will feature a high CR, high PCP capability, minimized top piston ring land crevice volume, and low friction. Actual Performance: December 2016 Three engine concepts were designed in sufficient detail to allow continued analytical modeling and comparisons using 1-D and 3-D methods (WERC effort). The primary difference in these configurations were included valve angle and valve sizes.

3-D analytical methods	
Task 7: Directional CFD Modeling of Concept Analysis (WERC) M7.1: Down selected engine technology / architecture to be used in the phase I prototype engine build.	Q4: M7.1 Down selected engine technology / architecture to be used in the phase I prototype engine build to achieve 40% electrical conversion efficiency. Comparative metrics for the natural gas engines will include volumetric efficiency, combustion duration, heat transfer, combustion efficiency, and peak cylinder pressure. Actual Performance: December 2016 Five engine concepts were assessed for comparative performance. Design variables included bore, stroke, compression ratio, valve diameter and angle. Of the 5 designs considered, the final selection ranked 2nd in volumetric efficiency, 1st in combustion duration, 2nd in heat transfer, 2nd in combustion efficiency and 5th in peak pressure, although the peak pressure was within design limits.
Task 8: Development and Testing of Advanced Boosting System (UW-Madison) M8.1: Bench testing of three boosting concepts M8.2: Optimized boost system delivering required boost to achieve 40% electrical conversion efficiency	Q8: M8.1 Bench testing of three (3) boosting concepts completed. A most favorable method to achieve an intake pressure of 1.3 bar is identified. Actual Performance: December 2017 The program constructed and evaluated the following boosting systems 1) Pressure wave supercharger 2) Small turbo-charger 3) Membrane charger The pressure wave supercharger has the most potential to reach relatively high boost levels; however, the system is complex and would likely result in an increase in system cost. Q12: M8.2 Boost system is installed and tested on Phase II prototype engine. Boost system on the engine delivers an intake pressure of 1.3 bar at the operating condition identified in M3.1. Actual Performance: December 2018 This milestone was not relevant since Task 8 was re-directed based on Q11 discussions with ARPA-e to evaluate the use of pre-chamber spark plugs.
Task 9: Design and Construction of Phase I Prototypes (BASCO) M9.1: Two prototype engines available for testing	Q6: M9.1 Two identical prototype engines are built and ready to be delivered for testing (one for ARPA-E and one for Univ. of Wisconsin – ERC). These engines will feature a high CR, high PCP capability, minimized top piston ring land crevice volume, and low friction. Actual Performance: June 2017 Engine Concept #1 components were fabricated and assembled using a custom cylinder and piston to mate the Honda PCX 125 crankcase to the Honda Metro cylinder head. Engine Concept #3 design and analysis was completed based on Task 7 modeling optimization. Custom castings were ordered with machining of castings and other billet parts being accomplished by the Briggs & Stratton Rapid Prototype Center or outside model shops. Specialty engine components (valves, piston rings, seals, bearings, etc.) were procured from traditional Briggs & Stratton suppliers. Two engines were assembled, one for mechanical development testing at Briggs & Stratton, the other for performance development at UW-Madison ERC.

Task 10: Experimental Evaluation of Phase I Prototype Engine (UW-Madison) M10.1: The Phase I Prototype engine is tested at 1-KWe. M10.2: Go/No-Go checkpoint	Q8: M10.1 The Phase I Prototype engine is tested at 1-KWe. Deliver a phase 1 prototype engine to ARPA-E for independent testing. Brake efficiency of 33% demonstrated using electrically assisted boosting if necessary. Ability to operate with varied methane number from 80-98. Emissions (except NOx) within an order of magnitude from FOA targets. Operable for 50 hours, demonstrating performance towards a lifetime of at least 10 years. TBC is included in the phase 1 prototype engine and survive the 50 hour durability test. Identify a pathway to system noise less than or equal to 55 dB(A) from 3 feet away. Actual Performance: December 2017 An engine was constructed to demonstrate a peak BTE of over 34% with emissions meeting program targets over methane numbers from 70 to 100. TBC evaluation resulted in an efficiency improvement of ~0.7%. Q10: M10.2 Demonstration of working prototype of 29% brake efficiency at 1 kWe on the Phase 1 Prototype engine with all the other metrics of M10.1 accomplished. Actual Performance: December 2017 An engine was constructed to demonstrate a peak BTE of over 34%.
Task 11: GT-Power Validation using Phase I Prototype Engine Data (WERC) M11.1: Validated GT-Power model with refined/enhanced sub-models	Q7: M11.1 Validated GT-Power model with refined/enhanced sub-models. All metrics except exhaust temperature will be validated within 3% of test data. Exhaust temperature validated within 100K of thermocouple reading. Actual Performance: September 2017 A GT-Power model was built and validated to engine concept 1 test data. Seven different operating conditions were considered with program objective of <3% error of BTE, Pmax and airflow and exhaust temperature was within 100K of the thermocouple reading.
Task 12: GT-Power Optimization Using Validated Model (WERC) M12.1: Updated base engine design and operating parameters to achieve 40% electrical conversion efficiency	Q8: M12.1 Updated base engine design and operating parameters to achieve 40% electrical conversion efficiency. The engine design parameters will potentially include engine displacement, bore, stroke, engine speed, compression ratio, valve timings, valve lift profiles, valve diameter, required intake pressure and temperature, EGR level, intake and exhaust geometries, identification of heat loss reduction required from TBCs, boosting system specification, combustion duration and phasing. Actual Performance: December 2017 Validated concept 1 engine model was exercised to optimize bore/stroke ratio, displacement, valve duration and timing, compression ratio, EGR, boost and intake cooling. BTE improvements were on the order of ~0.2% over the concept 1 engine. This was deemed insufficient improvement to alter design direction.
Task 13: CFD Model Validation to Measured Data from Phase I	Q9: M13.1 The CFD model capable of reproducing the measured emissions, performance, and knocking characteristics of the phase I prototype CHP engine. Prediction of peak cylinder pressure within 10%, IMEP within 5%, combustion

Prototype CHP Engine (WERC) M13.1: The CFD model capable of reproducing the measured emissions, performance, and knocking characteristics of the phase I prototype CHP engine	efficiency within 3%, NOx emissions within 20%, onset of knock within 3 crank angle degrees (spark timing). Actual Performance: March 2018 A CFD model of the concept 1 engine was built and validated to test data consisting of 3 critical speed/load points. The maximum error of the 3 point was IMEP=4.7%, Pmax=0.44%, combustion efficiency=2.94% and NOx emissions=8.32%.
Task 14: Computational Optimization of First Generation Prototype CHP Engine (WERC) M14.1: Computational analysis of the First-Generation Prototype CHP engine is complete.	Q14: M14.1 Computational analysis of the First-Generation Prototype CHP engine is complete. Pathways to further engine performance improvement will be identified. Actual Performance: December 2018 CFD analysis was performed to support testing development as well as consider design changes for improved performance. These changes included several spark pre-chamber and intake port designs. As confirmed with test data, spark pre-chamber has performance improvement potential but with careful design optimization. Analysis of the flow structure led to 3 different intake port designs, although none improved performance above current design.
Task 15: Durability Testing. (BASCO) M15.1: 500 hour durability test completed	Q14: M15.1 Based on initial development testing, the Briggs & Stratton dynamometer test engine will be rebuilt with necessary Phase II upgraded components (see below) and endurance tested for a period of not less than 500 hours or until a major engine failure occurs. Engine inspection and wear measurements will be obtained to estimate the engine durability life. Actual Performance: September 2019 Based on initial development testing, the Briggs & Stratton dynamometer test engine was rebuilt with necessary Phase II upgraded components and endurance tested for 535 hours. End of test engine inspection and wear measurements were accomplished and summarized in a report included in the project Q15 update.
Task 16: Development of Improved TBCs with Very Low Thermal or Heat Capacitance and Minimum Porosity (Adiabatics) M16.1: Demonstration of	Q6: M16.1 Depending on results in M1.3 and engine model results, demonstrate second iteration of TBCs based on thermal cycle tests considering options such as reinforced fibers, binder chemistry, post treatment of the coating while maintaining low cost and durability. Actual Performance: December 2016 Three improved TBCs were selected as they showed improvements in thermal properties (density, thermal conductivity, and specific heat) and durability (bond strength and thermal shock) over that of the baseline TBC. Q8: M16.2 Validation and durability testing of new coatings completed. Bench test 1,000 thermal shock cycle test on engine components provided by B-S

at least three improved TBCs M16.2: Validation and durability testing of new coatings completed M16.3: Production variation testing of new coatings completed	(prototype I materials). Durability of current TBCs evaluated using OTS natural gas engine from M1. Thermal cycle test to 400 hours. Inspect integrity of TBCs (different for valves and piston head). Actual Performance: December 2017 Multiple iterations of coatings were subjected to standard thermal shock rig testing as well as 400-hour genset durability testing. The DZ, Dual Layer, RC, PC, and LD PC coatings were all subjected to testing and all of them passed and outperformed the baseline TBC. Q14: M16.3 Production variation testing of new coatings completed. Bench test 1,000 thermal shock cycle test on engine components provided by B-S (prototype I materials) with variations due to production (i.e. thickness, composition, supply variation, etc.) Actual Performance: June 2019 The RC and PC coatings were applied at different thicknesses and subjected to thermal shock rig testing. The coatings were run for 50 cycles at 14 seconds on an updated oxy-acetylene thermal shock rig. All of the coatings passed except for the PC Coating applied at 0.008".
Task 17: Computational Optimization of Low Cost Boosting System (WERC) M17.1: Design of the low cost boosting system meeting the targets specified in Task 3 is complete	Q9: M17.1 Design of the low cost boosting system meeting the targets specified in Task 3 is complete. Expected minimum pressure ratio of 1.2 at the engine operating condition specified / determined in M3.1. Actual Performance: March 2018 A low cost, pressure wave supercharger (PWS) boosting system was modeled. A DoE of PWS design parameters was performed. Boost pressure of 1.25bar was achieved with an improvement of ~1% BTE over non-boosted system. Additionally, several zero to very small cost add-on boosting systems were examined including tuned intake runner lengths and a simple intake resonator. No improvement over baseline engine was found.
Task 18: Design and Construction Phase II Prototype Engines (BASCO) M18.1: Design review of phase II prototype M18.2: Phase II prototype engine available for testing	Q14: M18.1 Design review of Phase II prototype engine detailing plan to reach 40% efficiency incorporating endurance data, performance data and modeling results. Development testing has shown necessary upgrades (see report). Initial performance testing has shown that engine friction is higher than desired. The development test result items detailed above are thought to reduce friction - better piston fit, improved chain guide geometry and material, reduced crankshaft oil windage (oiling jet revision), and improved crankshaft to dynamometer alignment. A properly coated DLC sleeve will reduce piston ring friction. A reduced tension oil ring assembly will also be procured for evaluation. Performance testing to date has also shown that combustion speed is slower than desired. Improved in-cylinder motion (tumble) will be evaluated using WERC modeling capabilities. Improved combustion via a pre-chamber spark plug (passive) will also be considered. Actual Performance: May 2019 Design review of Phase II prototype engine completed. Q14: M18.2 An updated engine will be provided to UW-ERC with Phase II upgrades based on mechanical development and performance testing results

	listed in M18.1. Additional components will include parts for thermal barrier coating, a DLC coated cylinder sleeve, and a reduced tension oil ring assembly. Briggs & Stratton and UW-ERC will work together to procure components for combustion speed improvements. Actual Performance: May 2019 An engine with Phase II upgrades was provided to UW-ERC for continued performance testing. Upgrades included an improved piston profile, a reduced tension oil ring assembly, aligned camshaft lobes, a refinement of the camshaft chain guide geometry, optimized crankshaft/rod lubrication, and an improved crankshaft to flywheel joint design. Testing showed a significant reduction in engine friction and improvement in thermal efficiency. Additional improvement was seen with thermal barrier coating (TBC's) on the piston, cylinder head, and valves. Spark plug passive pre-chamber evaluations yielded a significant increase in combustion speed, but due to heat transfer increases, thermal efficiency improvement was not realized. Performance results are provided under Task 19.
Task 19: Performance Testing of Phase II Prototype (UW-Madison) M19.1: Demonstration of path to 40% electrical conversion efficiency. M19.2: Final demonstration of 40% electrical conversion efficiency (38.6% brake thermal efficiency with assumed 96% efficient alternator and 3% increase from thermoelectric generator)	Q10: M19.1 Demonstration via experimental tests of path to 40% electrical conversion efficiency for 1 kW electric power generation capacity and capacity factor $\geq 99.9\%$, while meeting emissions targets, a system noise ≤ 55 dB, lifetime ≥ 10 years and capital cost $\leq \\$3,000/\text{kWe}$. Actual Performance: December 2018 A combination of simulation efforts and experimental data were used to identify a pathway to achieve 40% efficiency at 1 kWe. Q14: M19.2 40% electrical conversion efficiency (38.6% brake thermal efficiency with assumed 96% efficient alternator and 3% increase from thermoelectric generator) is demonstrated for 1 kW electric power generation capacity and capacity factor $\geq 99.9\%$, system noise ≤ 55 dB, lifetime ≥ 10 years and capital cost $\leq \\$3,000/\text{kWe}$. Ability to operate with varied methane number from 70-98. Steady-state emissions meet FOA targets. Efficiency measured with and without thermal barrier coating. Actual Performance: June 2019 The final system demonstrated a brake thermal efficiency of 35.9% at stoichiometric conditions suitable for meeting NOx requirements with a three way catalyst.
Task 20: Evaluation of project developed technology (BASCO) M20.1: TBC evaluation and	Q14: M20.1 In lieu of further endurance testing of a Phase II version of the GENSETS engine, an evaluation of project developed technology on a current production Briggs & Stratton engine is desired. Outcomes in reducing engine heat rejection with the use of thermal barrier coatings (TBCs) could have important implications in the performance and durability of Briggs & Stratton engines. Likewise, friction and wear reducing coatings, such as diamond-like coatings (DLCs), could provide similar benefits. Performance attributes

friction and wear reduction coatings evaluation	including component operating temperature, exhaust emissions, fuel consumption, and engine power will be evaluated with standard and “treated” engine assemblies. If results are significantly positive, endurance testing will be carried out on a targeted engine for a period of not less than 500 hours or until a major engine failure occurs. Actual Performance: September 2019 Thermal Barrier Coating (TBC’s) could have important implications in the performance and durability of Briggs & Stratton engines. Initial results from the Briggs & Stratton Vanguard 810cc EFI V-twin engine with PC treated pistons and heads and RC coated valves showed encouraging reductions in critical head temperatures and engine knock tendency. However, a significant increase in exhaust hydrocarbons needs to be understood and corrected for TBC’s to be considered on current production engines.
Task 21: Manufacturing and Cost Study (BASCO) M21.1: Estimated manufacturing cost of target system	Q12: M21.1 Estimated manufacturing cost of target system is less than \$3,000/kWe. Actual Performance: December 2019 Engine prototyping costs have been documented at \$21,200 for builds of less than 10 engines. For pilot production volumes around 1,000 pieces, engine cost of \$5,600 has been estimated. For initial production volumes of about 10,000 engines, a cost of \$1,050 has been estimated.
Task 22: Commercialization Assessment (BASCO) M22.1: Complete Concept Analysis Gate	Q12: M22.1 Complete Concept Analysis Gate. Develop initial financial and commercial presentation for B&S stage gate process. Includes outcome of manufacturing cost estimates, Concept Gate presentation as deliverable. Actual Performance: December 2019 Briggs & Stratton provided a presentation of various elements related to successful commercialization of new technology in an established market. The challenges of retrofitting traditional systems and providing an attractive payback period would limit the mCHP market acceptance. Sales channel development and after-sales service support were additional concerns. For these reasons, B&S opted not to pursue development of a residential mCHP appliance.
Task 23: Manufacturing and Cost Study of TBC Coatings (Adiabatics) M23.1: Demonstration of a quality control plan for manufacturing engine components with TBC applied at the component factory	Q14: M23.1 Demonstration of a quality control plan for manufacturing engine components with TBC applied at the component factory. Included in this plan will be an FMEA and PFMEA with input from production intent suppliers. Actual Performance: June 2019 A Process Failure Modes and Effects Analysis (PFMEA) was created and revised several times before it was finalized. Preliminary discussions were had with Permawick Corporation if large quantities of coating would have to be applied in the future.

Task 24: Techno-Economic Analysis with Quarterly Updates (BASCO) M24.1: Define the unmet market need and required value proposition M24.2: Deliver v1.0 of Techno-Economic Analysis: Quantify technology differentiation M24.3: Assess technical and commercial feasibility M24.4: Deliver v2.0 of Techno-Economic Analysis: Establish Commercial Relevancy M24.5: Deliver v3.0 of Techno-Economic Analysis: Establish technology / product market viability	Q2: M24.1 Identify and quantify the unmet need in the residential space and water heating markets and determine the required value proposition. Actual Performance: December 2019 Over the course of the three year project there have been significant changes to the commercialization related activities for the GENSETS program. Techno-Economic Analysis (TEA) focus shifted from the mCHP application, the focus at program initiation, to the prime-mover technology powering a mCHP appliance. Q2: M24.2 Identify and assess the individual technical components of the complete solution (adjust with initial modeling results). Actual Performance: December 2019 It was demonstrated that the GENSETS performance and cost targets for the prime-mover may not be appropriate for large scale adoption of the technologies. This has led to a reconsideration of the deliverables, focusing on project developed technologies that may be appropriate for current production Briggs & Stratton engines. Q6: M24.3 Determine the technical feasibility of addressing the identified unmet market need with the proposed technology. This will include the following and will be informed by modeling and experimental activities conducted as part of technology development. Actual Performance: December 2019 Task 20 summarizes the technical evaluation of Thermal Barrier Coatings (TBC's). If the hydrocarbon emissions increase can be mitigated and the temperature and knock tendency further improved, there would be considerable interest in TBC application to certain Briggs & Stratton engines. Q6: M24.4 Further refine the analysis outlined in M24.1. Actual Performance: December 2019 Discussions with Adiabatics staff indicated the TBC application would be in the \$0.50 - \$1.00 per cylinder range. This could be of value with increases in engine power, fuel efficiency, and durability, improvements thought possible directly with TBC application or enabled with the use of TBC's (such as allowing higher compression ratio or operating with lean fuel mixtures). Q12: M24.5 Summarize project findings with respect to primary component technologies that provide benefit relative to standard internal combustion engine technologies. Actual Performance: December 2019 Further development of TBC's must mitigate the increase in HC emissions seen with initial testing. If further development proves significant benefits to engine power, fuel efficiency, and durability at the projected cost adder, it is likely that Briggs & Stratton would implement TBC's on some engines.
Task 25: Technology-to-Market (T2M) with	Q6: M25.1 Deliver moderate fidelity, Go-to-market strategy and plan, including mitigation of risk factors, presented to and approved by ARPA-E. Include v2.0 Techno-Economic Analysis and 1st-year technical milestone information

Quarterly Updates (BASCO) M25.1: Deliver v1.0 of T2M Plan M25.2: Deliver v2.0 of T2M Plan M25.3: Report on IP Strategy M25.4.1: Continuation Scenarios M25.4.2: Continuation Funding M25.4.3: T2M Summary	Actual Performance: December 2019 Technology to market activities concluded. Q10: M25.2 Deliver high fidelity, Go-to-market strategy and plan, including mitigation of risk factors, presented to and approved by ARPA-E. Include v3.0 Techno-Economic Analysis and 2nd-year technical milestone information Actual Performance: December 2019 Technology to market activities concluded. Q6: M25.3 Report on IP strategy identifying topics generating IP, patent filings,, potential avenues for additional IP, and assessing potential of generated IP to other fields of interest Actual Performance: December 2019 Briggs & Stratton filed US Provisional Application No. 62/641952 titled "Engine with Compact Valvetrain " on March 12, 2018. The GENSETS engine is described in detail, with focus on the unique Type II valvetrain arrangement. The application was not converted (required by March 12, 2019 to retain the original priority date), after discussion with ARPA-e staff due to the narrow scope of the IP with limited usefulness to Briggs & Stratton. Briggs & Stratton is interested in further development of TBC technology as protected by Adiabatics filed IP. Q7: M25.4.1 Identify and analyze potential continuation scenarios that may include: sale of technology (some or all), continued development amongst project partners, channel strategies for commercialization, and funding mechanisms/organizations for different scenarios. Actual Performance: December 2019 The initial GENSETS concept engine developed by the team proved to have exceptional efficiency with early indications of good durability; however, no continuation plans for this engine have been identified. The potential exists to implement the TBC technology developed during the course of the project on existing Briggs & Stratton production engines. Q9: M25.4.2 Report on initial conversations with partners / capital providers in support of leading scenarios identified in previous milestone. Actual Performance: December 2019 Technology to market activities concluded. Q12: M25.4.3 Deliver at written summary and presentation regarding Tech-to-Market activities related to the complete prime-mover and component technologies. Including: ● Market assessment and challenges ● Techno-Economic Analysis summary ● Summary of technical development to date ● Intellectual property summary ● Continuation plans for technologies and project participants Actual Performance: December 2019 Technology to market activities concluded.
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Task 26: Spark Ignition TBC's (UW-Madison) M26.1: TBC's and HC Emissions	Q18: M26.1 Compare HC emissions with the latest TBC formula on the Cummins B SI engine using gaseous fuel (NG) and gasoline. Determine how fuel type affects TBC performance (temperature reduction, knock propensity, etc). Actual Performance: October 2021 Results show increases in HC emissions when using the SPC piston for both gasoline and methane cases. Additional efforts would be required to minimize HC emissions.
Task 27: Evaluation of Adiabatics Coatings in a Medium Duty Diesel Engine (UW-Madison) M27.1: Baseline Diesel Experiments Complete M27.2: Pistons coated with Adiabatics TBC M27.3: Piston coated with conventional plasma sprayed coating M27.4: Engine Experiments Complete with Adiabatics Coating M27.5: Engine Experiments Complete with Plasma Coating	Q16: M27.1 Single-cylinder engine experiments are performed in a medium duty diesel engine with an uncoated piston to provide baseline data for comparison at 3 operating conditions. Actual Performance: October 2020 Baseline engine experiments were completed to enable comparison of emissions and performance when using coated pistons. Q17: M27.2 3 pistons are coated with Adiabatics thermal barrier coating at 3 different thicknesses. Actual Performance: October 2020 Multiple pistons were procured with Adiabatics coating and used for testing. Q17: M27.3 1 piston is coated with a conventional plasma spray coating Actual Performance: October 2020 A piston was coated with a conventional YSZ TBC. Q18: M27.4 The engine experiments of M27.1 are repeated with the Adiabatics coated pistons from M27.2 and emissions, fuel consumption, and exhaust temperature are measured and compared to the results from M27.1 Actual Performance: March 2021 Engine experiments with coated pistons were completed. Overall, an improvement in efficiency was observed at the higher load conditions. Minimal impact in emissions was observed. Coated pistons enabled increased exhaust temperature. Q19: M27.5 The engine experiments of M27.1 are repeated with the Plasma coated pistons from M27.3 and emissions, fuel consumption, and exhaust temperature are measured and compared to the results from M27.1 and M27.4. Actual Performance: June 2021 The engine experiments of M27.1 were repeated with YSZ coated pistons. In most cases, the YSZ coated pistons showed similar performance to that of the uncoated pistons.
Task 28: TBC's for gasoline SI engines (Adiabatics) M28.1: Provide TBC coated parts for UW-ERC test engine(s)	Q16: M28.1 Provide TBC coated parts for UW-ERC test engine(s) Actual Performance: March 2020 The SPC TBC was applied to two sets of cylinder heads and pistons, while the RC TBC was applied to the valves of a Briggs & Stratton engine. Their thicknesses were documented and sent to Briggs & Stratton to undergo HC emissions testing.

M28.2: Run durability test on new TBC on spark ignited gasoline engine.	Q17: M28.2 Run durability test on new TBC on spark ignited gasoline engine. Actual Performance: December 2020 The SPC Coating + Sealer was subjected to 100-hours of durability testing. After the 100 hours were finished, a HC lambda sweep using a Snap-On 5-gas analyzer was completed. After disassembly, the coated parts were examined which showed that the TBCs on both sets of cylinder heads and pistons remained in good condition.
Task 29: Diesel Engine TBC's (Adiabatics) M29.1: Furnish plasma sprayed partially stabilized ZrO2 and new TBC	Q18: M29.1 Furnish plasma sprayed partially stabilized ZrO2 and new TBC to Univ. of Wisconsin for testing. Adiabatics will contribute to analysis of TBCs after testing. Actual Performance: December 2020 Adiabatics coated four pistons: two with PSZ and two with the SPC Coating. After the coatings were complete, Adiabatics sent the pistons to UW for diesel engine testing and agreed to assist with any analysis requested.

Task 1 - Production Engine Build Zero for TBC Selection

Summary

Figure 1.1 shows the timeline for the Production Engine Build Zero for TBC Selection. This task objective was focused on selecting and validating two kinds of Adiabatics' original lightweight TBCs on bench and engine durability testing. The original best two coatings were selected and the properties were documented. The two selected original TBCs were a polysilazane-bonded TBC which adheres well to aluminum alloys and the second TBC type selected was a metal phosphate-bonded TBC which can withstand high temperatures for primary application to both ferrous (such as valves) and non-ferrous substrates.

These two coatings were subjected to 1,000 cycles of thermal cycle bench testing in which they were tested on an accelerated failure laboratory thermal shock test rig in Q2 of 2016 with base metals of A356 aluminum (polysilazane TBC), Ductile cast iron (metal phosphate TBC), and Silchrome-1 steel valve material (metal phosphate TBC) which represent materials that Briggs and Stratton plan use in the demonstration engine.

After undergoing 1,000 cycles of thermal cycle bench testing, these coatings were then installed into a production natural gas engine supplied by Briggs & Stratton and were run for 400 hours for durability testing.

Engine modifications were also made to the Briggs & Stratton natural gas engine in which the engine back of the engine was machined so that access to the pistons would be much easier.



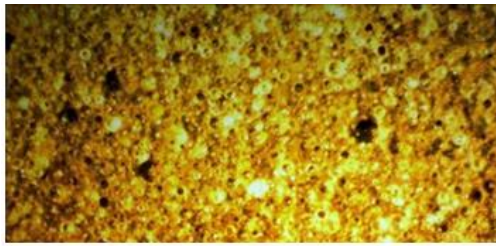
Figure 1.1. Timeline for Task 1, Current TBC Selection.

Details

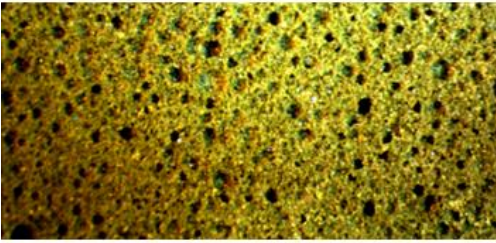
The first type of TBC selected was a mixed polysilazane-bonded TBC which adheres well to aluminum alloys. The coating targets pistons and cylinder heads made of non-ferrous materials such as aluminum. This TBC was selected because of high temperature capability for aluminum (non-ferrous) substrates, ease of application, and manufacturability.

The second TBC type selected was a metal phosphate-bonded TBC. The primary application for this type of coating is to both ferrous (such as valves) and non-ferrous substrates. This TBC was selected because of the high temperature capability for ferrous substrates, ease of application, and manufacturability.

Figure 1.2 shows the two coatings that were selected and where they fall on the performance chart relative to traditional coatings.



Polysilazane TBC



Phosphate TBC

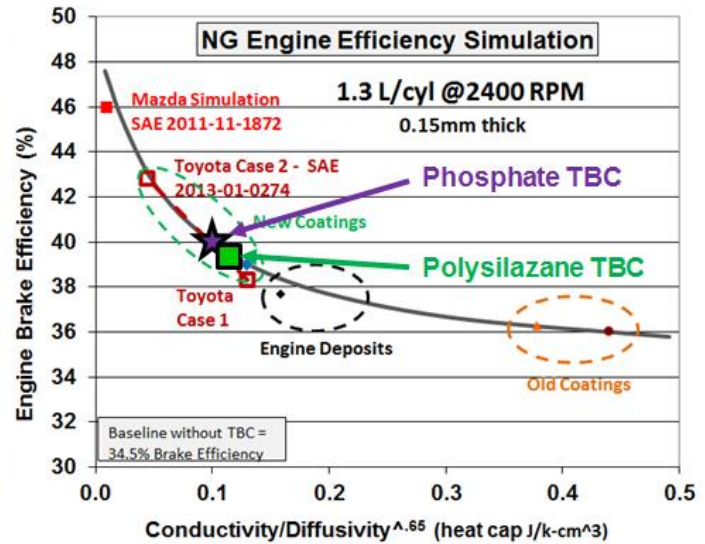


Figure 1.2. Adiabatics' original coating selections.

In order to meet Milestone 1.3: Durability of Current TBCs Evaluated Using OTS Natural Gas Engine, the two coatings that passed 1,000 cycles of standard thermal shock rig testing were installed into the genset and durability tested for 400 hours. Within the first 60 hours of testing, some flaking of the TBC occurred on Piston 1. Replicating this same occurrence was done using a re-designed test so that the torch was switched from natural gas to oxy-acetylene. It was found that a 50 cycle test on the oxy-acetylene rig was comparable to 1,000 cycles on the natural gas rig. Therefore, the coatings were run for 50 cycles at 14 seconds rather than 1,000 cycles at 24 seconds. The rest of the coating remained intact and in good condition as did Piston 2 throughout the entire 400 hours. Cylinder 1 experienced craze cracking within the first 60 hours, but coating did not deteriorate thereafter. This craze cracking was due to a thick spot of coating on the cylinder. Coating on Cylinder 2, however, did not experience any craze cracking and only experienced very slight chipping at the end of inspection. An in-depth test report was done at the conclusion of testing and submitted to Briggs & Stratton for review.

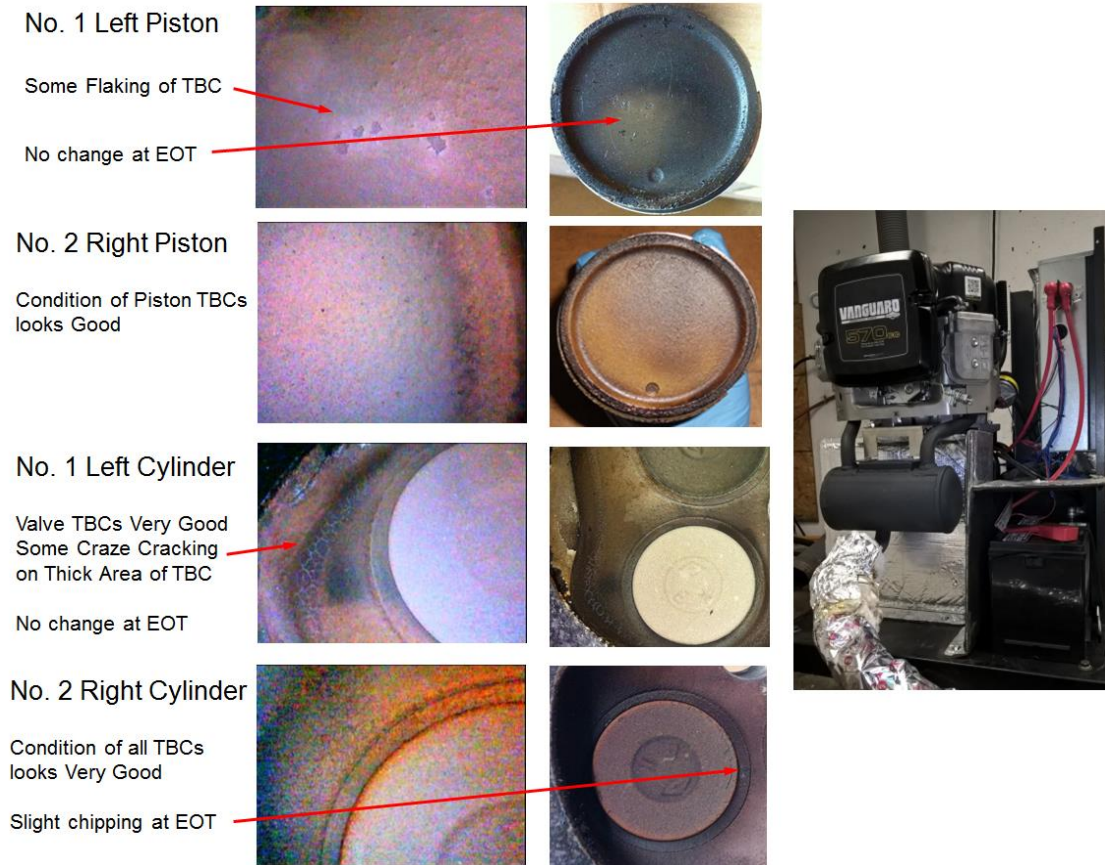


Figure 1.3. Inspection of coated engine parts after 400 hours of testing as compared to 60 hours.

Conclusions

Milestone 1.1: Bench Test 1,000 Thermal Cycle Test on Aluminum Engine Components was completed on time at the end of Q2 of 2016 after the two TBCs selected for initial testing were subjected to 1,000 standard cycles on Adiabatics' thermal shock rig. The TBCs both passed, although the first engine durability test proved that the thermal shock rig did not duplicate the minor failures experienced during testing. The thermal shock rig was later switched from natural gas to oxy-acetylene later in testing (for more detail, see Task 16) which allowed for duplicating coating failures much quicker and more accurately.

Milestone 1.2: Engine Build and Validation Completed was completed on time at the end of Q2 of 2016 after a production natural gas engine was supplied by Briggs & Stratton. This engine was then installed into Adiabatics' test cell and instrumented with a fuel flow meter, air-fuel meter, and cooling fans.

Milestone 1.3: Durability of Current TBCs Evaluated Using OTS Natural Gas Engine was completed on time at the end of Q2 of 2016 after the two TBCs that passed 1,000 cycles of standard thermal shock rig testing were installed into the genset and durability tested for 400 hours. Some of the coated engine parts experienced minor failures towards the beginning of testing (within the first 60 hours), but after the initial coating deterioration, the coatings remained intact for the rest of the 400 hours.

Task 2 - Baseline Development of SA-HCCI

Summary

The objective of this task is to perform preliminary engine testing in the SA-HCCI regime to provide data for computational model validation and initial assessment of the advanced combustion engine. To complete this effort, we constructed a 1 kWe test stand with an A/C motoring dynamometer, high-speed instrumentation for indicating pressure measurements, a torque sensor for brake measurements, state-of-the-art instrumentation, and a fully flexible control system. During Q5 effort focused on completing the design of an intermediate engine setup to collect data for an appropriately sized configuration before the delivery of the first-generation prototype engine from Briggs and Stratton. This engine system coupled the crank train and cylinder head assembly/cam train from two off-the-shelf engines. The system provided a displacement and bore/stroke ratio similar to the optimized configuration. The Q6 effort focused on collecting data for SA-HCCI combustion to determine the compression ratio (CR), air-fuel ratio (AFR), exhaust gas recirculation (EGR) level, intake temperature, and spark timing required for successful SA-HCCI combustion. Milestone M2.2: "SA-HCCI combustion is demonstrated" was successfully completed. The test stand developed in this task was further used in Task 10 to evaluate the first-generation prototype and in Task 19 to evaluate the final prototype.

Details

An engine test stand was constructed and a baseline engine (Honda PCX 125 cc) was installed. Engine control uses a Woodward® LECM engine controller. Fig. 2.1 shows the test stand.

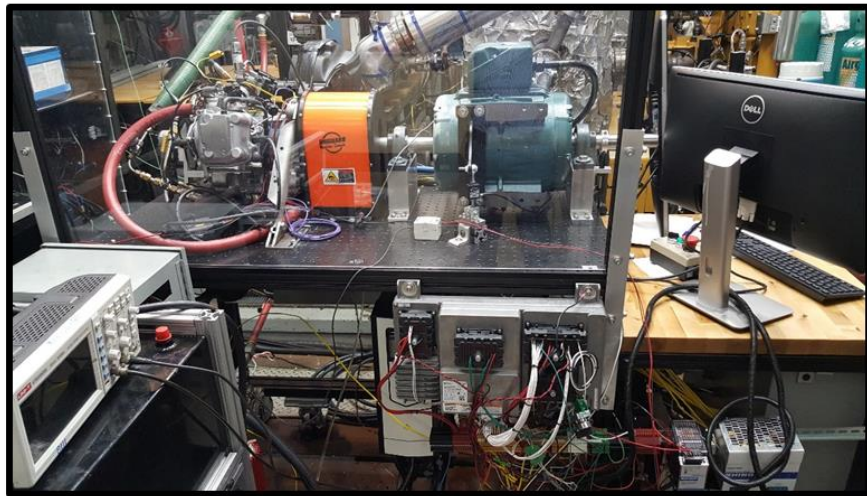


Figure 2.1. Test stand setup completed at the end of Q4.

Table 2.1 shows a detail of the instrumentation used in this laboratory. Cylinder pressure is acquired with a Kistler piezoelectric transient pressure transducer. This is a non-water-cooled version due to size constraints and a flame shield is used to reduce the effects of thermal shock. The charge signal is amplified by a dual-channel Kistler 5064C charge amp with the Kistler 2852A signal conditioning platform with drift compensation and adjustable time constant. Data is digitally encoded using a BEI digital encoder giving 0.25 CAD resolution. An Omega MMA050 pressure transducer records the manifold air pressure and allow for synchronous pegging of the cylinder pressure. A Transducer

Techniques MDB-25 load cell measures brake torque. A hall effect sensor was mounted to the motor shaft, providing feedback to the motor drive. Temperature measurements are acquired using Omega k-type thermocouples with low drift specification. All pressure measurements are acquired using various Omega laboratory-grade pressure transducers typically with accuracy better than 0.25% full scale. When operating on shop air, mass airflow is measured using an Endress and Hauser Promass 80A04 coriolis flow meter. This is positioned on the high-pressure side of the intake pressure controller to eliminate choking issues and minimize error relating to charge density and is accurate to 0.5% of measurement. Fuel flow is measured using an Endress and Hauser Promass 80A01 with an accuracy of 0.25% of measurement down to flow rates of 0.0167 kg/hr.

Table 2.1. Laboratory Instrumentation.

Measurement	Instrument	Accuracy
Cylinder Pressure	Kistler PiezoStar Pressure Sensor	<0.3 % full scale
Charge Amplifier	Kistler 5064C11	0.1 % full scale
Signal Conditioner	Kistler 2852A	0.1 % full scale
Shaft Encoder	BEI 2880 Pulse/Rev	0.25 [CAD]
Manifold Absolute Pressure	Omega MMA050	0.25% Full Scale
Intake Air Temperature	Omega Type K Thermocouple	<2.8 [°C]
Manifold Absolute Pressure	Omega MMA050	0.25% Full Scale
Mass Air Flow	Endress and Hauser Promass 80A04 (when boosted)	0.5 % of measurement
Exhaust Temperature	Omega Type K Thermocouple	<2.8 [°C]
Exhaust Pressure	Omega PX15	0.2% Full Scale
EGR Temperature	Omega Type K Thermocouple	<2.8 [°C]
Coolant Temperature	Omega Type K Thermocouple	<2.8 [°C]
Oil Temperature	Omega Type K Thermocouple	<2.8 [°C]
Intake O2 Measurement	Bosch LSU 4.2 Lambda Sensor	± 0.007, 0.01, 0.05 (0.8-1.7)
Exhaust O2 Measurement	Bosch LSU 4.2 Lambda Sensor	± 0.007, 0.01, 0.05 (0.8-1.7)
Fuel Flow	Endress and Hauser Promass 80A01	0.25 % of measurement

Exhaust emissions are sampled using a heated line connected to a Thermo-Nicolet Nexus 670 FT-IR. Twenty-two exhaust gas species are measured using the OMNIC software based on an UW-developed emissions quantification method. Concentrations of CO₂, NO_x, unburnt hydrocarbons and CO are post-processed in real-time with an in-house developed LABVIEW code.

The completion of the test stand allowed initial testing to be performed to analyze the system level performance and emissions of a commercial off-the-shelf (COTS) engine. The engine is a Honda PCX 125 cc engine with a bore of 5.24 cm and a stroke of 5.79 cm. The engine has a geometric compression ratio

of 11:1 and is referred to as Concept 0. The engine was operated at 1500 rev/min and wide-open throttle. A spark timing sweep was performed at a stoichiometric equivalence ratio and an equivalence ratio sweep was performed at maximum brake torque (MBT) spark timing. The operating conditions are summarized in Table 2.2.

Table 2.2. Operating conditions for baseline P0 engine testing.

Parameter	Spark Timing Sweep	ϕ Sweep
Engine Speed (rev/min)	1500	
Throttle Position (%)	100	
EGR (%)	0	
Intake Pressure (bar)	1	
Intake Temperature (°C)	25	
Coolant Temperature (°C)	90	
Equivalence Ratio (-)	1	0.75-1
Spark Timing (°ATDC)	sweep	MBT

Figure 2.2 shows the cylinder pressure and apparent heat release rate (AHRR) for the spark timing sweep and equivalence ratio sweep and Figure 2.3 shows the emissions and performance data. The data shows the test cell and data acquisition systems are functional, meeting Milestone M2.1, “Engine test stand is complete and baseline engine is installed”.

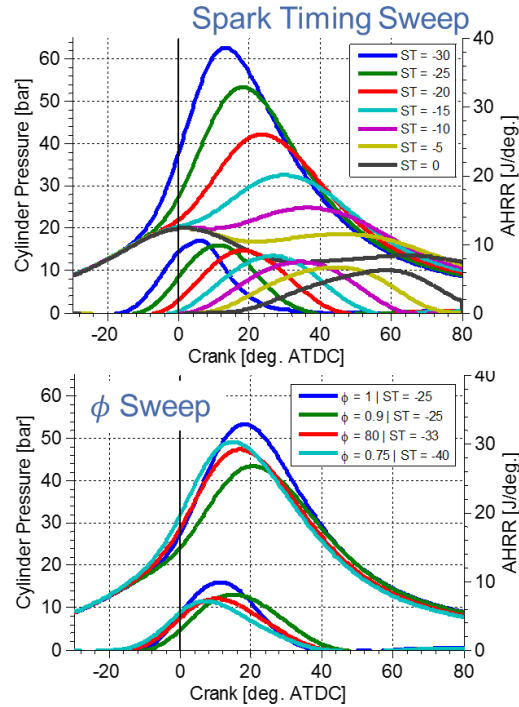


Figure 2.2. Cylinder pressure and apparent heat release rate (AHRR) for (top) the spark timing sweep and (bottom) equivalence ratio sweep.

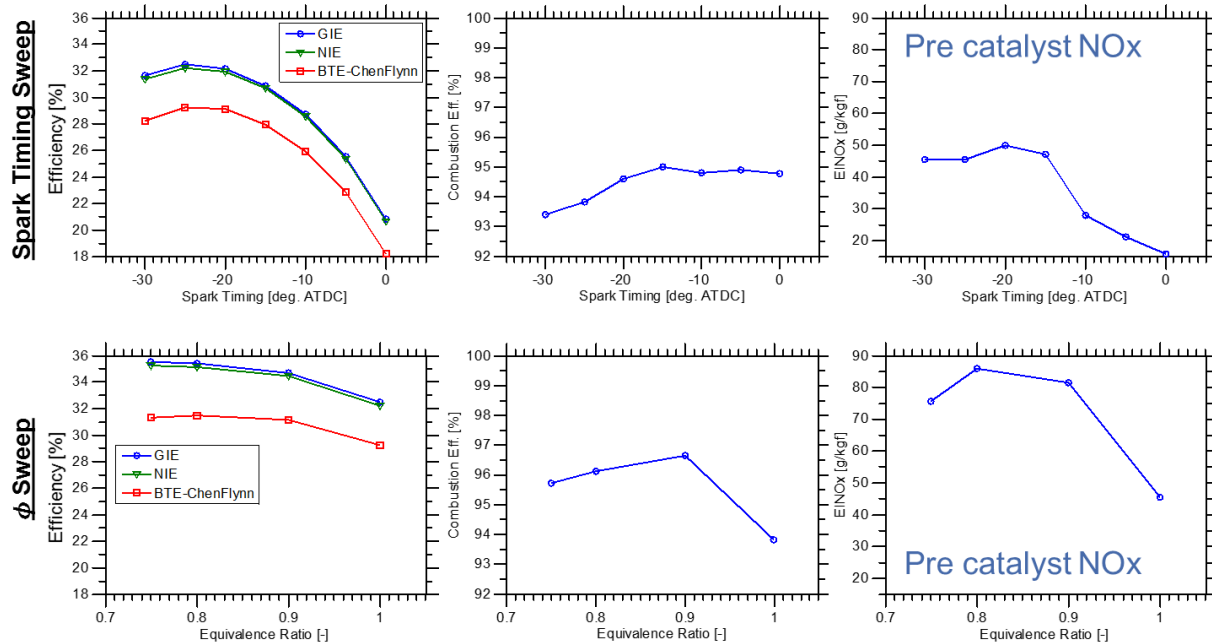


Figure 2.3. Emissions and performance for (top) the spark timing sweep and (bottom) equivalence ratio sweep. GIE = Gross Indicated Efficiency. NIE = Gross Indicated Efficiency. BTE = Brake Thermal Efficiency. Note that a force transducer failed during data acquisition, accordingly, the BTE was calculated using a correlated friction value.

Simulation identified an optimized engine configuration with a bore of 40 mm, a stroke of 56 mm and a compression ratio of 18:1. Off the shelf hardware was identified and a version of this engine was created by mating a Honda PCX 125 lower end with a Honda Metro 50 top end (named Concept 1 engine). Table 2.3 compares the important geometric parameters of the GT-Power optimized configuration and the Concept 1 engine. The largest difference is in the intake seat diameter. The optimized configuration used a diameter of 14.5 mm while the Concept 1 head has an intake diameter of 17 mm. This difference should be kept in mind but is not expected to have a substantial impact on the results.

Table 2.3. Comparison of geometric parameters for the optimized configuration and Concept 1 engine.

	Optimized	Concept 1	% diff
Displacement (cc)	70	71	1.4
Bore (mm)	40	39.5	-1.3
Stroke (mm)	56	57.9	3.4
Bore/Stroke Ratio (-)	0.71	0.68	-4.2
CR	18	17	-5.6
Intake Seat Dia. (mm)	14.5	17	17.2
Exhaust Seat Dia. (mm)	12.5	12.5	0.0

The effort focused on collecting data for spark-assisted HCCI to determine the CR, AFR, EGR, intake temperature, and spark timing required for successful SA-HCCI combustion. Table 2.4 shows the operating parameters for the demonstration of SA-HCCI combustion. The engine was operated at 1200 rev/min near bar 5 bar IMEP. Intake temperature was increased until stable HCCI combustion was achieved using 80% methane by mole (balance n-heptane). An intake temperature near 100° C was required due to the low compression ratio of the current test engine. A spark was added near 40° BTDC to control combustion phasing.

Table 2.4. Operating parameters for spark assisted HCCI (SA-HCCI) demonstration.

	Parameters	[Units]	Value(s)
Constant Parameters	Speed	[RPM]	1200
	Injection Strategy	[-]	Fully Premixed
PV Parameters (controlled)	Load – IMEP _g	[bar]	4-5
	CA50	[°ATDC]	Any
Swept Variables (DOE)	Intake Air Temperature	[°C]	100
	Intake Air Pressure	[kPa]	101 - 120
	Exh. Back Pressure	[kPa]	101
	Coolant Temp	[C]	80
	EGR Rate	[% by mass]	0
	Methane Fraction	[Molar %]	80, 86, 90

Figure 2.3 shows cylinder pressure and apparent heat release rate for SA-HCCI for several methane quantities. As the methane quantity is increased, combustion is phased later in the cycle due to the decreased reactivity of the charge.

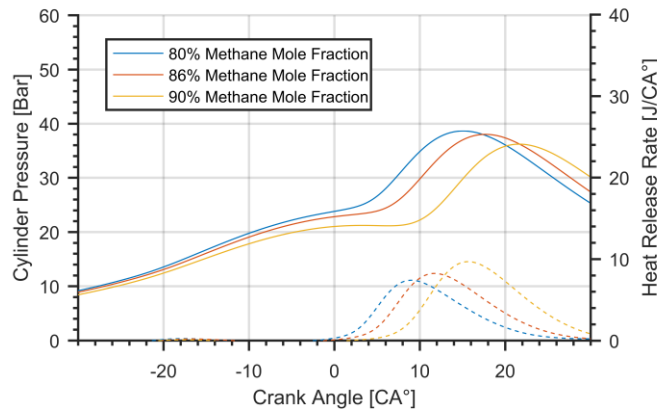


Figure 2.3. Combustion phasing for SA-HCCI at several methane mole fractions (balance n-heptane).

Figure 2.4 shows the coefficient of variance (COV) of indicated mean effective pressure (IMEP) and combustion efficiency over the same methane mole fraction sweep. For all cases the COV of IMEP is near 1%. This is substantially better than the target value of 5% and shows achievement of Milestone M2.2 “SA-HCCI combustion is demonstrated”.

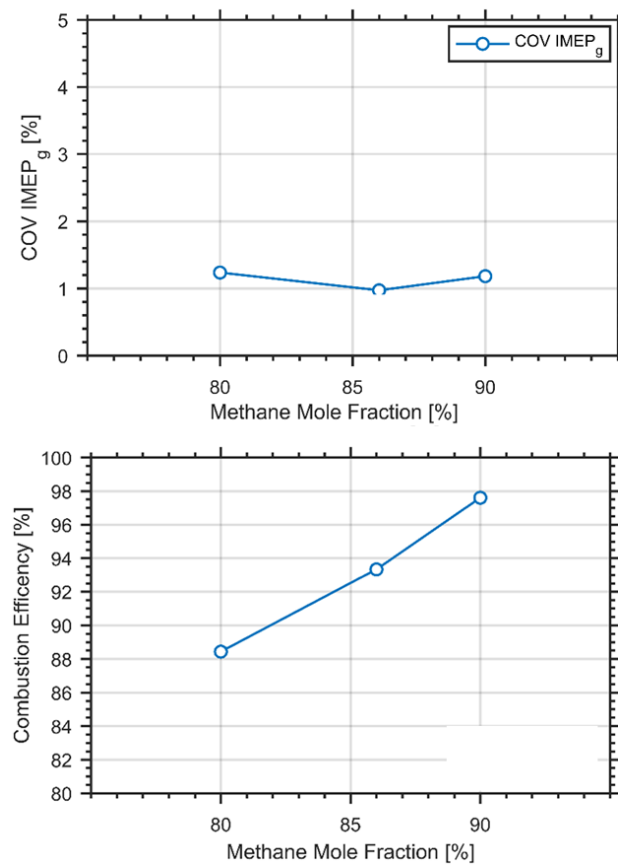


Figure 2.4. COV of IMEP and combustion efficiency as a function of methane mole fraction for SA-HCCI. Note that the target COV of IMEP is less than 5%.

The data from the experiments described above were then used to identify the required compression ratio and exhaust gas recirculation rate for operation on higher methane number fuels. Constant volume reactor simulations using Cantera [1] were performed with the reduced reaction mechanism of Wang et al. [2]. Initial conditions were taken from a polytropic compression process from bottom dead center (BDC) to top dead center (TDC). Results of the analysis are shown in Figure 2.5. The engine used in the experiments had a compression ratio near 11:1. With the 90% methane blend used and an equivalence ratio of 0.6, this results in a representative ignition delay of ~12 crank angle degrees. The fuel is then replaced with 97% methane and the equivalence ratio is increased to unity. To achieve the same ignition delay, the compression ratio needed to be increased to ~17:1. Simulations were also repeated adding 20% EGR. The EGR increases the ignition delay and suggests the compression ratio should be near 19:1. The findings of the study are consistent with the design intent for the phase I prototype under development in Task 9.

¹ David G. Goodwin, Harry K. Moffat, and Raymond L. Speth. *Cantera: An object-oriented software toolkit for chemical kinetics, thermodynamics, and transport processes*. <http://www.cantera.org>, 2017. Version 2.3.0. doi:10.5281/zenodo.170284

² Hu Wang, Qi Jiao, Mingfa Yao, Binbin Yang, Lu Qiu, Rolf D Reitz, "Development of an n-heptane/toluene/polyaromatic hydrocarbon mechanism and its application for combustion and soot prediction" *Int. J. Engine Research*. Vol 14, Issue 5, pp. 434 – 451, 2013, doi: 10.1177/1468087412471056.

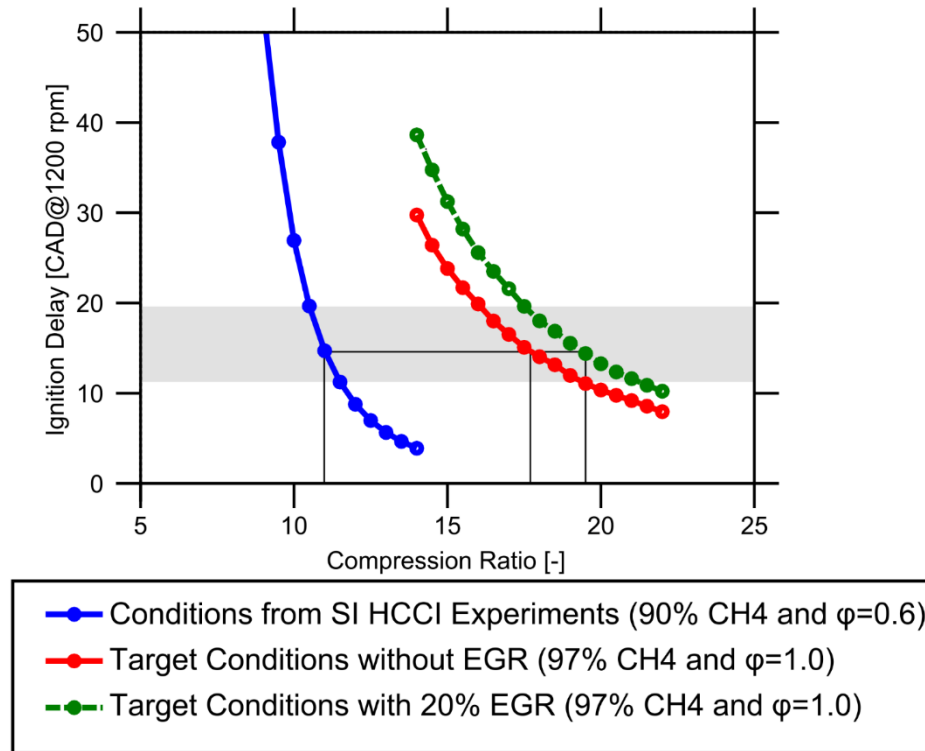


Figure 2.5. Ignition delay as a function of compression ratio for constant volume reactor simulations.

Task 3 - Base Engine Design: 1-D Modeling and Boosting System

Summary

A GT-Power model with corresponding test data was provided. Test data included 3 wide open throttle speeds; 2400rev/min, 3060rev/min, and 3600rev/min. The model was acceptably correlated to test data with emphasis on lower speed match. Boundary conditions from the model were provided for full cycle CFD simulations.

An analytical optimization of design and calibration variables was performed using GT-Power. A model was built of the Honda PCX125 engine and validated to test data generated at Briggs&Stratton. The optimization study focused on base engine geometry including displacement, bore/stroke ratio, cam durations and phasing and compression ratio and boundary conditions including boost pressure, inlet temperature and EGR. An all inclusive DoE was performed and an optimum was determined. Sensitivity of design and calibration variables including combustion parameters based on the optimum was performed. The model was updated with an initial design from Briggs&Stratton. This model was exercised with several different Thermal Barrier Coatings (TBC). A final optimized design was determined and evaluated by the team to rank parameters. This was completed by end of project Q2 satisfying Milestone M3.1, "Design review of engine that meets FOA targets and is approved by expert panel. GT-Power design with engine design parameters selected such as bore, stroke, displacement, engine speed and load,

identification of heat loss reduction required from TBCs, EGR (or residual mass) requirements, and intake temperature and pressure.”

Details

In Task 3, Computer modeling will be utilized to guide the initial prime path design as well as identify any viable alternatives. The outcome of this task will be an identification of the base engine design parameters needed to achieve 40% electrical conversion efficiency. The engine design parameters will include engine displacement, bore, stroke, engine speed, engine load, compression ratio, valve timings, valve lift profiles, valve diameter, required intake pressure and temperature, EGR level, intake and exhaust geometries, identification of heat loss reduction required from TBCs, boosting system specification, combustion duration and phasing.

A GT-Power model was correlated to Honda PCX 125cc engine data (see Figure 3.1). The validated PCX125 model was parameterized for optimization studies. Variables considered are summarized in Table 3.1.

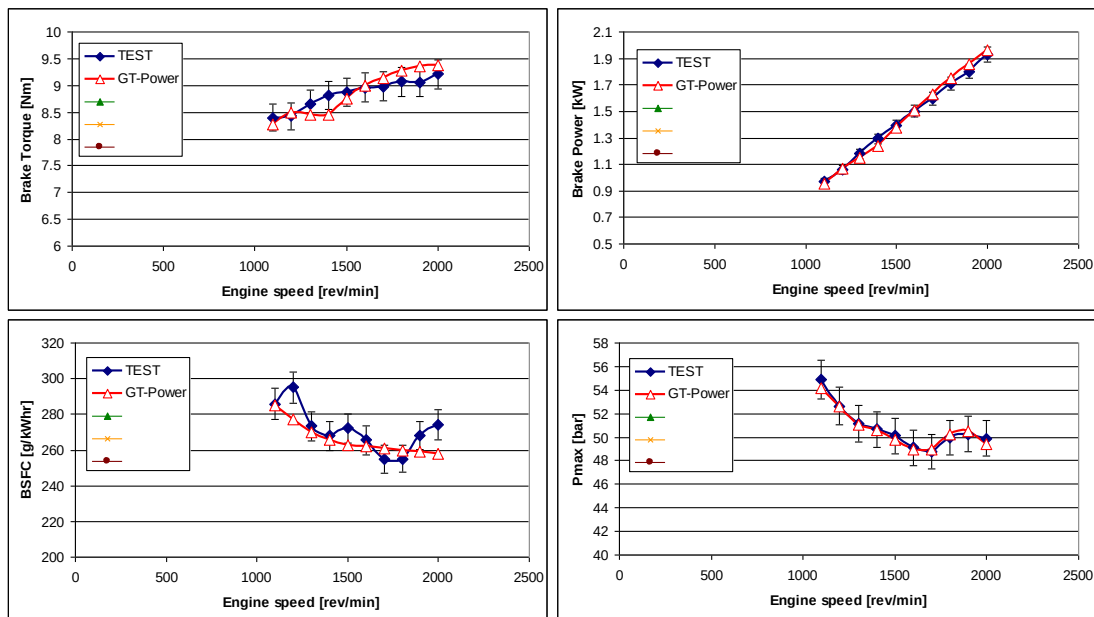


Figure 3.1. Comparison of GT-Power analytical results and measured test data.

Table 3.1. DoE variables and ranges.

Variable	Description	Units	min	max
BS_RATIO	Bore/stroke ratio	-	0.5	1
DISP	Displacement	cc	50	100
EVC	Exhaust valve closing phase	deg aTDC	450	470
EDUR	Exhaust cam duration scale	-	0.9	1.1
IVO	Intake valve opening phase	deg aTDC	260	300
IDUR	Intake cam duration scale	-	0.9	1.1
CR	Compression ratio	-	13	18
EGR	Exhaust gas recirculation rate	-	0	0.3
PINLET	Intake boost pressure	bar abs.	1	1.5
TINLET	Intake gas temperature	degC	40	100

Optimum engine configuration was selected by fitting neural network models to salient parameters and optimizing for maximum BTE. Sensitivity was performed on the optimized design including combustion parameters not included in initial DoE study. An initial design concept based on the recommendation from the optimization study was analyzed including with and without various levels of Thermal Barrier Coating (TBC). Figure 3.2 shows predicted BTE at each step of the process. Table 3.2 shows the recommended engine configuration with review and parameter importance ranking. This satisfies Milestone M3.1, "Design review of engine that meets FOA targets and is approved by expert panel. GT-Power design with engine design parameters selected such as bore, stroke, displacement, engine speed and load, identification of heat loss reduction required from TBCs, EGR (or residual mass) requirements, and intake temperature and pressure."

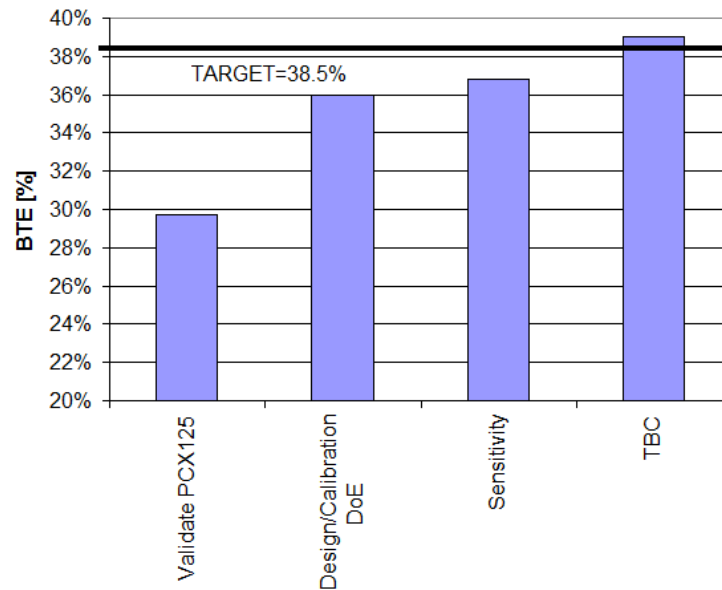


Figure 3.2. Analytical BTE development.

Table 3.2. Final optimized design for combustion and air system development.

Variable	Value	Comments	Importance
Displacement	70cc	Best tradeoff for achieving CR.	1
Bore	40mm	For optimum 0.7B/S ratio.	1
Stroke	56mm	For optimum 0.7B/S ratio.	1
CR	18:1	Anticipated practical limit.	2
Valve timings	IVO=273deg aTDC, EVC=460deg aTDC	Advanced intake may require valve cutouts.	2
Valve lift	Int=6.01mm, Exh=6.38mm	Low velocity and acceleration for durability.	3
Valve duration	Int=282deg, Exh=229deg	Overexpanded	2
Valve diameter	Int=16.2mm, Exh=13.6mm	Feasible design with offset, centrally directed plug.	3
Intake geom	Port dia matched to valve	Will be refined during air-system development.	2
Exhaust geom	Port dia matched to valve	Will be refined during air-system development.	2
RPM	1800rev/min	Required for target power.	2
BMEP	10bar	Point of diminishing mechanical efficiency gains.	2
Pint	1.5bar abs.	Demonstrated analytically.	1
Tint	50deg C	Will require cooling.	3
EGR	30%	Maximum assumed possible.	1
CA50	12-14deg aTDC	Optimum with assumed 20deg burn duration.	2
BDUR	20deg	High tumble port and/or SA-HCCI.	1
TBC specs	0.2K+0.36D	In-development spec.	1
Boost system	TBD		1

Task 4 – Provide Benchmark Current-State Engine Models and Performance Data

Summary

The objective of this task is to review available Briggs & Stratton engine models and performance test data to identify appropriate benchmark, current-state engine for multi-dimensional engine model development.

Delivery of benchmark validated GT-Power model(s) and test data for the Briggs and Stratton 126cc single cylinder engine run on gasoline. Supply all required input data for initial CFD modeling of the same engine at engine speeds similar to those to be used on in the final engine (e.g., 1200, 1800 rpm).

Details

Data was provided for two current production air cooled vertical crankshaft Briggs & Stratton engines. Both a 125cc and 190cc data sets were provided, including the following information:

Engine Geometry
Combustion data
Port flow data
Valve lift and timing information
Brake power data
Exhaust emissions data

Combustion, power, and emissions operational data at 2400, 3060, and 3600 rpm was provided.

It was determined that baseline operational data for an engine closer to a configuration appropriate for the GENSETS engine was also desirable for initial model validation. The team reviewed available engine configurations and settled on the engine from the Honda PCX125 motor scooter as shown in Table 4-1. This 125cc, water cooled, high power density, high efficiency design was adapted to mount to a Briggs & Stratton dynamometer, as shown in Figures 4.1 and 4.2. Instrumentation was added to acquire performance, combustion, and emissions data.

Table 4.1. Engine options closer to configuration for GENSET engine.

Engine/Application	Bore (mm)	Stroke (mm)	CR	Vd (cc)	Valvetrain
Honda CRF150R	66	43.7	11.7	149.5	4 valve
Honda PCX125	52	57.9	10.7	124.9	2 valve
Honda Ruckus	38	44.0	11.9	49.4	2 valve

Wide open throttle (WOT) data was acquired from 1100 to 2000 rpm in increments of 100 rpm. Additionally, cylinder head port flow, in-cylinder swirl, and in-cylinder tumble data was acquired using

pecially developed fixtures and the Briggs & Stratton in-house flow bench as shown in Figure 4.3. Port flow data is used in the GT-Power modeling process.

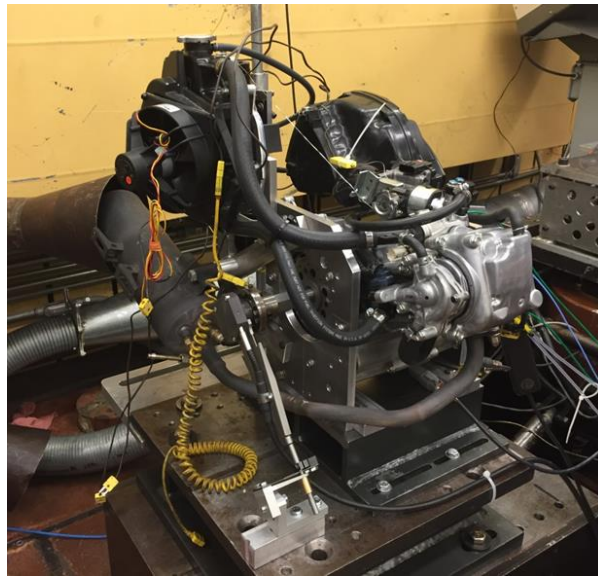


Figure 4.1. PCX125 Engine Adapted to Briggs & Stratton Dyno.

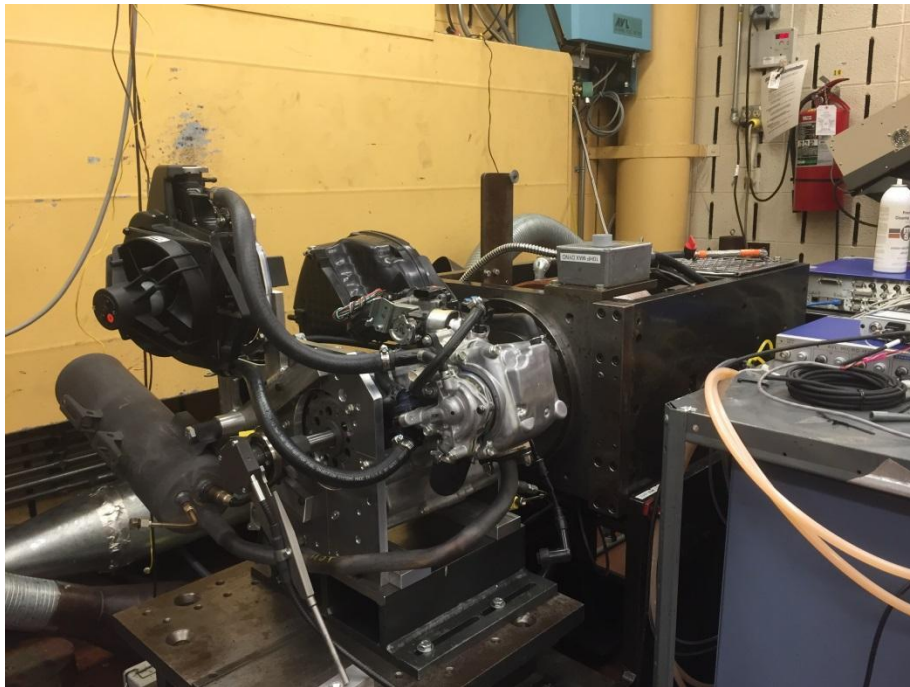


Figure 4.2. PCX125 Engine Adapted to Briggs & Stratton Dyno.



Figure 4.3. PCX125 Cylinder Head Flow, Swirl, Tumble Fixture.

Conclusions

Benchmark models and test data was supplied for 2 Briggs & Stratton engines to allow initial CFD modeling. Additionally, test data and modeling information was acquired and provided for the Honda PCX125 engine, an engine closer in configuration to the proposed GENSETS engine. Task 4 was successfully completed.

Task 5 - Initial CFD Model Validation

Summary

A CFD model was created for the Briggs and Stratton W14 engine and the model was validated at 3 engine speeds at full load (i.e., wide-open-throttle). The model was able to predict the peak cylinder pressure, Indicated Mean Effective Pressure (IMEP), combustion efficiency, and NOx emissions within the prescribed tolerance of the SOPO. This was completed by end of project Q1, satisfying Milestone M5.1, "CFD model capable of reproducing the measured emissions and performance characteristics of a base engine was completed."

Details

For Task 5, Data from Task 4 was used to validate the CFD model used to optimize the combustion system. The outcome of this task is a validated combustion model that can be used to design the SA-HCCI combustion needed to achieve 40% electrical conversion efficiency. Targets were prediction of peak cylinder pressure within 10%, IMEP within 5%, combustion efficiency within 3%, NOx emissions within 20%.

Test data for the CFD Model Validation was supplied by team member Briggs and Stratton Corp. The data set from their W14 engine consists of 3 speed points and is summarized in Table 5.1. The engine has a bore and stroke of 6.83 cm and 5.18 cm respectively which yield a total displacement of 190 cm³.

Table 5.1. Test data for CFD model validation.

RPM	Throttle	Spark Timing (°BTDC)	AFR	BMEP (bar)
3600	100%	20	12.63	6.33
3060	100%	20	12.59	7.47
2400	100%	20	12.46	7.59

The CFD model validation began with grid construction based on the supplied CAD data as shown on the left in Figure 5.1. For the purposes of CFD engine modeling, the intake and exhaust ducts are truncated to limit computational time needed to establish the pressure transients in the duct system and allow the application of boundary conditions supplied by the 1-D GT-Power model. For example, the boundary conditions for the intake port were applied at the orange cross section as shown on the left side of Figure 5.1. In addition to the duct boundary conditions, the GT-Power model supplied the average thermodynamic conditions in each of the three primary volumes: intake and exhaust port and cylinder.

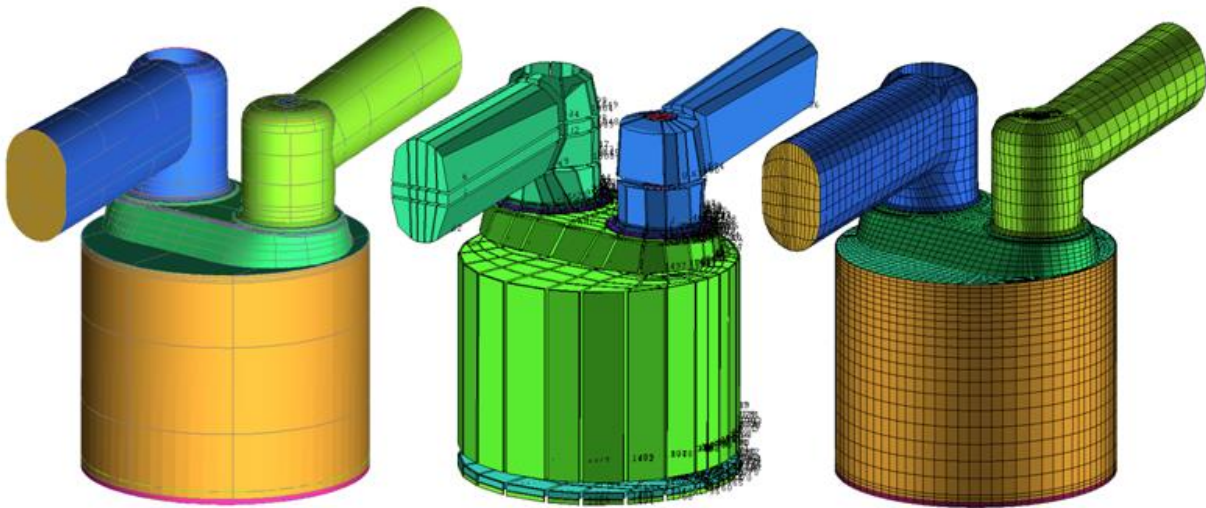


Figure 5.1. Geometry blocking and mesh for W14 engine.

Figure 5.2 shows the CFD simulation at Bottom Dead Center (BDC) during the exhaust stroke. High velocities in the exhaust duct during the blow down period can be seen with the length of velocity vectors (shown as black arrows). The different temperatures in each of the respective regions (intake port, exhaust port and cylinder) can be seen. The simulations ran from Exhaust Valve Opening (EVO) to Intake Valve Closing (IVC) and provided initial conditions for the closed valve (compression and expansion) portion of the cycle.

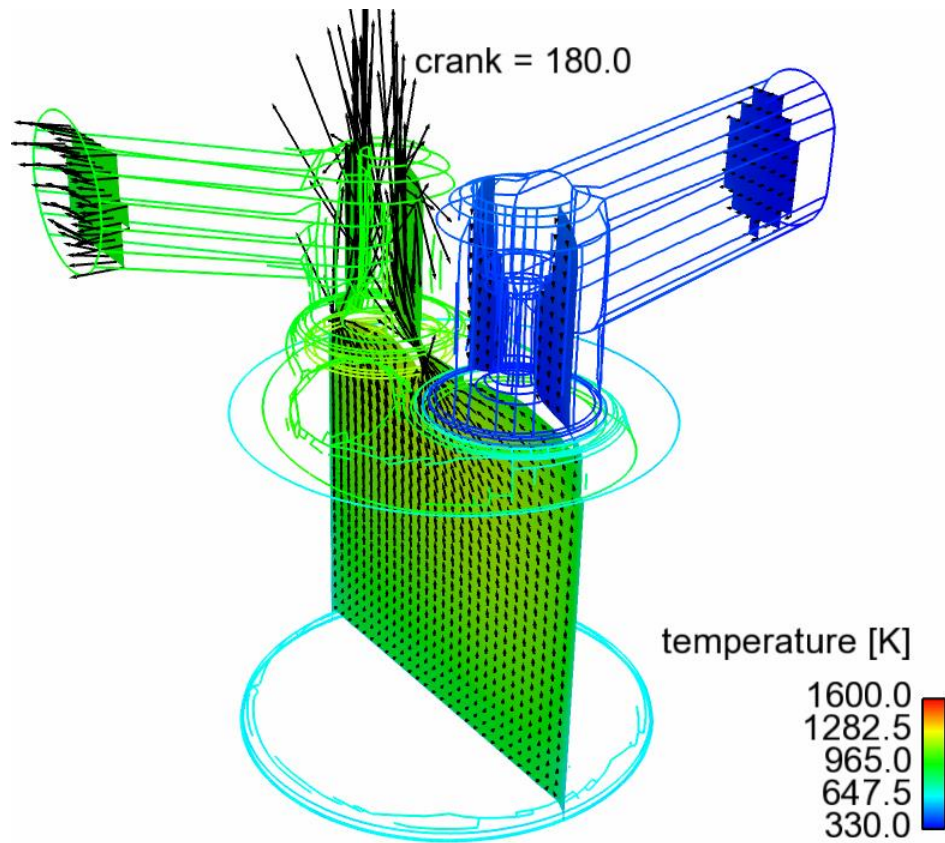


Figure 5.2. W14 engine at BDC during the exhaust stroke. Contours of temperature shown.

The results of the combustion simulations and comparisons to the test data are shown in Figure 5.3 and Table 5.2. Figure 5.3 shows the cylinder pressure and apparent heat release rate for each of the three cases starting with the 2400 rpm case at the top. The peak cylinder pressure from the model is within 10% of the measured value for all 3 cases. Comparisons between the model results and test data for peak cylinder pressure, Indicated Mean Effective Pressure (IMEP), combustion efficiency, and NO_x emissions are shown in Table 5.2. The experimental emissions data was available for the 3060 rpm case only.

Figure 5.4 shows the error of the model results relative to the allowable error, per the SOPO and Milestone M5.1. The model results are well within the targets relative to the measured engine data.

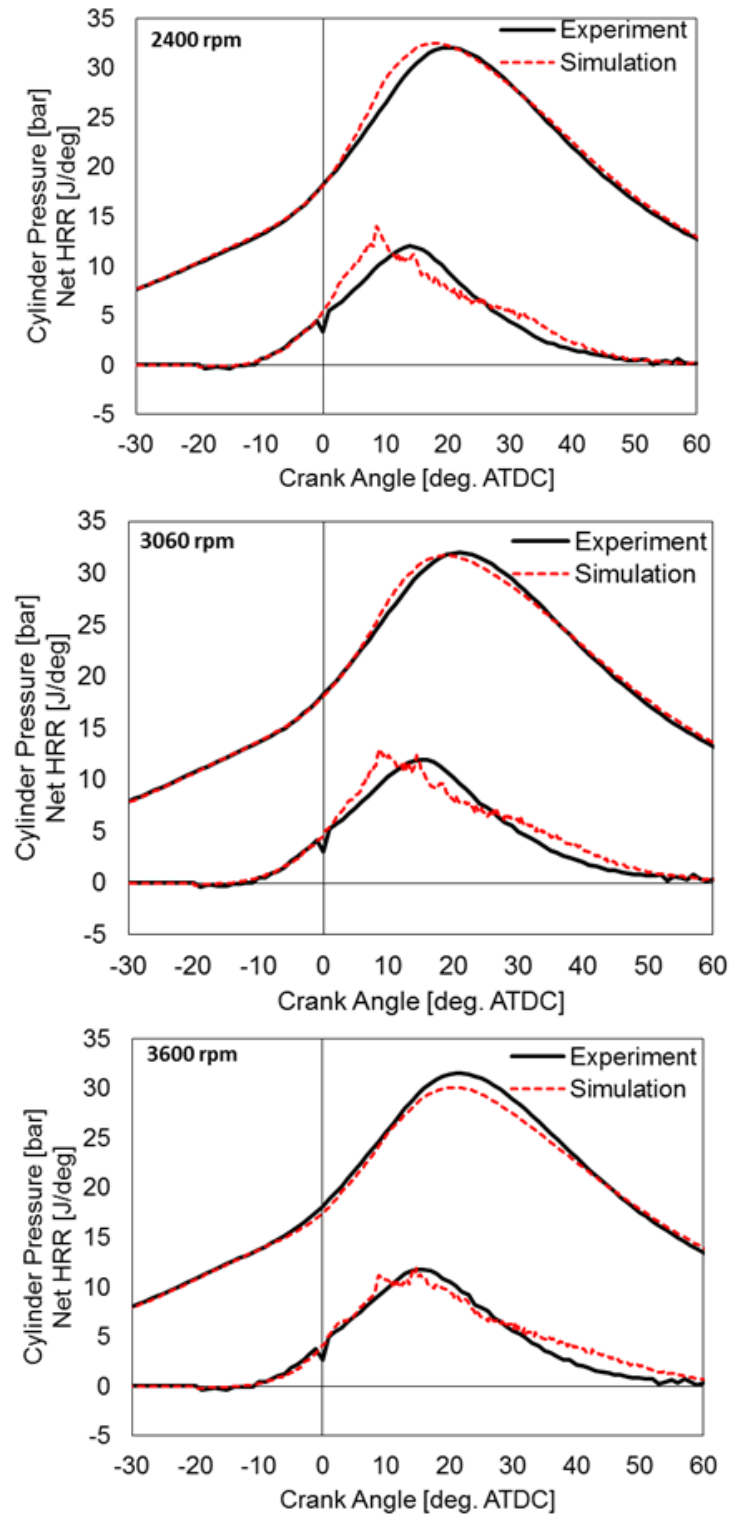


Figure 5.3. Cylinder pressure and net heat release rate for the 2400, 3060 and 3600 rpm cases (starting from top).

Table 5.2. Model results compared to test data for W14 engine.

	speed	IMEP gross	IMEP net	Pmax	comb. eff.	NOx
	rpm	bar	bar	bar	%	g/kW-hr
measured	2400	8.13	7.93	32.04		
	3060	8.40	8.03	31.96	77.98	3.74
	3600	8.50	7.97	31.48		
model	2400	8.35	8.07	32.44	77.40	3.81
	3060	8.61	8.25	31.70	77.32	3.68
	3600	8.61	8.19	30.07	76.45	3.13

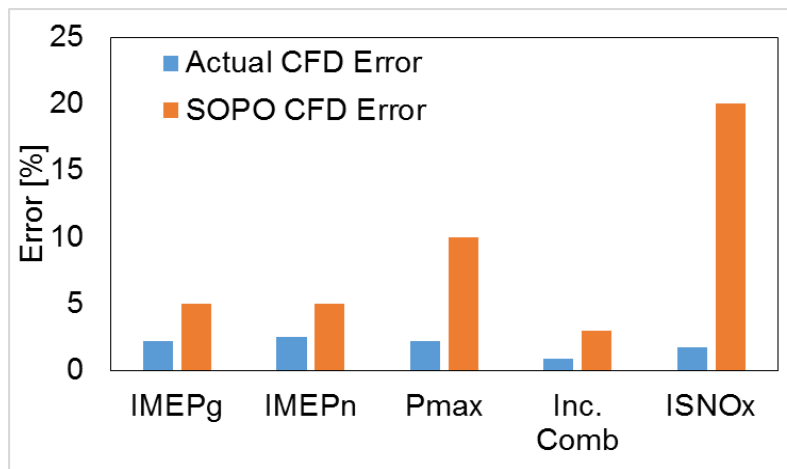


Figure 5.4. Error in the CFD model relative to the allowable error.

Task 6 – Conceptual Evaluation of Multiple Engine Concepts

Summary

The objective of this task is to use the findings of initial 1-D modeling (Task 3) to develop multiple natural gas engine concepts. Concepts will consider the performance modeling as well and materials and manufacturing limitations. At least one engine concept will be designed to sufficient fidelity to allow continued analysis via 1-D and 3-D analytical methods with natural gas as a fuel. Engine will feature a high CR, high PCP capability, minimized top piston ring land crevice volume, and low friction.

Results

The Honda PCX125 engine was used for as a basis for initial engine modeling and concept optimization efforts during Task 3. Briggs & Stratton performed combustion analysis on this engine to gather data to allow WERC to tune and validate GT-Power simulations (Task 4). The next steps were to perform a broad DoE of design and calibration parameters for engine concepts that lead to the results shown in Table 6.1.

Table 6.1. Engine Design Parameters.

	PCX engine	Concept #1	Concept #2	Concept #3
Displacement	125 cc	71 cc	70 cc	70 cc
Bore	52.4 mm	39.5	40 mm	40 mm
Stroke	57.9 mm	57.9	56 mm	56 mm
Bore/Stroke Ratio	0.91	0.68	0.71	0.71
CR	11:1	17:1	18:1	18:1
Included Valve Angle	43°	43°	43°	29°
Intake Seat Dia.	25.4 mm	17.0 mm	14.5 mm	14.5 mm
Exhaust Seat Dia.	20.2 mm	12.5 mm	12.5 mm	12.5 mm

Engine Concept #1 parameters are based on a combination of off-the-shelf production parts from Honda motor scooters - the PCX125 engine used to validate modeling and a cylinder head and bore from the much smaller Metro engine. Engine Concept #2 parameters were the outcome from Task 3 – Engine Modeling Optimization. As can be seen from these parameters, the adapted production parts are very close to the results from the GT-Power optimization simulations. Because adapting off-the-shelf parts can expedite critical early performance development, efforts were initiated to design the components necessary for Engine Concept #1.

The combination of the Honda PCX125 crankcase, crankshaft, and connecting rod with the smaller Honda 50cc Metro piston and cylinder head provided this early hardware opportunity. To execute this combination, a special adapting cylinder configuration was required. Figures 6.1 and 6.2 illustrate the required design elements.

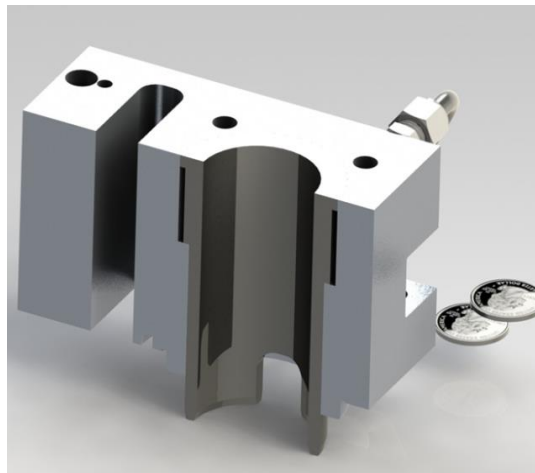


Figure 6.1. Special Adapting Cylinder (CAD Model).

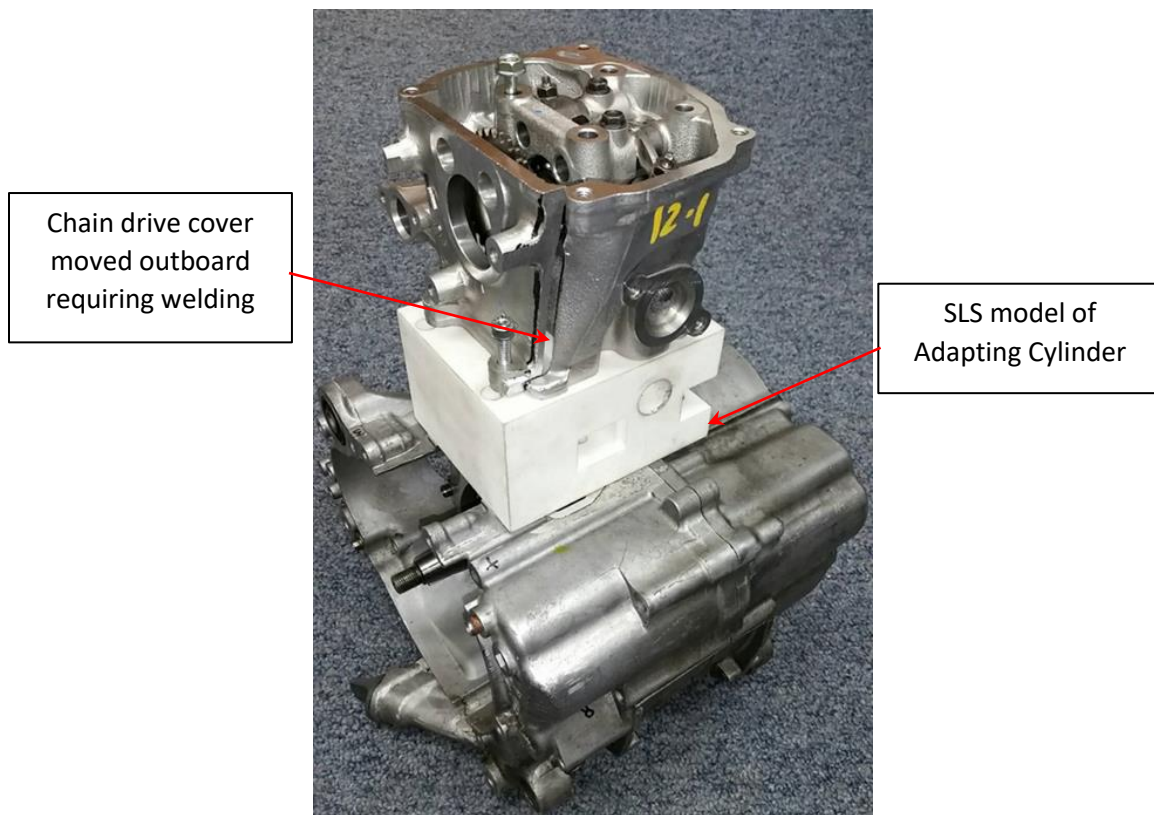


Figure 6.2. Engine Concept #1 Mock-Up.

Because the PCX125 cam drive system is further outboard than that of the Metro cylinder head, cutting and welding of the assembly for chain drive alignment was required. Figure 6.2 shows the chain drive cover in the required outboard location on the Metro head. The use of Alumite (Weldcote Metals) material allowed a low temperature welding process, reducing the likelihood of cylinder head distortion. Additionally, a custom piston was required for the Concept #1 engine. First, the PCX125 connecting rod has a piston pin bore of 13mm versus 10mm on the Metro engine. The larger size is preferred for piston pin loading due to the high compression ratio and predicted cylinder pressure at the engine operating point. Secondly, the piston crown shape required adjustment for valve sizes and desired compression ratio. A source for prototype pistons was found with a short 4-5 week delivery. Further study showed that the Metro engine valve timing was very close to the predicted optimized timing from the GT-Power simulations, see Figure 6.3. Cylinder head flow testing of the Metro head was accomplished and compared to that initially used in the optimization study (PCX125 results). These results and comparisons are shown in Figure 6.4 (intake) and 6.5 (exhaust). Metro head flow results were then used to improve the fidelity of the GT-Power modeling of the Concept #1 engine.

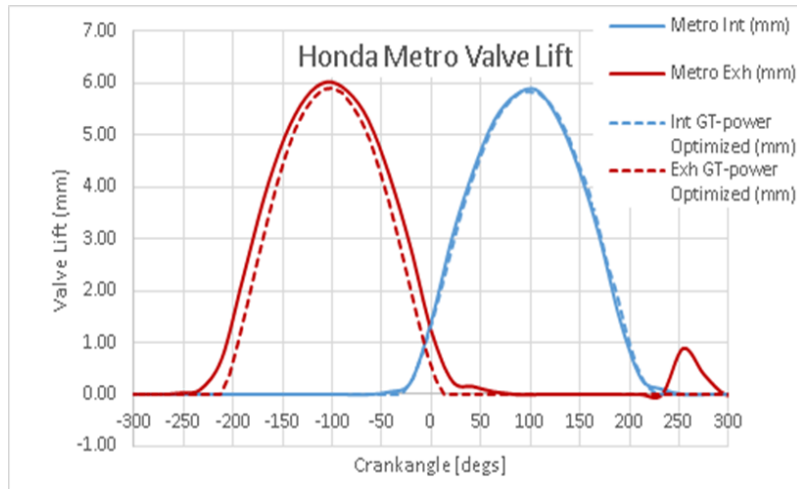


Figure 6.3. Valve lift and timing of Metro engine similar to optimized model.

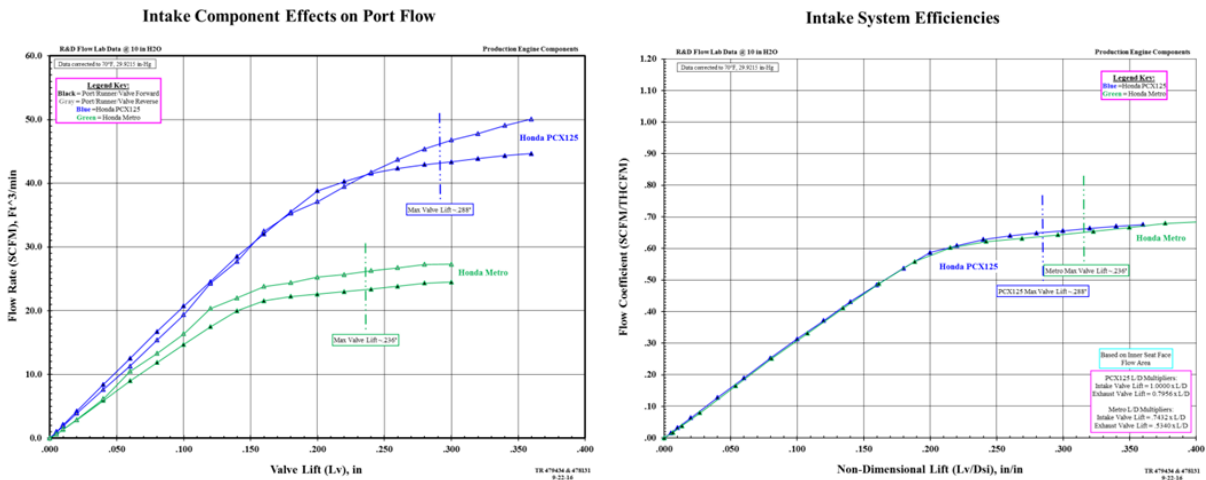


Figure 6.4. PCX125 and Metro cylinder head intake port flow.

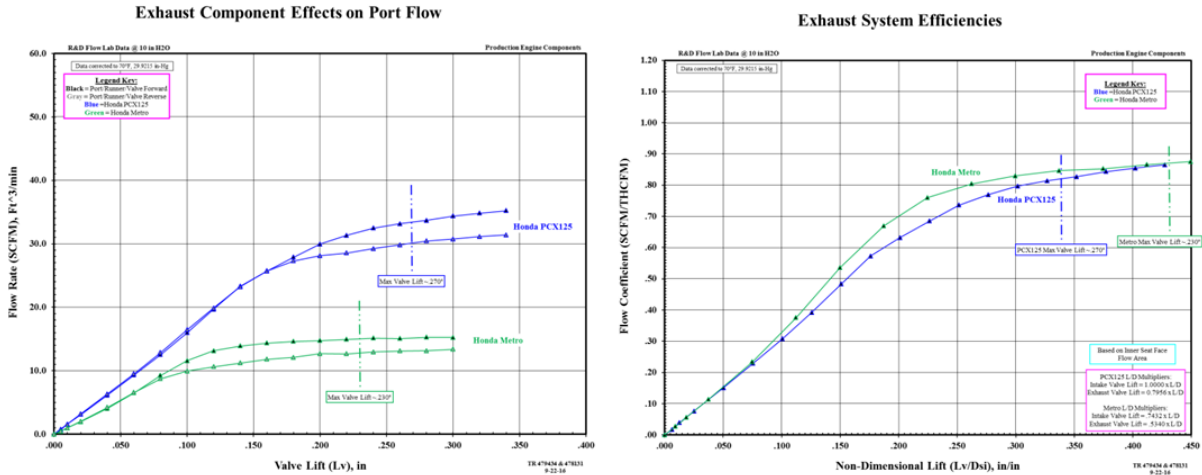


Figure 6.5. PCX and Metro cylinder head exhaust port flow.

Concept models were completed to allow continuing 1-D and 3-D modeling comparisons by WERC. The cylinder head porting was schemed based on the initial Task 3 modeling optimization results. Model based valve diameters of 14.5mm intake, 12.5mm exhaust were utilized. Included valve angles designs of 43° (Concept #2) and 29° (Concept #3) were developed to allow a comparison of flow efficiency and charge motion. Images of these features are shown in Figures 6.6. Combustion chamber design and volume optimization is shown in Figure 6.7.

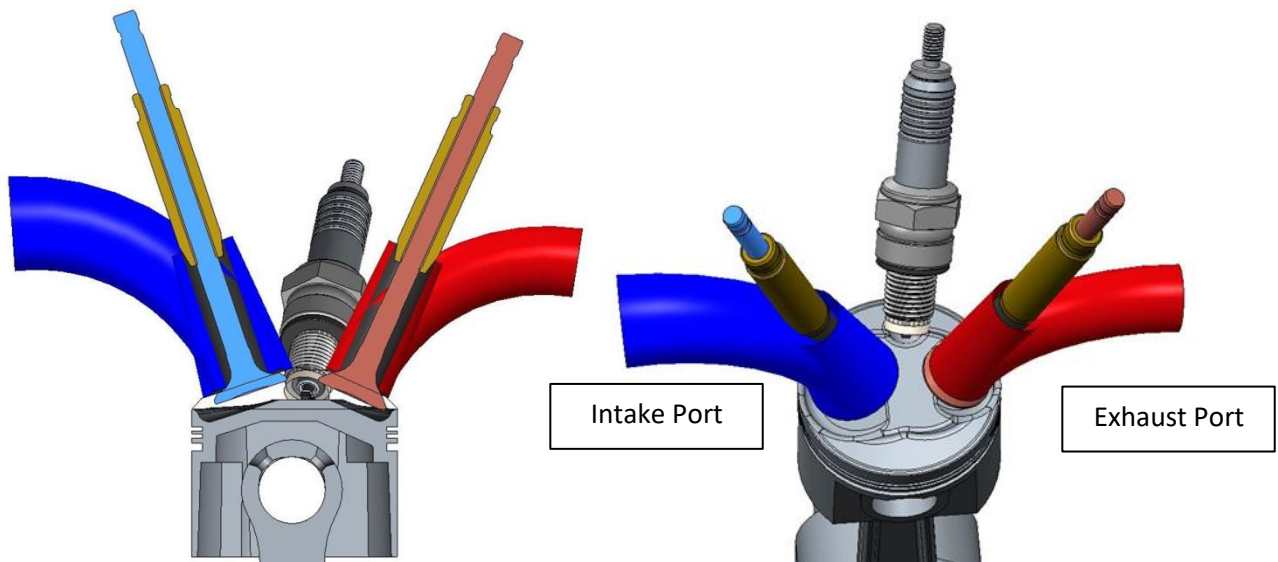


Figure 6.6. Engine Concept #2 head design.

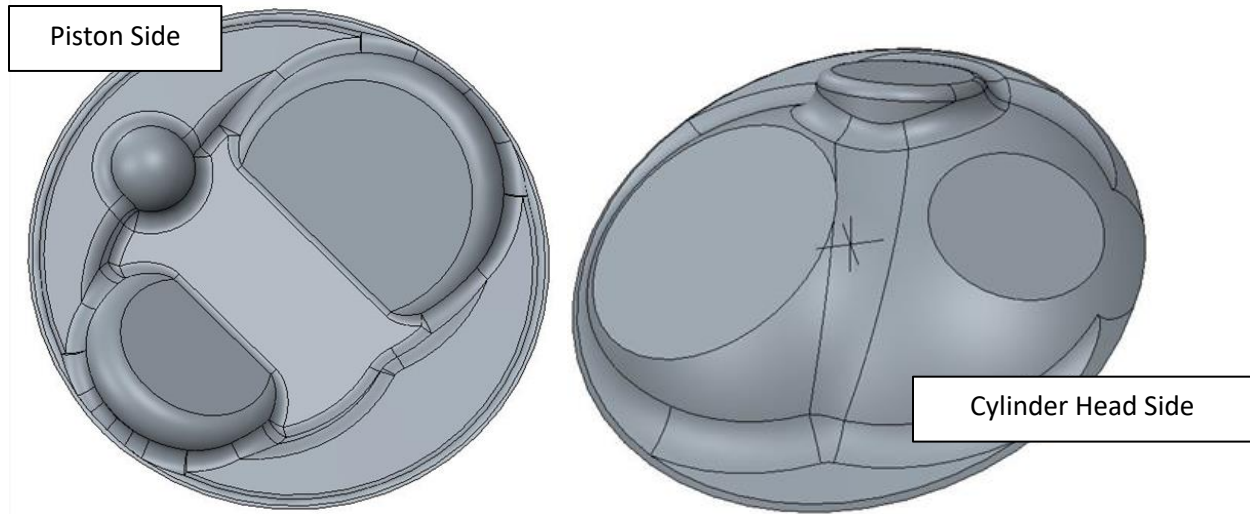


Figure 6.7. Engine Concept #2 combustion chamber design Conclusions.

Three engine concepts were designed in sufficient detail to allow continued analytical modeling and comparisons using 1-D and 3-D methods (WERC effort). The primary difference in these configurations were included valve angle and valve sizes. Task 6 was successfully completed.

Task 7 - Directional CFD Modeling of Concept Engines

Summary

Three design concepts analyzed with full cycle CFD.

- 1) Concept1 based on off-the-shelf production parts.
- 2) Concept2 based on GTPower optimized design with 43deg included valve angle.
- 3) Concept3 based on GTPower optimized design with 29deg included valve angle.

Two versions of Concepts2 and 3 investigated, Version a (1mm clearance) and Version b (2mm clearance). Concept3a had the best performance. Version a had better performance than Version b. Concept2a showed potential to achieve program targets. This was completed by end of project Q4, satisfying Milestone M7.1, "Down selected engine technology / architecture to be used in the phase I prototype engine build to achieve 40% electrical conversion efficiency. Comparative metrics for the natural gas engines will include volumetric efficiency, combustion duration, heat transfer, combustion efficiency, and peak cylinder pressure."

Results

In Task 7, up to 3 advanced CHP natural gas engine concepts designed in Task 6 will be analyzed with CFD. The outcome of this task will be combustion modeling results that provide directional guidance in selecting the appropriate advanced CHP engine architecture for achieving 40% electrical conversion efficiency.

The first step in this process was to build and validate a CFD model of the Honda Ecowill engine run with natural gas. The purpose of this step is to establish combustion model settings to be used in analysis of engine concepts developed for this program.

ARPAe arranged for engine testing and scanning of head and piston surface. Test data and scan results were provided by Professor Chris Hagen of Oregon State University (OSU). The scanned data was turned into a closed surface CFD model as shown in Figure 7.1.

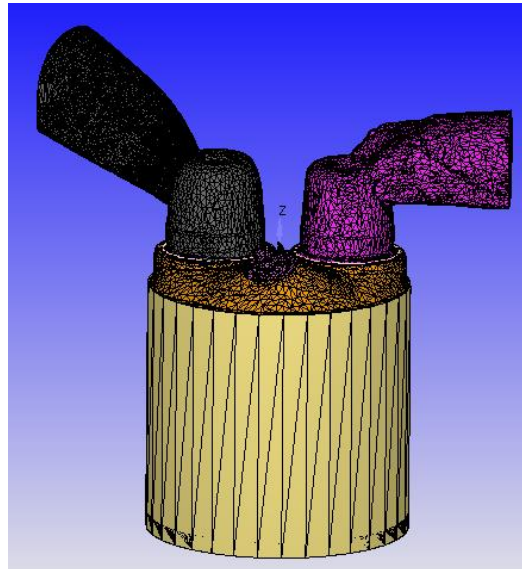


Figure 7.1. Honda Ecowill CFD model geometry.

Additionally, OSU measured key engine geometry features. Table 7.1 summarizes the measured engine geometry.

Table 7.1. Measured engine geometry.

Dimension	Measured at OSU
Bore [mm]	62
Stroke [mm]	54
Connecting Rod [mm]	84
Head Gasket Thickness [mm]	0.7
Piston to Deck Height [mm]	0.25
Head Vol [cm ³]	12.5
Head Gasket Vol [cm ³]	2.11
Piston to Deck Vol [cm ³]	0.77
Vtdc [cm ³]	15.38
Swept Vol [cm ³]	163
CR	11.60

CFD model was setup based on supplied information including measured/scanned geometry and intake/exhaust boundary conditions. Additionally, experience based wall temperatures and model settings were implemented. Several full cycle simulations have been performed to stabilize initial conditions. Results in Figure 7.2 show CFD match compared to engine test data.

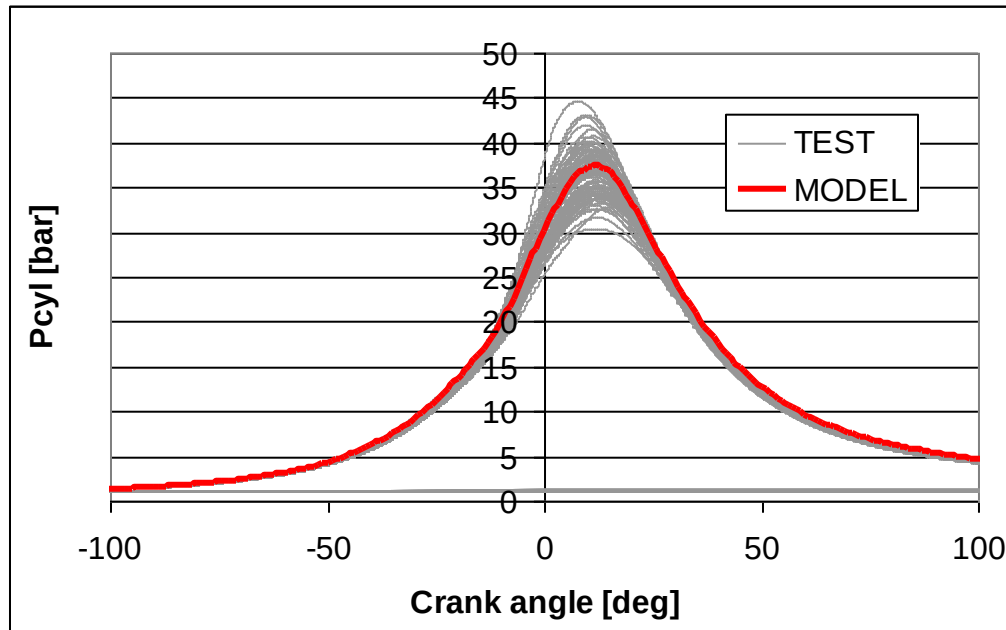


Figure 7.2. Honda Ecovill CFD model cylinder pressure compared to test.

Combustion model settings used for final match to Ecovill data were fixed for analysis of concept designs.

Three engine concepts were analyzed where designs were based on parameters set from Task 3-Engine Modeling Optimization.

- 1) Concept1: Adapted Off-the-Shelf production parts used for expediting early performance development.
- 2) Concept2: Parameters set from Task 3 – Engine Modeling Optimization. Valve included angle (angle between intake and exhaust valve stem centerlines) matched to Concept1.
- 3) Concept3: is based on Concept2 with change in valve included angle.

For Concepts 2 and 3, 2 versions simulated.

- 1) Version a. 1mm minimum valve/piston clearance.
- 2) Version b. 2mm minimum valve/piston clearance.

Key concept dimensions are summarized in Table 7.2 and shown in Figure 7.3.

Table 7.2. Engine concept key dimension summary.

	Concept1	Concept 2	Concept3
Displacement	71 cc	70 cc	70 cc
Bore	39.5	40 mm	40 mm
Stroke	57.9	56 mm	56 mm
Bore/Stroke Ratio	0.68	0.71	0.71
CR	17.5:1	18:1	18:1
Intake Seat Dia.	17.0 mm	14.5 mm	14.5 mm
Exhaust Seat Dia.	12.5 mm	12.5 mm	12.5 mm
Valve included angle	43deg	43deg	29deg

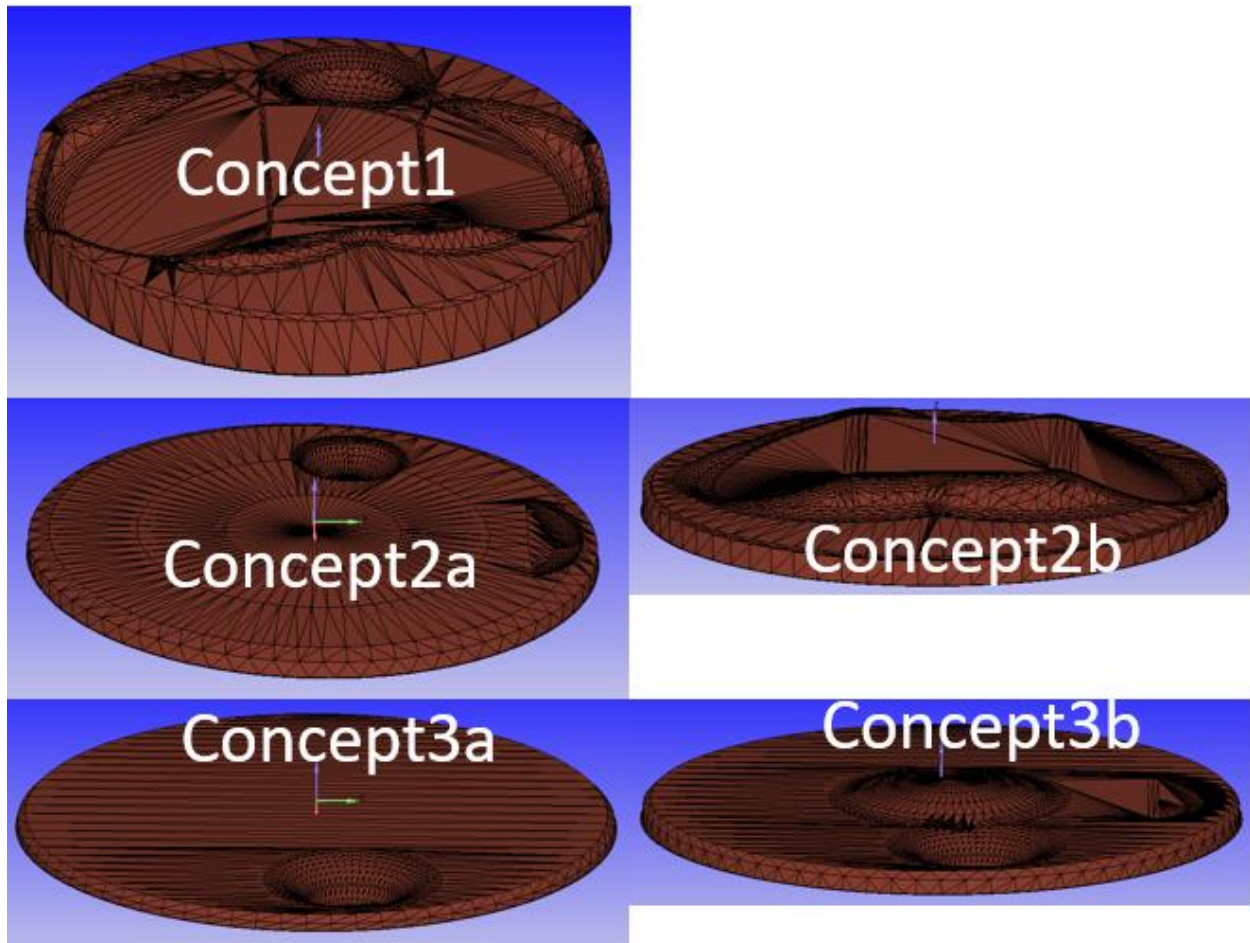


Figure 7.3. Pistons for CFD study.

Performance results can be seen in Table 7.3 where concepts are ordered from lowest (left) to highest (right) indicated thermal efficiency (ITE).

Table 7.3. Design performance summary at -10aTDC spark timing ordered from worst ITE (leftmost) to best ITE (rightmost).

	CONCEPT1	CONCEPT2b	CONCEPT2a	CONCEPT3b	CONCEPT3a
Spark [deg aTDC]	-10	-10	-10	-10	-10
Air Volef [%]	47.95	48.99	49.08	49.75	49.38
10-90dur [deg]	32.78	32.86	30.02	29.25	28.91
Heat Transfer [kW]	0.4490	0.4512	0.4085	0.4293	0.4219
Pmax [bar]	74.39	78.86	77.36	80.51	86.58
ITE [%]	40.21	40.50	41.49	42.53	42.90
Comb. Eff. [%]	92.35	92.13	92.51	93.71	92.90
Spark TKE [m ² /s ²]	1.2571	1.0645	1.1272	1.2845	1.4000

Concept3, particularly Concept3a had the best performance. Concept2a with optimal spark timing appears capable of achieving program performance targets. This will be useful as the design matures and things like manufacturing tolerances and valve train layout are more fully understood. This satisfies Milestone M7.1, Down selected engine technology / architecture to be used in the phase I prototype engine build to achieve 40% electrical conversion efficiency. Comparative metrics for the natural gas engines will include volumetric efficiency, combustion duration, heat transfer, combustion efficiency, and peak cylinder pressure.

Task 8 - Development and Testing of Advanced Boosting System

Summary

The objective of this task is to develop and test intake air boosting systems that can enable the high brake mean effective pressure (BMEP) operation necessary to achieve 40% electrical efficiency. Initial efforts focused on testing a prototype pressure wave supercharger (PWS) on the air handling bench developed in year 1. Testing demonstrated the potential to meet the design target of 1.3 bar absolute intake pressure at the flow rates suitable for 1 kW operation. The efforts in Q5-6 focused on testing the prototype pressure wave supercharger (PWS) on the air handling bench developed in year 1. Abradable coatings were added to avoid short-circuiting from the exhaust stream to the intake stream and failure modes of the bearing system were investigated. The Q7-8 effort tested a small production turbocharger and assessed alternative approaches based on supplier data to complete Milestone M8.1: Bench Testing of Three Boosting Concepts. Additional efforts focused on static and dynamic evaluation of a membrane-based charger. Static testing showed that a material with properties similar to polycarbonate will produce suitable membrane movement to build boost. The polycarbonate membrane was installed on the engine test stand and shown to build a closed pressure of 1.06 bar. Analysis and testing included tuned intake and exhaust lengths to enable evaluation of a low-cost solution. The results demonstrated the potential of the drum charger with an aluminum membrane to produce a pressure of 0.05 to 0.07 bar. However, concurrent engine testing has shown that peak efficiency is achieved at higher engine speeds than suggested by the initial modeling effort. At these higher speeds, intake boosting is not expected to provide a significant advantage; accordingly, further efforts focused on tuned intake and

exhaust lengths to enable the evaluation of a low-cost solution. Table 8.1 shows the summary of the intake pressure boosting approaches evaluated and/or tested.

Table 8.1. Boosting system progress and selection matrix.

Approach	Status	Achieves program targets	Notes
Pressure Wave Supercharger	Design, fabrication, and testing complete	Yes	System is capable of achieving program targets at a rotor speed near 7000 rev/min. System is sensitive to rotor clearance, which may make mass manufacturing challenging.
RHB31 Turbocharger	Testing complete	No	Flowrates are too low to achieve boost targets. No further testing is expected.
Electric Supercharger	Evaluation complete	Yes (supplier based)	Boost can be achieved; however, parasitic losses are expected to outweigh benefit.
Membrane Based Charger	Design and fabrication complete	TBD	Supplier testing shows potential to achieve program targets
Intake Runner Tuning	Testing in process with Concept 1 engine	TBD	Focus is to minimize pumping losses from gas exchange.

Details

A selection matrix was used to identify and evaluate six pathways to provide small engine intake air boosting. The selection matrix is shown in Table 8.2. The systems identified were a turbocharger operating using exhaust enthalpy, a mechanical supercharger driven off the crankshaft, an electric supercharger driven using an external electric motor, a pressure wave supercharger (PWS) [³,⁴], tuning of the intake and exhaust system, and crankcase boosting. Team members were polled and each system was ranked in five categories:

- 1) Potential Boost. The projected ability to provide intake boosting to at least 1.3 bar;
- 2) Manufacturing. Expected challenges in fabricating components including component size and potential tolerance challenges;
- 3) Cost. Cost of materials and manufacturing processes;
- 4) Parasitic loss. Energy input required to drive the system to supply intake air boosting. This includes any added pressure drop and losses associated with additional rotating components;

³ Pohorelsky, L., Zak, Z., Macek, J., and Vitek, O., "Study of Pressure Wave Supercharger Potential using a 1-D and a 0-D Approach," *SAE Int. J. Engines* 4(1):1331-1353, 2011, doi:10.4271/2011-01-1143.

⁴ Mataczynski, M., Hoke, J., Paxson, D., and Polanka, M., "Design, Simulation, and Testing of a Pressure Wave Supercharger for a Small Internal Combustion Engine," SAE Technical Paper 2014-01-2136, 2014, doi:10.4271/2014-01-2136.

- 5) Energy Recovery Potential. The potential to recover high availability exhaust energy and convert it to electrical energy.

Each system was ranked from best (1) to worst (6) in each category and the final ranking was determined from the weighted sum. The weighting factor for potential boost was set at 2 and the weighting factors for all other categories were unity. The potential boost was weighted highest because, without the ability to provide the necessary boost, the other categories are not relevant.

Table 8.2. Boost system selection matrix. 1=best rating and 6=worst rating.

	Potential Boost	Manufacturing	Cost	Parasitic Loss	Energy Recovery Potential	Weighted Sum	Rank
Turbocharger	3.65	5.7	4.3	2.0	1.7	21.0	4
Mechanical Supercharger	1	3.0	3.3	5.0	5.3	18.7	2
Electric Supercharger	1.35	4.0	5.3	5.3	5.7	23.0	5
Pressure Wave Supercharger	3	2.3	3.0	2.0	1.3	14.7	1
Intake / Exh. Tuning	5.35	1.3	1.0	2.0	3.7	18.7	2
Crankcase Boosting	4.35	3.3	3.0	4.0	4.0	23.0	5

Pressure Wave Supercharger Testing

The PWS performed best in the selection matrix, but is less mature than competing systems (intake and exhaust tuning and mechanical supercharging). Accordingly, the initial design effort will focus on the PWS with a risk mitigation strategy of a mechanical supercharger or intake/exhaust tuning.

The initial design of the PWS system was performed by developing a 1-D model in GT-Power^[5]. The model was exercised to determine important design variables. It was found that the intake pressure was sensitive to rotor speed and phasing between the air and exhaust ports. The amount of flow-through air from the air inlet to the exhaust outlet was found to be sensitive to the exhaust outlet opening duration. An optimization was then performed to maximize the brake mean effective pressure (BMEP). Figure 8.1 shows the optimization results. After 184 iterations of the GT-Power model, the optimization converged to a solution providing over 1.5 bar intake pressure with negligible exhaust residuals. The final solution used a rotor speed of 4289 rev/min, a phasing of -9.7°, and an exhaust opening port width of 21°.

⁵ GT-Power. Vers. 7.5., Gamma Technologies, 2016, Computer software.

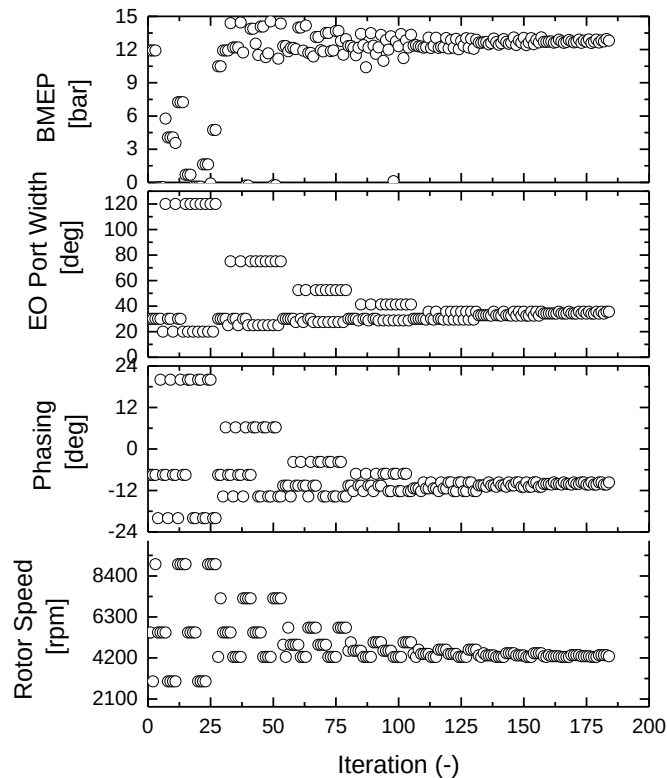
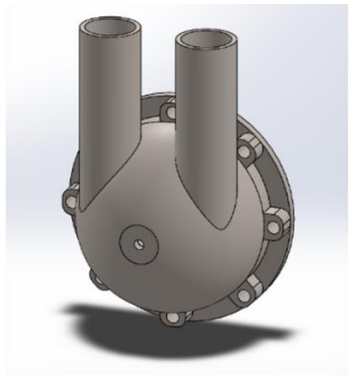


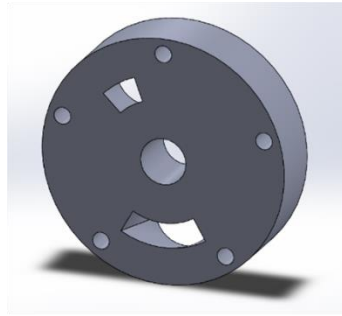
Figure 8.1. Optimization process of the initial pressure wave supercharger design.

With a preliminary design complete, the effort focused on manufacturing the pressure wave supercharger to provide data for model validation and verification. Figure 8.2 shows the design evolution of the pressure wave supercharger end housing beginning with a design based on the work of Mataczynski ^[6] (Design 1.1). It was found that much of the manufacturing complexity of Mataczynski's design was based on the need to include geometry in the end housing to enable the PWS to perform well over a range of engine speeds. The CHP system will operate at a single engine speed; therefore, the manufacturing complexity can be reduced by removing the complex housing geometry (see Design 1.2). It is expected that the exhaust side temperatures will reach values greater than 500 K. Accordingly, water cooling passages were added to ensure bearing durability (Design 1.3).

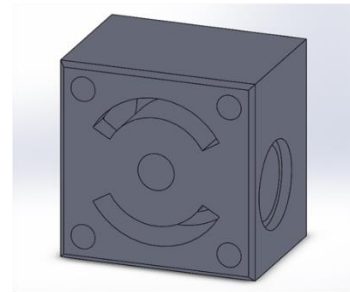
⁶ M. R. Mataczynski, *Air Force Institute of Technology*, 2014.



Design 1.1



Design 1.2



Design 1.3

Figure 8.2. Design evolution of PWS prototype housing.

Figure 8.3 shows a CAD assembly of the initial PWS prototype. The prototype was designed to include interchangeable shrouds to enable investigation and optimization of the port timing, interchangeable rotors to allow investigation of the rotor length effects, and a cooled exhaust side bearing housing. Additionally, initial experiments will be performed by driving the rotor with an electric motor to optimize the rotor speed.

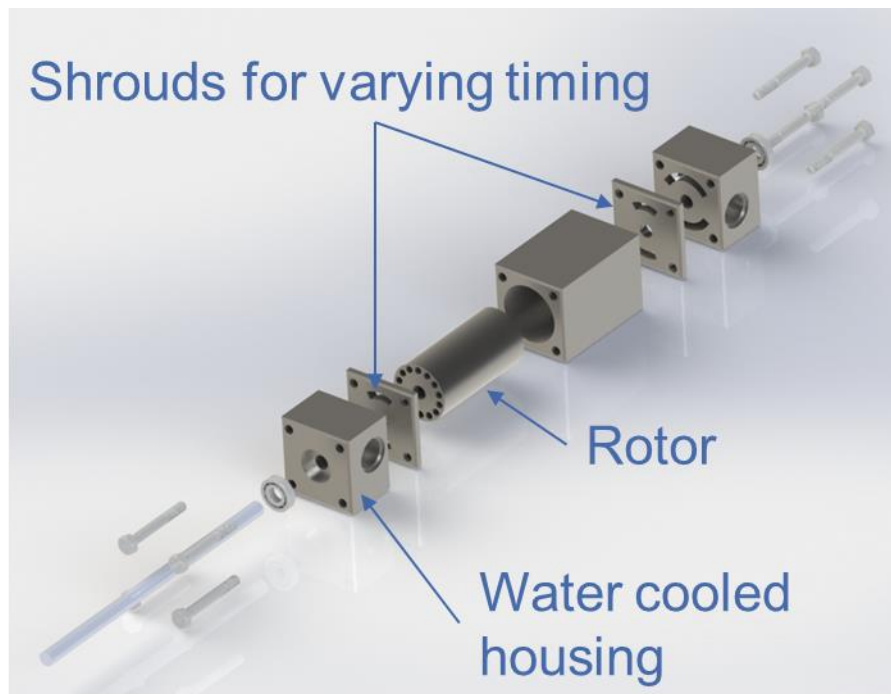


Figure 8.3. Exploded view of the PWS prototype Design 1.3.

In Q2, the design of the first prototype PWS system was completed and the PWS model was delivered to WERC. The thermal system design was performed using a 1-D gas dynamics code (GT-Power) discussed in the Q1 report. The GT-power model was used to assess the impact of manufacturing tolerances and it was found that boost levels are sensitive to the leakage gap between the rotor and ports. Figure 8.4

shows the load as a function of leakage gap. With a 0.010" leakage gap, either material was suitable for the application with Stainless Steel maintaining a 1.62 factor of safety.

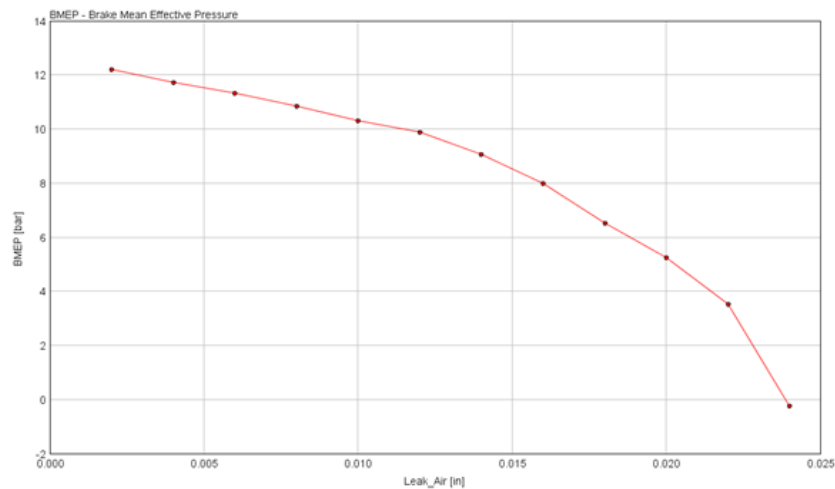


Figure 8.4. BMEP vs Leakage gap between rotor and ports. This plot shows the relationship between the gap between rotor and ports to the performance of the engine, a small gap is desired, but difficult to manufacture and maintain.

Figure 8.5 shows the completed prototype coupled to a variable speed electric motor. The system was designed with removable inserts to enable investigation of the impact of intake and exhaust port timing. This bench included a simple, air-cooled engine with a custom, flexible controller that allows investigation of a wide range of exhaust conditions to test and improve the PWS system.

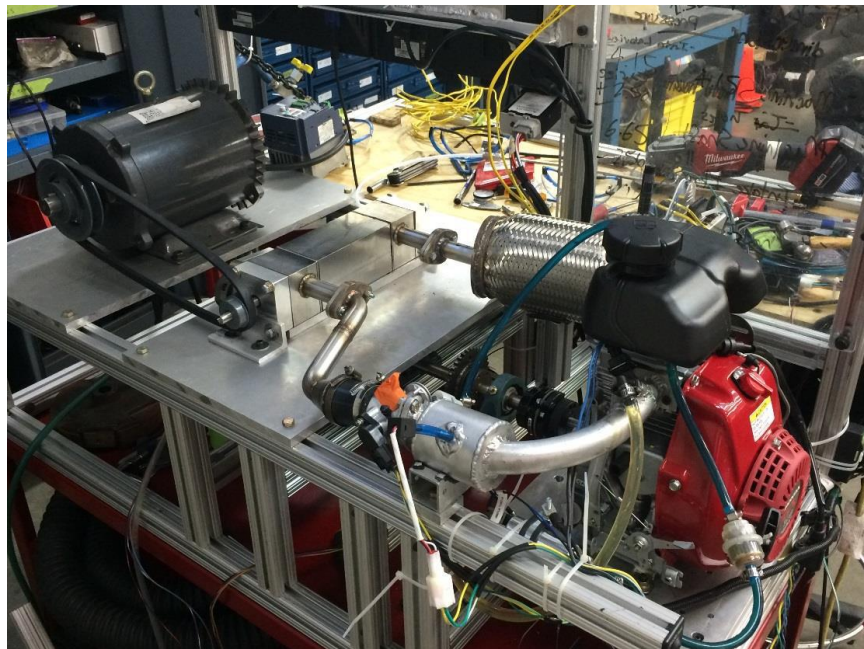


Figure 8.5. Test bench layout showing from left to right the AC motor, the PWS, and the engine. Also can be seen is the custom exhaust made to couple the PWS and engine.

The setup schematic shown in Figure 8.6 was instrumented with high-speed temperature and pressure readings into a National Instruments high-speed data acquisition system, which was used to correlate experimental data with predicted model data generated by WERC. Also instrumented in the setup are mass airflow (MAF) sensors for monitoring the mass balance in and out of each PWS intake and exhaust port. A critical part of the design was minimizing the amount of short-circuiting occurring from the exhaust into the PWS to the air out of the PWS into the engine. The MAF sensors in conjunction with the temperature sensors allow for monitoring of this condition which can arise in off-speed operation.

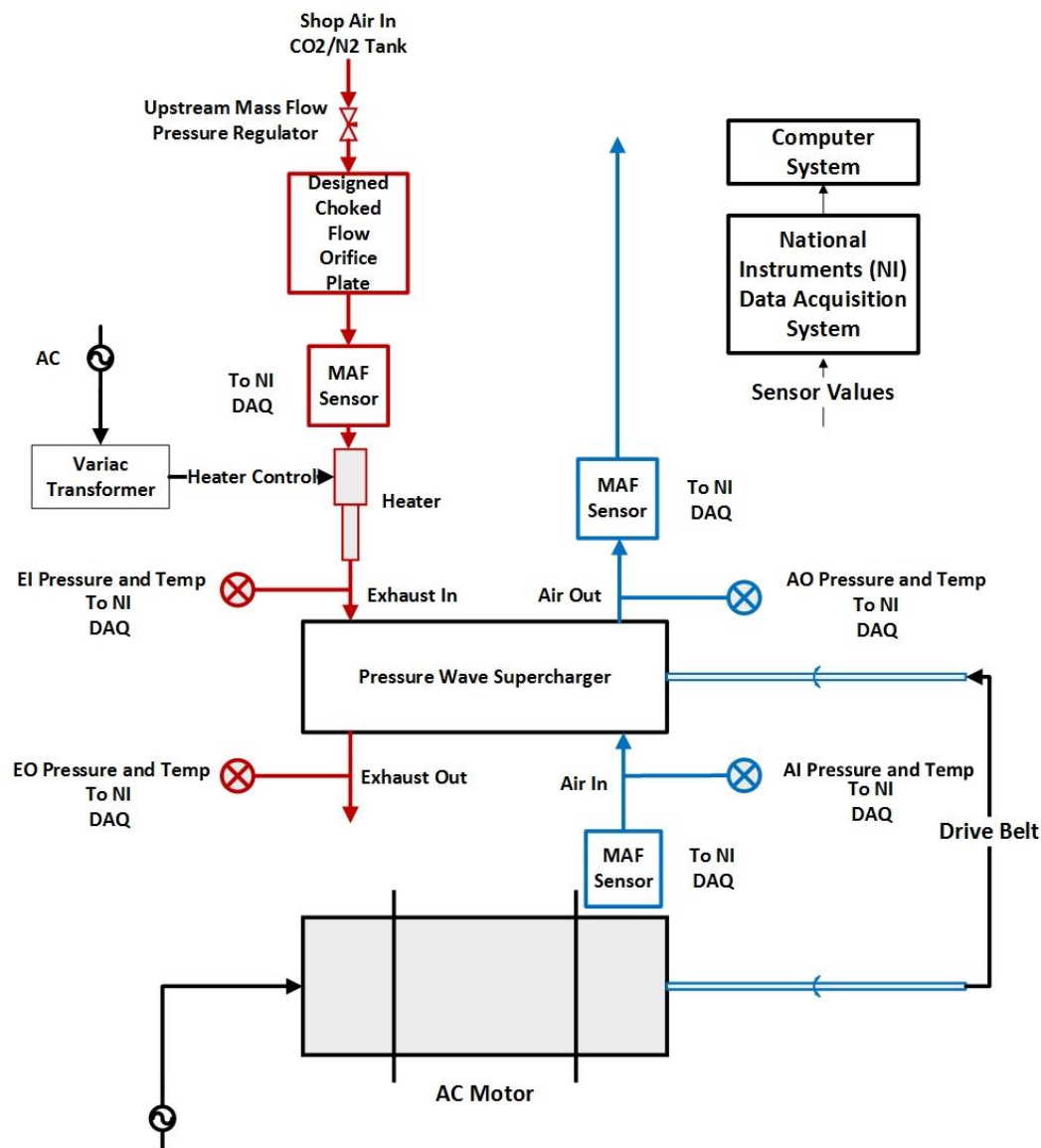


Figure 8.6. Schematic of flow bench testing setup. The schematic shows several of the pressure and temperature sensors around the PWS as well as mass airflow (MAF) sensors to characterize the PWS.

Preliminary testing showed unfavorable short-circuiting from the exhaust stream to the air intake stream. Figure 8.7 shows an example of a static test illustrating short-circuiting and backflow from the exhaust to the inlet. Notice that the air inlet (AI) flow is negative indicating reverse flow (leakage from

the exhaust to the intake). When the housing is properly sealed and the rotor is not spinning the only circuit for the gas flow should be from the exhaust inlet (EI) to the air outlet (AO) due to the overlap in port geometry. The gap between the rotor and variable geometry plates was large enough to allow the gas to move along the rotor face and exit out the AI port, which has lower resistance than the AO port to aid in scavenging.

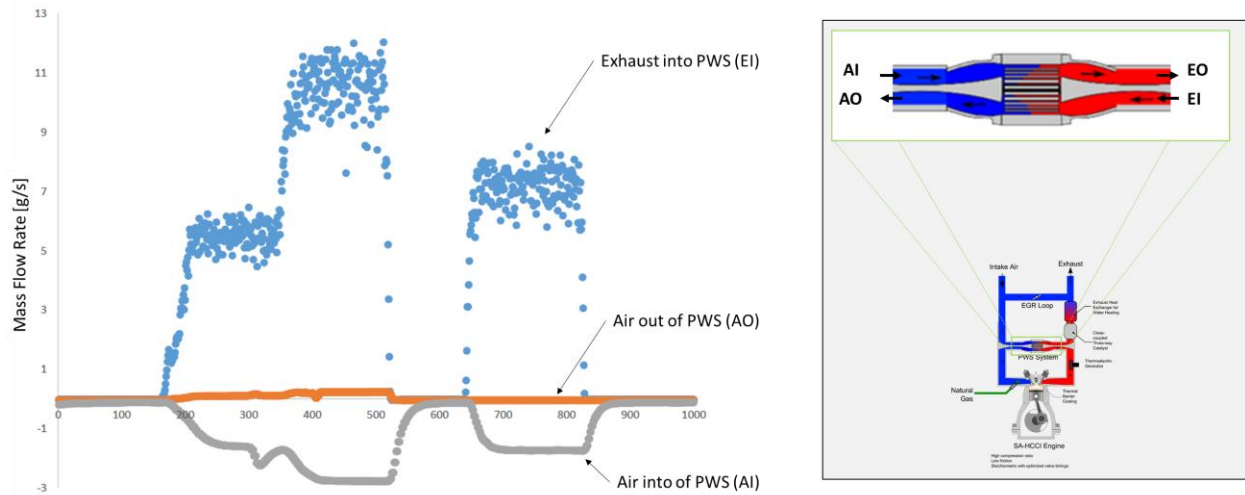


Figure 8.7. Mass flowrate under static test for the PWS system showing backflow into the intake system.

To minimize the leakage gap, new endplates were coated by Adiabatics Inc. with an abrasable coating. The coating will wear into place to minimize the leakage gap. Once the new end plates were coated and re-installed, testing continued. The PWS system was operated at rotor speeds from 6000 rev/min to 9000 rev/min and air outlet (AO) flowrates of 0 to 5 g/s. Exhaust inlet temperatures were controlled by an in-line heater and ranged from 300° to 330° C. The results of the testing are shown in Figure 8.8. The bench testing showed the potential to build charge pressure near 1.3 bar in the target intake flow rate range of 0.5 g/s to 1.5 g/s.

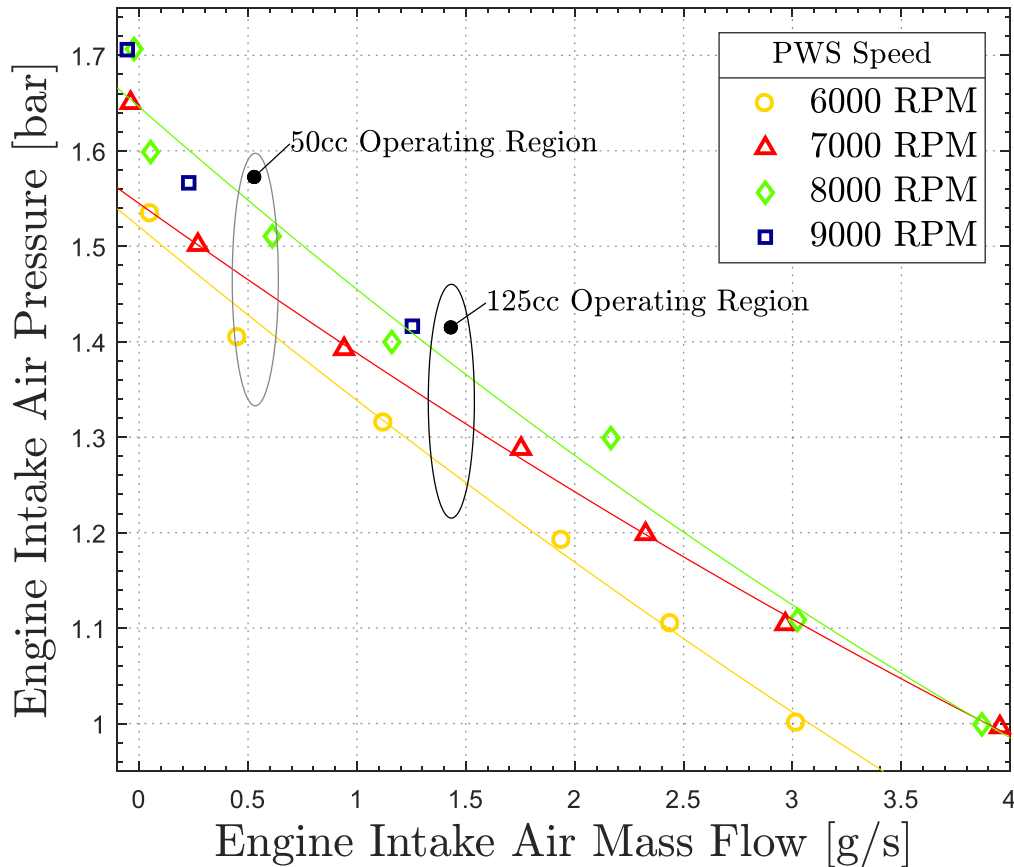


Figure 8.8. PWS outlet pressure as a function of intake mass flowrate.

These results show the promise of the PWS system to meet the desired boosting range; however, more testing was needed to identify any short-circuiting from the exhaust to the intake. Care must be taken to avoid excessive end-gap clearance to ensure maximum intake pressure. The pressure wave supercharger (PWS) was selected as the highest risk, highest reward system. Accordingly, most effort has focused on the development of this system. In Q6, testing of the PWS system took place and showed that the PWS system is capable of achieving the program targets of 1.3 bar intake pressure

The PWS system design is relatively complex and to avoid manufacturing complexities, additional novel technologies for intake pressurization, off-the-shelf components were evaluated to provide baseline data for comparison. The goal of this task is to perform analysis and experiments to select a boosting system capable of achieving the program targets (1.3 bar intake pressure).

Small Turbocharger

The IHI RHB31, also sold as the RHF3 or VZ21, turbocharger is marketed as the smallest commercially available turbocharger with a 24mm compressor and 27mm turbine (induction diameters). The procured turbocharger is shown in Figure 8.9.

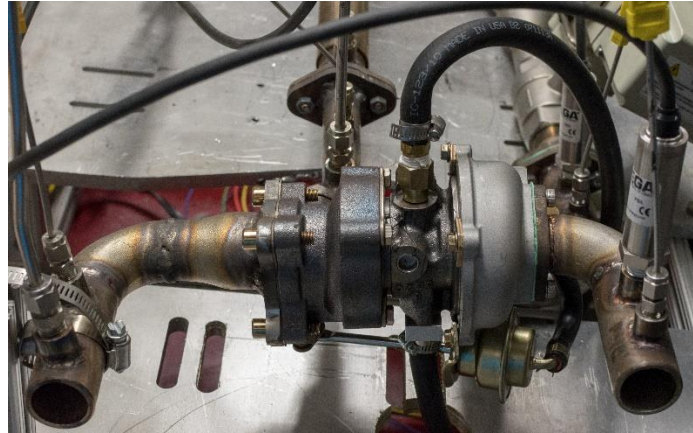


Figure 8.9. RHB31 Turbocharger as implemented in the flow bench. The left side is the turbine outlet (exhaust/EO) and the right side is the compressor intake (AI). The black lines visible in the image are the lubrication system that operates on a 12v pump circuit (not shown).

Experiments were performed over a range of mass flow rates and temperatures to evaluate the potential of the RHB31 to meet program targets. A scatter plot showing the pressure ratios for the tested conditions is given in Figure 8.10. The program target airflow is 2-3 g/s. The experiments showed no boost was achieved until the mass flowrate was above 5 g/s. Accordingly, it can be concluded that the RHB31 is not suitable to meet current program targets.

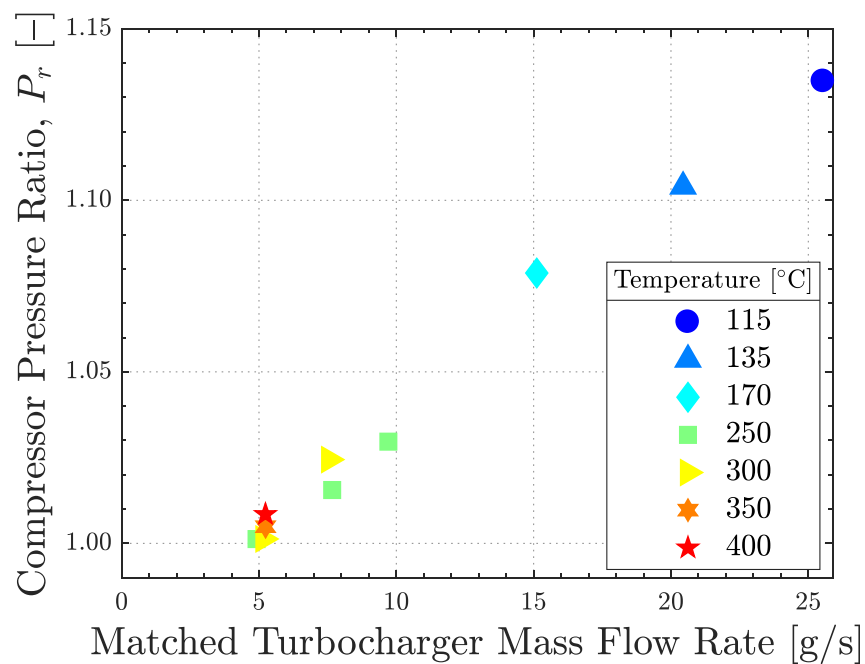


Figure 8.10. Compressor Pressure Ratio vs Mass Flow Rate of RHB31 Turbocharger as Implemented Data points Tested.

Passive Boosting

Based on engine testing in Task 2, the required intake pressure is expected to be less than 1.1 bar. Based on this analysis, a membrane-based intake charging mechanism, known commercially as a “Drum Charger”, was selected for investigation as an alternative to the PWS intake charging system where the intake and exhaust are separated by a flexible membrane. AEH Industrial Solutions has an impulse drum charger we tried working through a non-disclosure agreement (NDA) with AEH to procure a test version of the drum charger. The drum charger is a cylindrical chamber and with a membrane that separates the intake and exhaust flow streams. The exhaust pressure pulses (caused by the exhaust valve opening) are used to compress the intake charge. Details can be found at <http://www.alterego-hardware.com/images/Drum-Charger-Press-Kit-EICMA-EN.pdf>. Based on publicly available data (see link above), the drum charger has demonstrated an approximately 0.3 bar increase in intake pressure when appropriately tuned. Note that the demonstrated increase in pressure of 0.3 bar is similar to the value required for the current program.

Design of our prototype device was based on design features present in the extant prototype devices. To save on initial costs, the first prototype device was designed for manufacture with aluminum instead of an injection-molded polymer as would be done in a mass production (commercial) application. Figure 8.11 shows a cross-section of the first prototype device that was sent out for production. The device’s overall size is 225mm in diameter and approximately 150mm tall. The device’s membrane diameter is 150mm, which is approximately half the size of the commercial prototypes discussed previously.

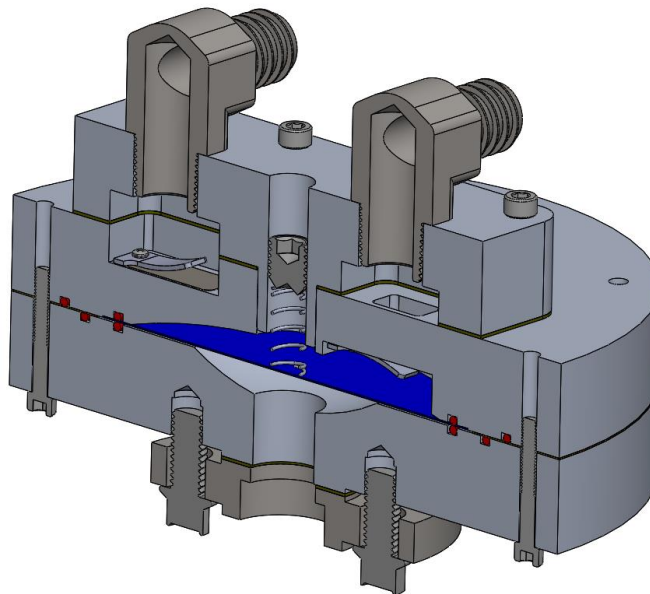


Figure 8.11. Section View of First Prototype Device out for Production. The fittings on the top of the intake chamber are standard 90° pipe elbows that permit interfacing with existing laboratory systems. The spring tension may be adjusted using the setscrew in the center of the intake chamber.

Static, off-engine membrane response tests were conducted for three different materials to evaluate their ability to displace (and compress) the air in the fresh air side of the membrane charger. The aluminum and polycarbonate materials selected for evaluation were chosen due to their decreased

material stiffness over the initial stainless steel membrane. The testing results presented in Figure 8.12 show the polycarbonate membrane being the most capable membrane in the range of pressures expected from the engine exhaust.

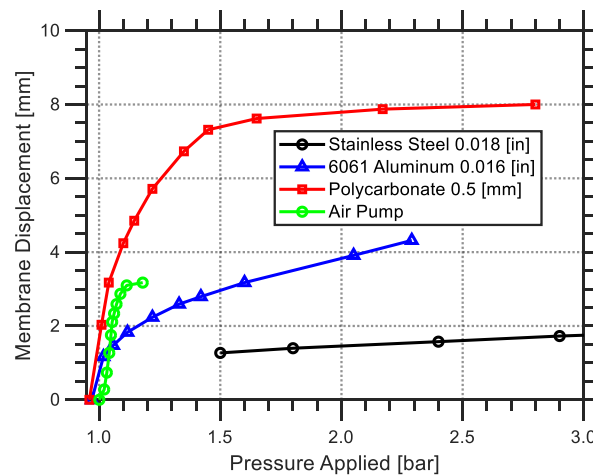


Figure 8.12. Membrane Displacement Behavior for Statically Applied Pressures.

The steel, aluminum, and polycarbonate membranes were installed on the membrane charger and have a maximum displacement of 8mm. The air pump is a modified device with a rubber membrane with a maximum displacement of 3.2mm.

A closed volume was constructed to simulate the closed volume nature of the intended intake system between the intake valve and the membrane charger intake outlet (AO). This closed volume consisted of an NPT pipe section, and a control valve to allow relief of pressure during the on-engine testing. Figure 8.13 shows this device setup on the engine with the high-speed instrumentation in place.

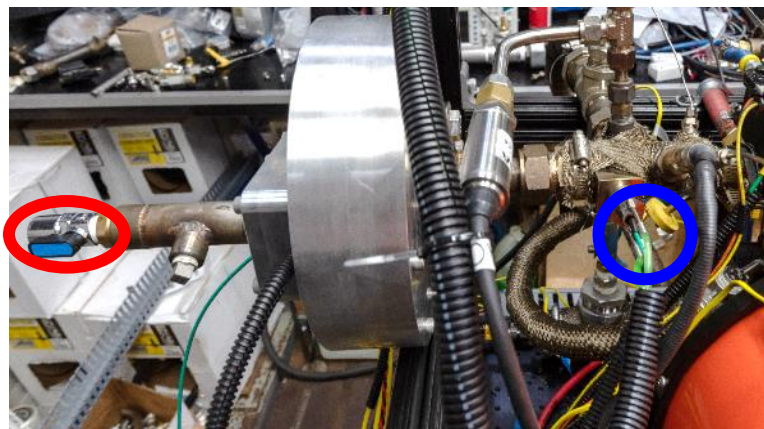


Figure 8.13. Drum Charger Installed On-Engine with Closed Volume and High Speed Pressure Transducers. The valve circled in red is used to control the buildup of pressure in the closed vole. The high-speed (liquid-cooled) pressure transducer is shown in the blue circle.

On-engine testing was performed on the generation 3 engine at operating speeds of 1800 and 2400 rev/min for the aluminum membrane and 1800 and 2600 rev/min for the polycarbonate membrane. The control valve was left open to establish a baseline pressure and closed to determine the boost pressure that the device would build to. Figure 8.14 shows the membrane charger can produce between 0.05 and 0.07 [bar] of boost for either membrane.

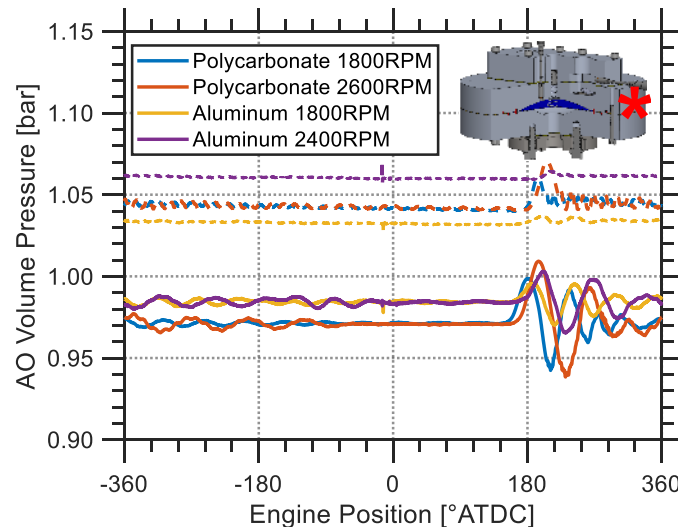


Figure 8.14. Boosting Capability of Membrane Charger in the Closed-Volume Configuration. The dotted lines indicate the valve was closed to build boost, whereas the solid lines show the open valve condition. Pressure measurement location is starred.

Overall Recommendation

Overall, the boosting system analysis suggests that a system can be designed to provide up to 1.3 bar absolute intake pressure (the pressure wave supercharger), but the cost is likely to be prohibitive. The lower cost, membrane charger is likely to provide less than 1.1 bar absolute intake pressure. While this level of boost is likely sufficient to enable 10 to 15% EGR, the efficiency improvement increase by adding boost is moderate (~1%). Accordingly, the Task 8 effort should focus on providing intake and exhaust flow data to the modeling effort and the effort should focus on methods to decrease the combustion duration as a more cost-effective method to improve brake thermal efficiency.

Task 9 – Design and Construction of Phase 1 Prototypes

Summary

The objective of this task is to create drawings and sourcing plans for Phase I prototypes. Source or manufacture components necessary to build two prototypes and additional parts to support development testing. Two identical prototype engines were built and ready to be delivered for testing (one for Briggs & Stratton and one for Univ. of Wisconsin – ERC).

Results

Concept # 1 Prototype

As reported under Task 6, Engine Concept #1 was formulated to allow early combustion development and modeling validation using many “off-the-shelf” production engine components. The engine was created by mating a Honda PCX 125 crankcase with a Honda Metro 50 head. This combination was selected to closely represent the optimized configuration presented at the Task 7 design review as Concept #3. Table 9.1 compares the important geometric parameters of the optimized Concept # 3 configuration and the Concept #1.

Table 9.1. Engine Design Parameters.

	Concept #1	Concept #3
Displacement	71 cc	70 cc
Bore	39.5	40 mm
Stroke	57.9	56 mm
Bore/Stroke Ratio	0.68	0.71
CR	17:1	18:1
Included Valve Angle	43°	29°
Intake Seat Dia.	17.0 mm	14.5 mm
Exhaust Seat Dia.	12.5 mm	12.5 mm

Partial and fully assembled views of Concept Engine 1 are included in Figures 9.1 and 9.2, showing a custom cylinder assembly to adapt the Metro bore size to the PCX crankcase. Briggs & Stratton fabricated the custom cylinder and sleeve to adapt the Metro 50 cylinder head and piston bore to the PCX125 long stroke crankcase. Analysis showed that the standard Metro piston would be severely overstressed with the predicted peak firing pressures of the Concept # 1 configuration. The redesigned piston, shown in Figure 9.3, was procured, allowing final engine assembly and installation on the UW-Madison ERC project dynamometer. Results of initial testing are provided under Tasks 11.

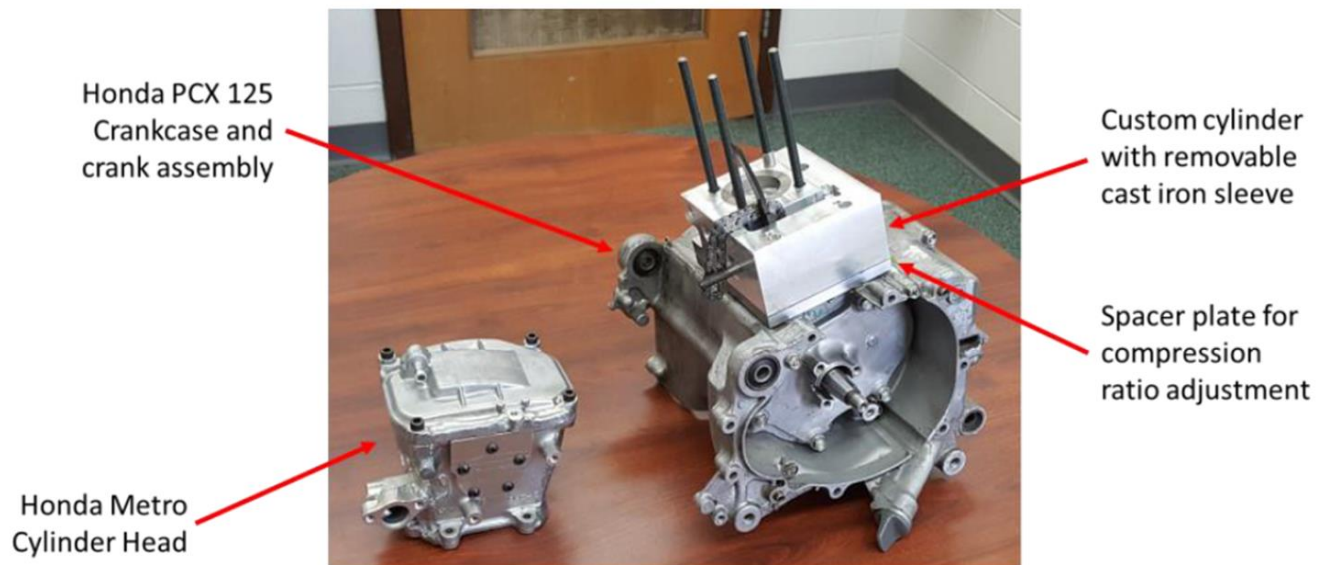


Figure 9.1. Engine Concept #1 Assembly.

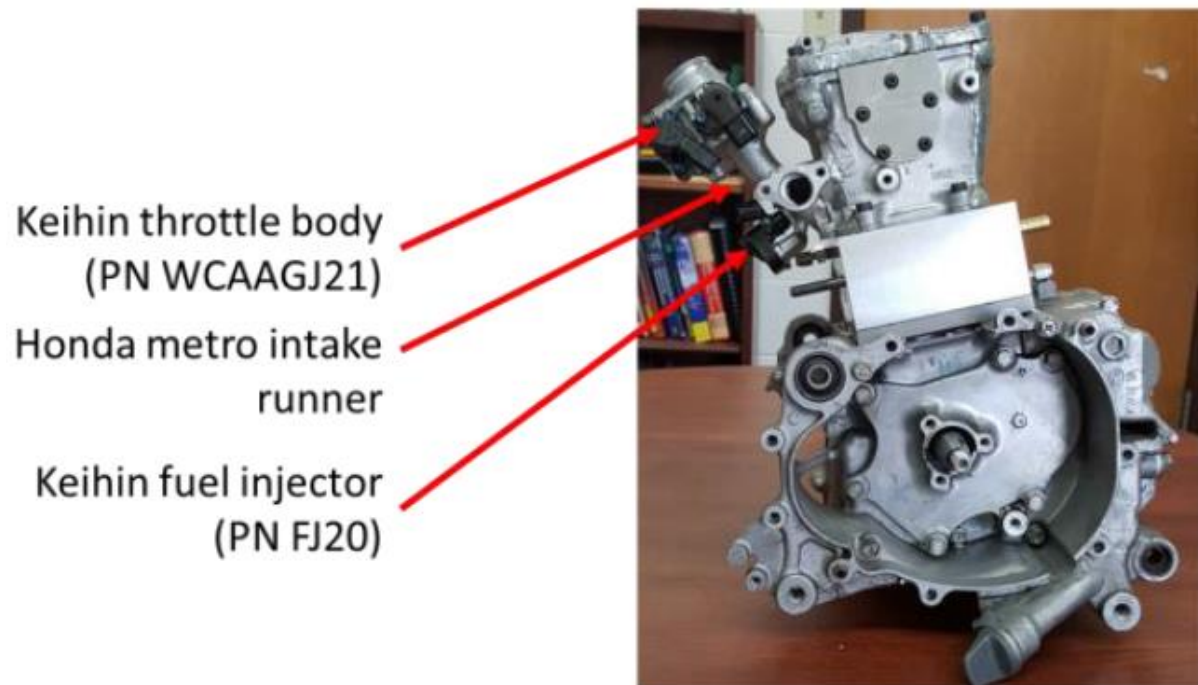


Figure 9.2. Engine Concept #1 Assembly.

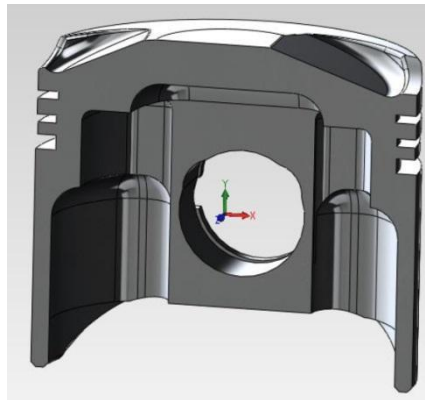


Figure 9.3. Engine Concept #1 Piston.

Concept # 3 Prototype

Based on the results of detailed CFD modeling included in Task 7, design efforts accelerated in the first quarter of 2017 of the Concept #3 configuration. Design layouts of the cylinder head, cylinder, and crankcase were the initial focus as these components involve long procurement times due to requiring prototype castings.

Cylinder Head

The cylinder head design required careful consideration of intake and exhaust porting, combustion chamber shape, cooling water jacket configuration, and the inclusion of the complete OHC valve train. Intake and exhaust porting as well as combustion chamber geometry used in the Task 7 modeling efforts were maintained in the detailed cylinder head design. The 29° included valve angle configuration, the desire to allow easy incorporation of hydraulic valve lash adjuster and to minimize valve train friction led to the selection of the end pivoting Type II valve train..

With the valves in the same axial plane, a unique dual roller, double arm rocker was schemed for the intake valve, thereby eliminating the need for offset rocker arms and the resulting twisting moment. The type II system allowed shorter valve stem lengths than a center pivoting system (known as a type I system). The camshaft drive system was configured to use an inverted tooth chain for good load capability, high efficiency, low noise, and good durability. The valve train arrangement is shown in Figures 9.4 and 9.5. Other cylinder head features include coolant inlet and outlet ports to allow separated cooling paths from the cylinder, a pressure transducer port, and oil galleries for the hydraulic lash adjusters. An overview of the cylinder head assembly is shown in Figure 9.6.

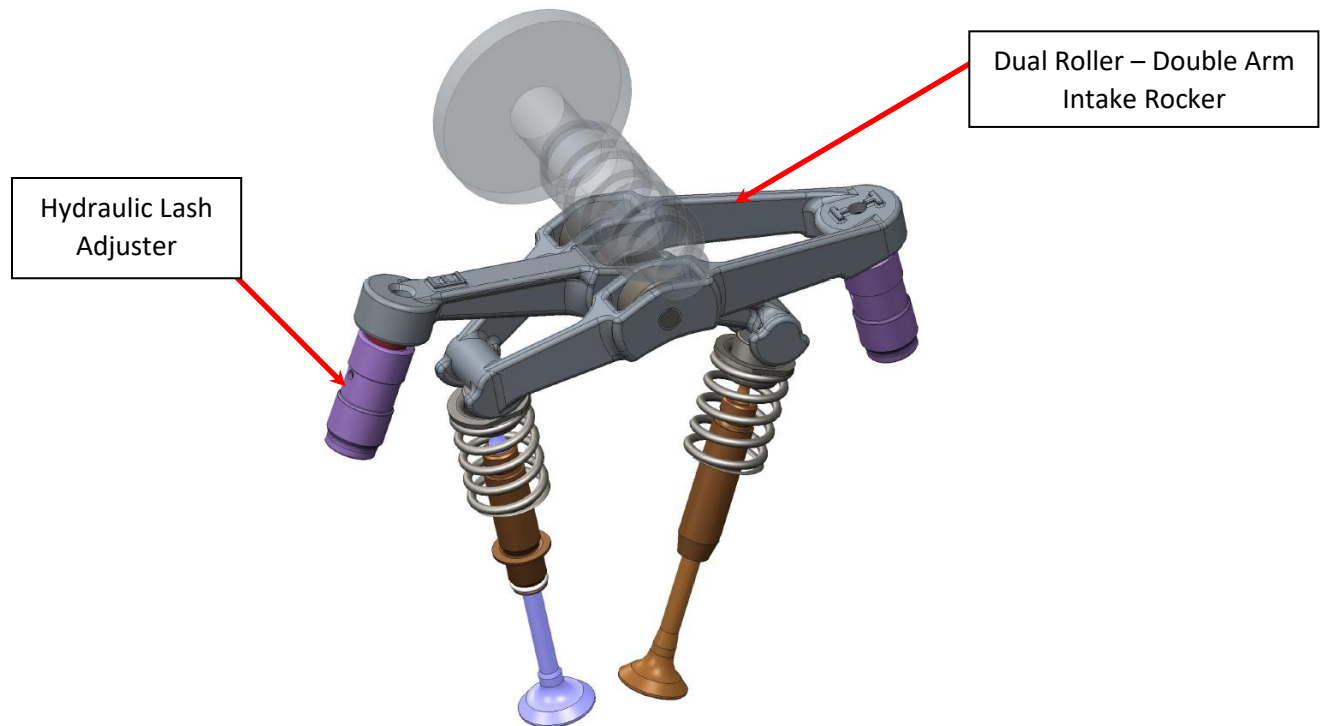


Figure 9.4. Concept #3 Valve Train Arrangement.

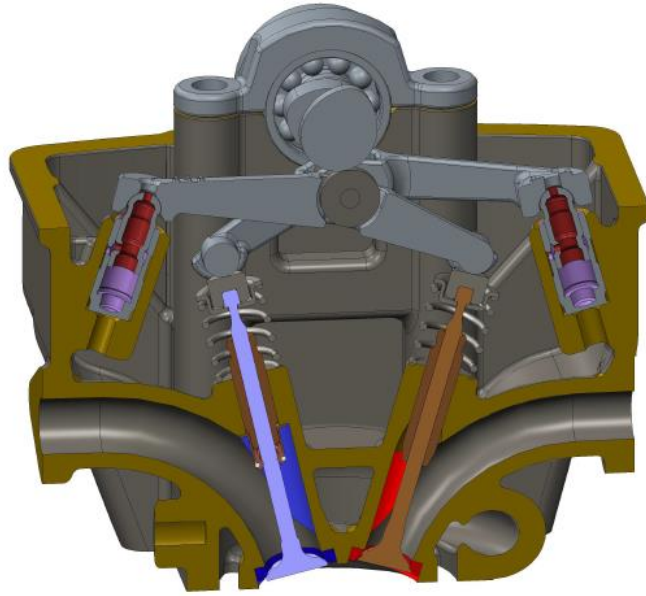


Figure 9.5. Concept #3 Cylinder Head.

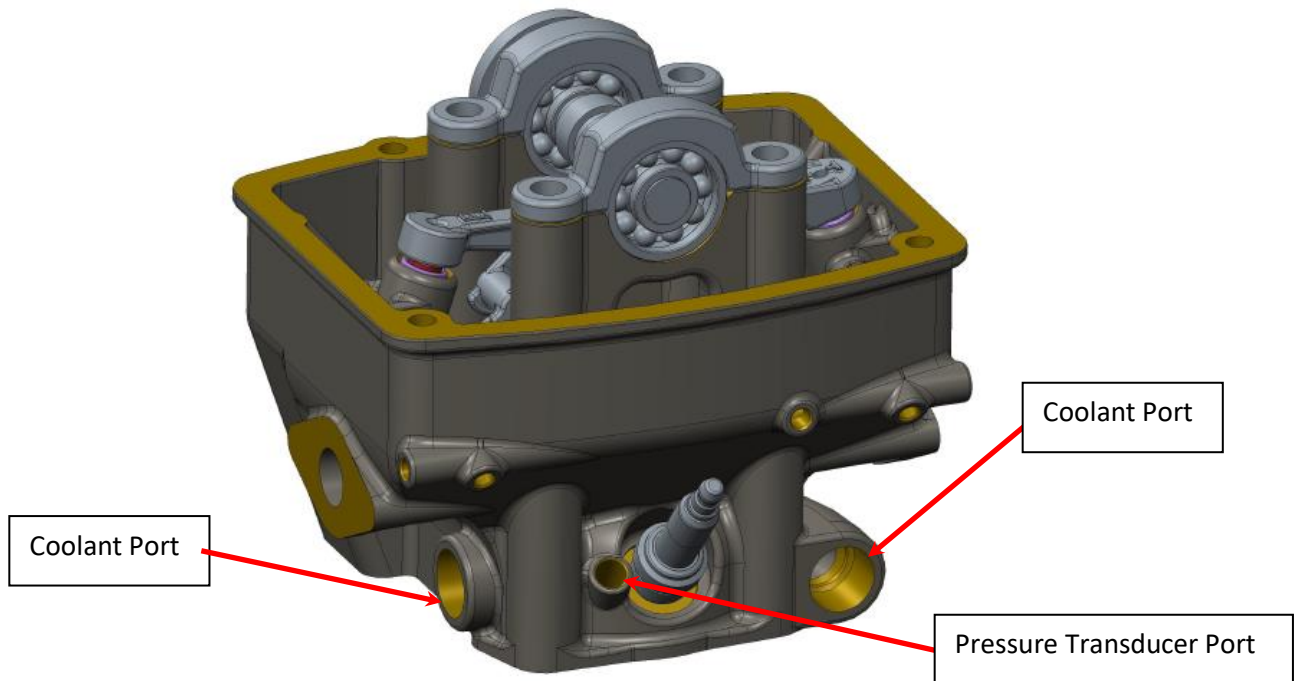


Figure 9.6. Concept #3 Cylinder Head.

Cylinder

A wet sleeve cylinder design was chosen to allow sleeves of different material and surface treatment to be installed during engine development. This allows investigation of materials with improved wear resistance and/or reduced thermal conductivity, as well as surface treatments for reduced friction

(diamond like coatings (DLC), thermal spray coatings such as SUMEBore, Nikasil, or electrochemical plating, etc) The cylinder has provision for 2 cooling port inlets to allow experimentation with cooling jacket length and includes a separated flow path from the cylinder head. The cylinder assembly is shown in sectional view in Figure 9.7 with an overall view in Figure 9.8.

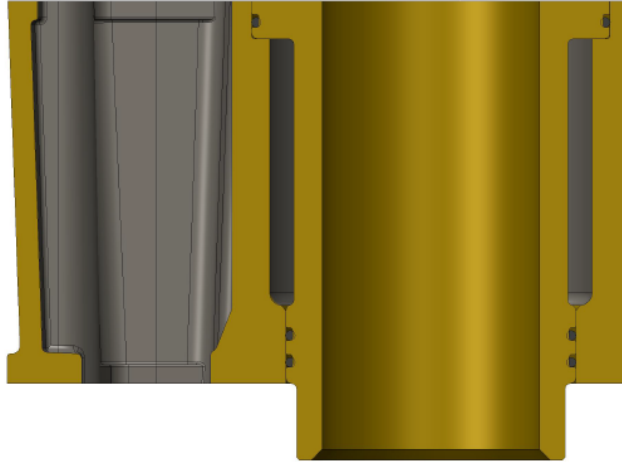


Figure 9.7. Cylinder and Sleeve Cross-Section.

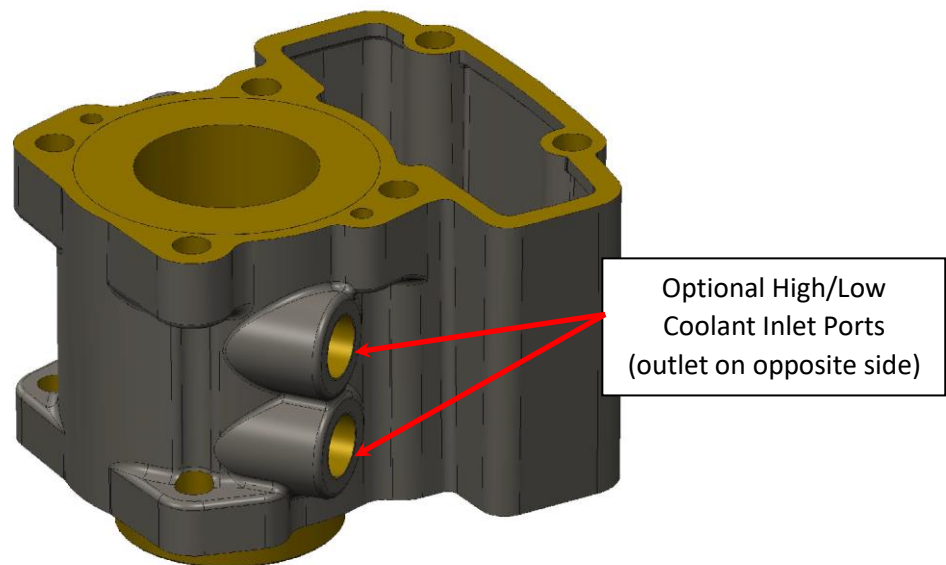


Figure 9.8. Cylinder and Sleeve Assembly.

Crankcase

A ladder construction crankcase design is incorporated in the phase 1 prototype engine configuration. The main bearing caps provide a ridged, equalized support structure for the crankshaft. To line bore the bearing caps, a relatively large crankshaft oil seal carrier is required at the open end. Four mount system supports are also provided to allow for dyno mounting or a focused, elastomeric mounting system in the generator application. See Figures 9.9 and 9.10 for crankcase design details.

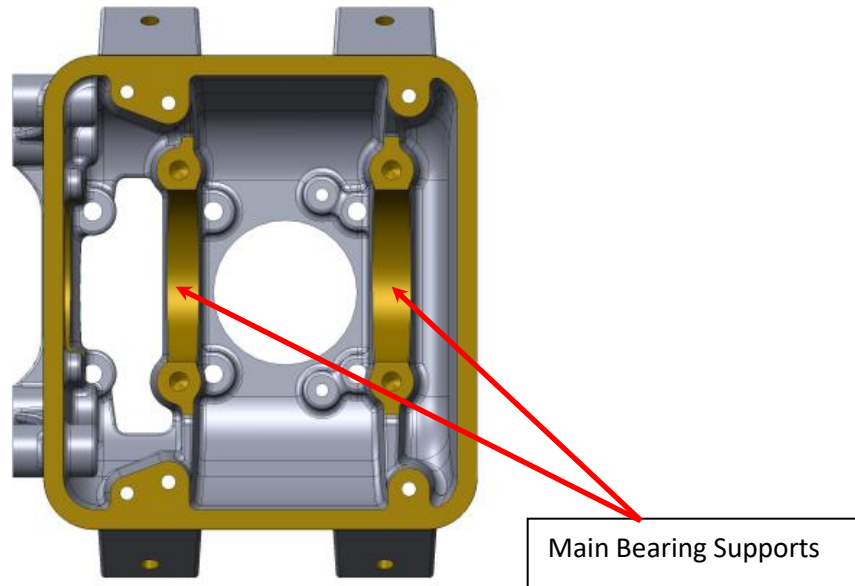


Figure 9.9. Crankcase Internal Structure.

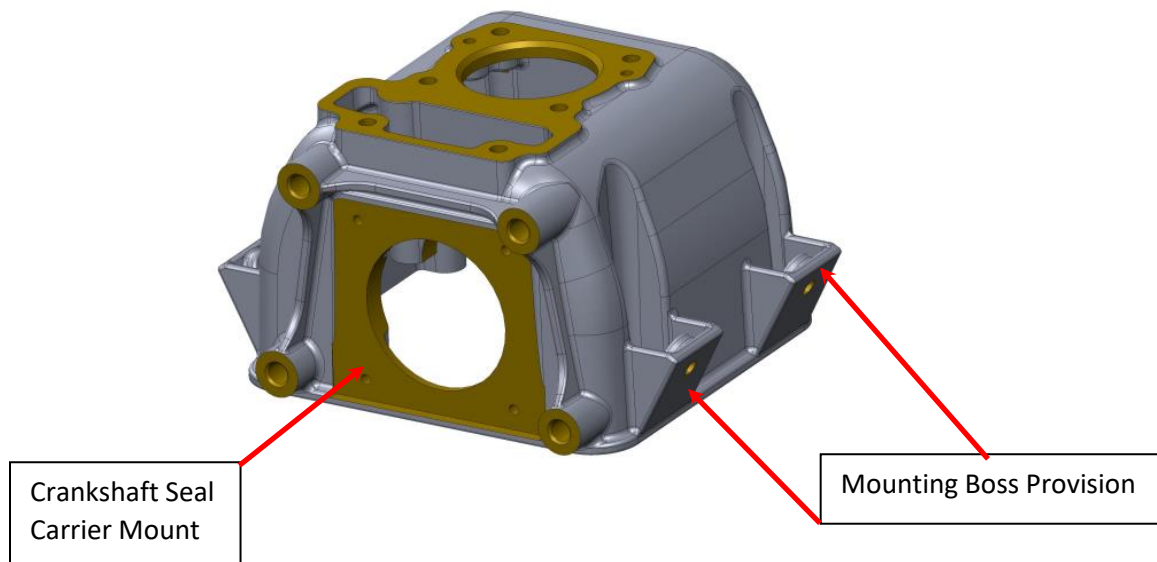


Figure 9.10. Crankcase External.

With design layouts of the cylinder head, cylinder, crankcase, and power transmission components complete, the primary focus turned to detail design and stress analysis to allow an initial prototype build design freeze. Numerous component analyses were accomplished, with notable improvements required for the cylinder head casting and piston/pin/connecting rod detail/

Cylinder head FEA (Finite Element Analysis) showed a high stress condition in the coolant jacket area at the cam drive side. A long coolant passage slot in the initial design was changed to two shorter conduits with a reinforcing rib and larger cast fillets to reduce stresses and deflection.

The initial analysis of the piston, piston pin, connecting rod, and crankshaft showed high stress in the piston pin areas of the connecting rod and piston caused by piston pin ovalization. A reduction of the ID hole size in the piston pin reduced stresses to an acceptable level.

CFD (computational fluid dynamics) analysis of the flow stream within the coolant jacket of the cylinder assembly initially showed a disorganized and inconsistent flow field. A well-organized flow field resulted with revisions to the inlet and outlet configuration.

Additional FEA, CFD, and manual calculations verified low stress, deflections and smooth flow conditions for the design, without changes to the initial details. Long lead time engine castings were ordered and coordination of machining orders for castings and billet parts began.

Valve Train Analysis

Valve train dynamic analysis simulations were completed to ensure valve motion stability. As mentioned above, the Concept 3 engine valve train features an overhead cam design with valves at a 29° included angle. A “Type II” end pivoting rocker system with hydraulic valve lash adjusters was selected to minimize valve train friction and to provide maintenance free operation. A unique dual roller, double arm rocker was schemed for the intake valve, thereby eliminating the need for offset rocker arms and the resulting twisting moment. .

The first step in the modeling process requires the determination of valve train component weight, inertia, and stiffness. CAD modeling and FEA analysis of component deflection under load were used to obtain these inputs. The optimized valve lift profiles from project Task 3 was the starting point for the dynamic analysis. Modifications to the initial lift ramps and the positive acceleration regions of the valve profiles were required to provide smooth dynamic response while at the same time matching the optimized lift curves at valve lifts greater than 1mm. Valve train vibration (velocity fluctuations) were seen to dampen quickly during the valve opening, a result due to the combination of high component stiffness and optimized valve event acceleration. Very controlled valve seating velocities (~7.5 in/sec) should result in long valve face and valve seat life. See Figure 9.11.

Because of the smooth dynamic valve train response, a reduced valve spring force margin is possible, further reducing valve train component loading, stress, and wear. Lobe/follower contact stress (~50 ksi) is well below design goal (≤ 100 ksi). Dynamic simulation of camshaft drive torque was used to design the camshaft drive system (sprockets, chain, and tensioner system).

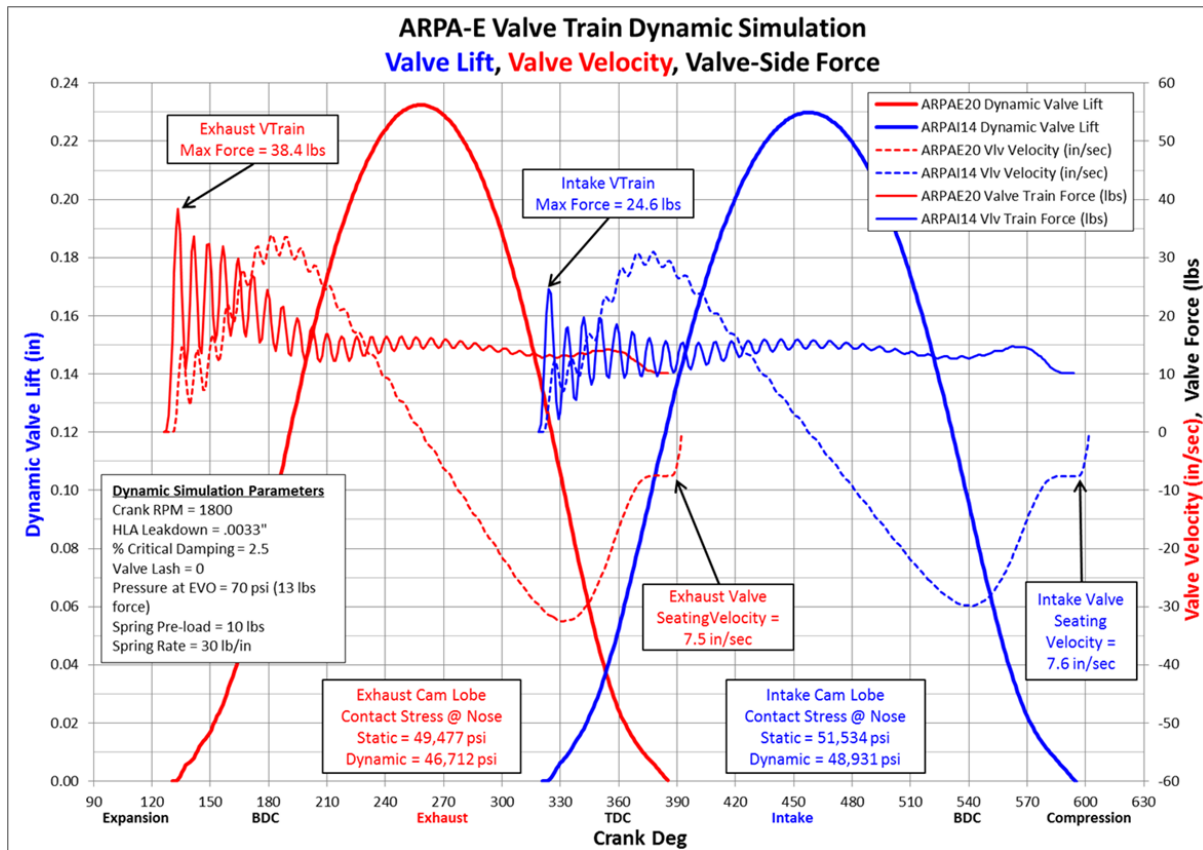


Figure 9.11. Dynamic Simulation – Lift, Velocity, and Forces.

An engine design cross section is provided in Figure 9.12. A complete engine Bill of Material (BOM) was developed with detail drawings completed for all components (Briggs & Stratton development project file 240d).

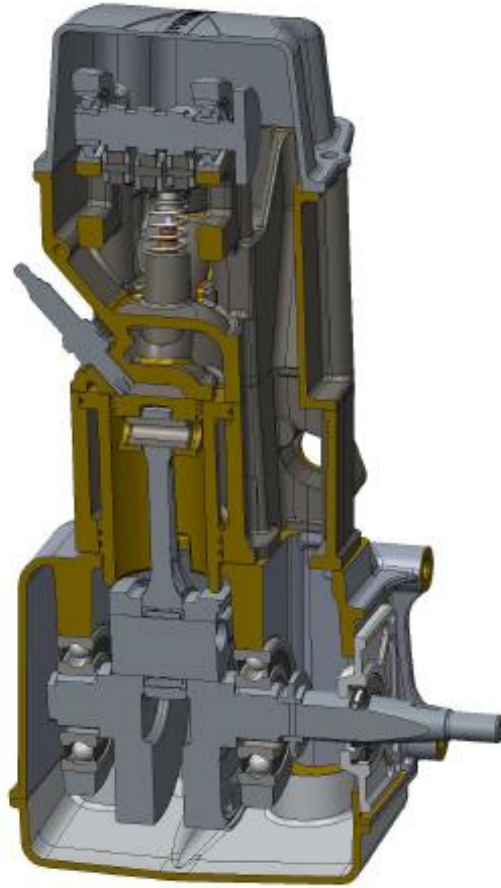


Figure 9.12. Engine Cross Section.

During project Q9 (Q1, 2018), all engine components necessary for the assembly of the first Concept #3 engines were completed. Three components provided challenges in procurement or internal Briggs & Stratton Model Shop machining – the piston, the cylinder head assembly, and the camshaft.

Piston

A significant learning curve was encountered with piston skirt machining. As our typical piston material is aluminum, changing to ductile iron required the development of alternate tooling and machining rates. Additionally, obtaining the desired vertical barrel profile was a challenge that required programming offsets to correct.

Cylinder Head Assembly

Final valve seat machining proved difficult due to the extremely hard valve seat material used for extended life requirements. Our initial custom sized cutters created a seat eccentric to the valve guide compromising valve sealing. A special multi-fluted carbide tool was carefully aligned to the finish guide, resulting in near perfect alignment and finish.

Camshaft

After lobe pre-machining and heat treat, prototype camshafts were sent out for final profile grind. Due to a miscommunication, the lobes were ground for a flat faced follower versus our roller follower design, resulting in an incorrect profile. The “cam blank” process was repeated with cams still in process at the end of the quarter. Two profile milled prototype cams, that were subsequently harden and polished, allowed the assembly of the first engines.

Specialty components procured from Briggs & Stratton suppliers are included in Table 9.1. All other components were custom castings (machined by the Briggs & Stratton Rapid Prototype Center, or RPC) or billet machined parts (machined by the Briggs & Stratton RPC or outside model shops).

Table 9.1. Specialty engine components received from suppliers.

Component	Supplier
Piston Rings	Federal Mogul
Valve Seat Inserts	Winsert
Valves	TRW (Federal Mogul)
Hydraulic Valve Lifters (HLA's)	Schaeffler (INA)
Cam Drive Chain	Tsubaki
Control/Fuel System	Delphi (injector, TMAP, position sensors)
Spark Plug	NGK
Ball Bearings	Nachi
Connecting Rod Roller Bearing	INA

A trial engine assembly allowed a check of the overhead camshaft drive system, including the crankshaft and camshaft sprockets, the roller chain, the chain guides, and the chain tensioner assembly. Sprocket alignment was good and the 111 link chain length proved correct. The chain guides aligned well with the chain assembly and the tensioner had acceptable travel.

Additionally, the crankshaft-connecting rod assembly was completed with good results. This assembly includes the crankpin pressed into the crank half shafts with the connecting rod and needle roller bearing. A specially designed assembly fixture was used to ensure proper main bearing alignment. With main bearing roller bearings installed on the assembled crankshaft, the fit-up to the crankcase showed good alignment and smooth rotation. See Figures 9.13 through 9.15.



Figure 9.13. Engine Concept #3 Assembly.

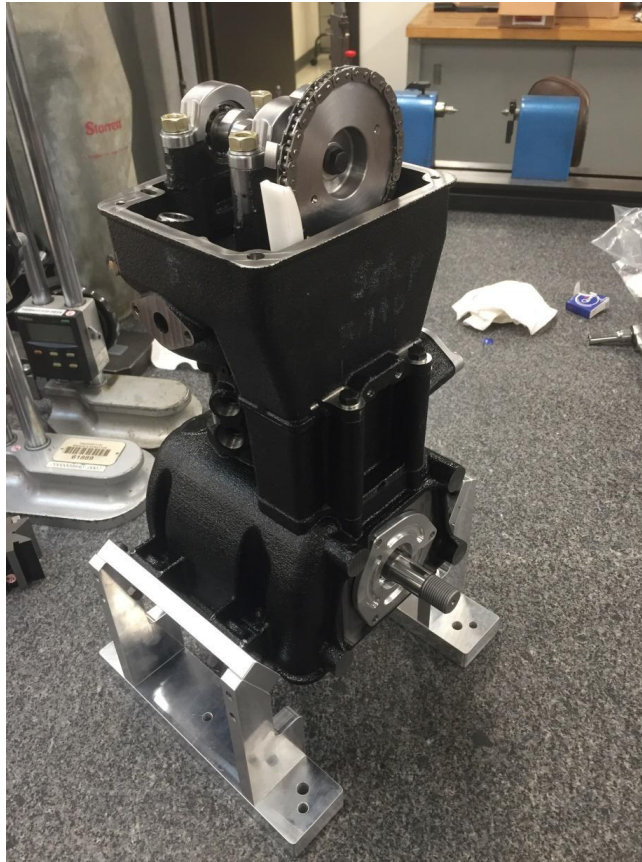


Figure 9.14. Engine Concept #3 Components.

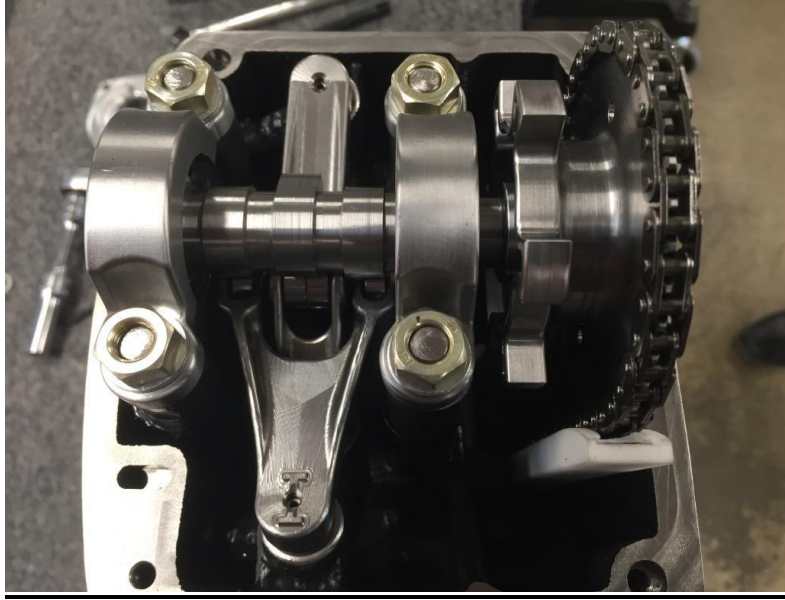


Figure 9.15. Concept Engine 3 Overhead Valve Train_Auxiliary Engine Systems.

Engine Concept #3 uses an external lubricating oil pump system with a scavenge pump return to an external reservoir. This allows a high degree of development flexibility where lubrication points (hydraulic lash adjusters, overhead cam ball bearing, connecting rod bearings, main bearings, and engine cylinder) can have flow rates optimized for performance and temperature. See Figure 9.16. Similarly, an external coolant pump and heat exchanger system was used to allow flow rates to be optimized separately for the engine cylinder and cylinder head. See Figure 9.17. Finally, a throttle body assembly with an integrated fuel injector was designed, including an integrated intake pressure and temperature sensor (TMAP) for calibration requirements (See Figure 9.13).



Figure 9.16. Briggs & Stratton GENSETS Lubrication System.



Figure 9.17. Briggs & Stratton GENSETS Cooling System Engine Operation.

After static engine lubrication system evaluation and initial engine motoring to refine the lubrication system flow and spray patterns, the engine was mounted to an eddy current dynamometer. First engine operation occurred on March 28th. Engine break-in went smoothly, but oil temperatures were lower than expected and coolant temperatures were difficult to control. Performance scans were obtained at 1800, 2000, and 2200 rpm at a fixed spark timing (20°) and at stoichiometric air-fuel ratio. After approximately 10 hours of operation, the engine was disassembled for inspection.

Engine component condition was very encouraging at the 10 hour inspection. Piston and cylinder bore marking showed narrow skirt support and rocking that is typical of an initial build condition. Reduced skirt ovality and reduced skirt axial cut-back opposite the piston crown is planned for the next piston iteration. The diamond like coated (DLC) cylinder sleeve showed evidence of coating delamination due to incorrect processing and is subject to further study. It is suspected that the DLC material caused the observed piston and bore scuffing damage.

Conclusions

Engine Concept #1 components were fabricated and assembled using a custom cylinder and piston to mate the Honda PCX 125 crankcase to the Honda Metro cylinder head. This combination closely represented the optimized configuration presented at the Task 7 design review as Concept #3 and allowed early combustion development and modeling validation. Testing of the Concept #1 prototype is reported under Task 10.

Engine Concept #3 design and analysis was completed based on Task 7 modeling optimization. Custom castings were ordered with machining of castings and other billet parts being accomplished by the Briggs & Stratton Rapid Prototype Center or outside model shops. Specialty engine components (valves, piston rings, seals, bearings, etc.) were procured from traditional Briggs & Stratton suppliers. Two engines were assembled, one for mechanical development testing at Briggs & Stratton, the other for performance development at UW-Madison ERC. At least two sets of spare components were also fabricated.

Task 9 was successfully completed.

Task 10 Experimental Evaluation of Phase I Prototype Engine

Summary

A detailed experimental campaign has been performed to identify the peak brake thermal efficiency of the concept 1 prototype (engine developed in this program). Experimental optimization of the Concept 0 engine has been completed, achieving a peak BTE without EGR of 29.4%. When EGR was added, the peak BTE was increased to 30.5%. The concept 1 prototype engine was installed and preliminary shakedown testing was performed to provide data to WERC. The increased compression ratio and improved bore-to-stroke ratio enabled a BTE of 33% without EGR. Experiments investigating the effects of EGR showed a peak BTE of 34.1% at 10% EGR. These data were compared to the model predictions and CFD modeling and the results were used to further refine the computational models for the final selection of the piston geometry, cylinder head geometry, valve timing, and compression ratio. This intermediate validation step using the final base engine architecture will enable full optimization of the final platform to demonstrate 40% electrical efficiency at 1-KWe. Both milestones associated with Task 10 have been met per the project schedule. Milestone 10.2 was the year 2 go/no-go decision point. We met this milestone by demonstrating over 29% brake thermal efficiency. Milestone 10.1 was focused on achieving a brake thermal efficiency of 33%. We met this Milestone by achieving a peak brake thermal efficiency of 34.1%. The current effort has fulfilled the targets set for deliverables M10.1 and M10.2.

Details

Three hardware sets were evaluated to achieve peak brake thermal efficiency (BTE) and supply model validation data to WERC. Table 10.1 shows the three engines evaluated. Concept 0 is an off-the-shelf engine used to evaluate the test stand. The concept 0 engine has a displacement of 125 cc and an 11:1 compression ratio. The engine was designed to operate on gasoline, but was coupled with a prototype ECU and modified to operate using natural gas. After system optimization, the peak BTE observed in the concept 0 engine was 29.4% without EGR. Adding cooled EGR increased the peak BTE to 30.5%.

The CFD analysis suggested that improved efficiency would be obtained using a 70 cc engine with a bore to stroke ratio of 0.71 and a compression ratio of 18:1. This engine, named Concept 3, was manufactured by Briggs and Stratton (see Task 9). To enable testing prior to completion of the Concept 3 engine, an intermediate engine was built at UW-Madison's Engine Research Center. This engine, named Concept 1, has specifications nearly identical to the Concept 3 engine, but was able to be fabricated out of mostly existing components. A custom cylinder, piston, and mating system was fabricated to provide a displacement of 71 cc, a stroke-to-bore ratio of 0.68, and a compression ratio of 17:1. Preliminary experiments have achieved a BTE of 29.6% without EGR; however, the friction mean effective pressure was found to be excessive due to the test stand setup. This has been modified and testing is being repeated to identify the peak BTE with and without EGR.

Figure 10.1 shows the Concept 1 engine disassembled (top) and assembled (bottom). To avoid the complexity and lead time associated with custom gaskets, the Concept 1 engine uses the stock base gasket from the Honda PCX 125 and stock head gasket from the Honda Metro. The system was designed with a removable cast iron sleeve to enable replacement if damage occurs during initial testing. UW – Madison designed the components and Briggs and Stratton fabricated most of the hardware in their

prototype shop. The system was designed with a custom “spacer plate” located between the custom cylinder and crankcase. This allows the compression ratio to be adjusted without making additional hardware changes. Several thicknesses have been fabricated to allow investigation of a range of compression ratios from approximately 12:1 up to 17:1.

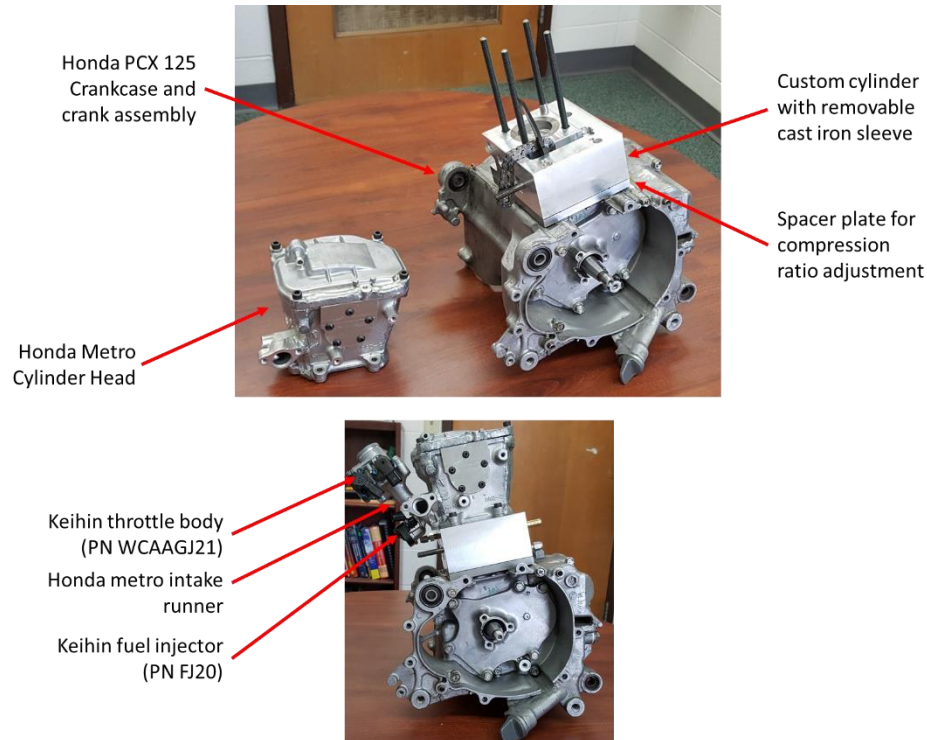


Figure 10.1. Concept 1 engine disassembled (top) and assembled (bottom).

Table 10.1 shows the dimensions of the Concept 1 engine and current progress between engine concepts. Figure 10.2 shows the summary of peak BTE measured during engine testing. As can be seen, we have exceeded the year 2 target by achieving a peak BTE of 34.1% at 1 kW. We expect further optimization and friction reduction to enable achievement of the program targets by the end of year 3.

Table 10.1. Engine Configurations and Current Status.

	Concept 0	Concept 1	Concept 3
Displacement (cc)	125	71	70
Bore (mm)	52.4	39.5	40
Stroke (mm)	57.9	57.9	56
Bore/Stroke Ratio (-)	0.91	0.68	0.71
CR	11	17	18
Status	Testing Complete	In-Progress	Future
Peak measured BTE at $\phi = 1.0$	29.4% without EGR 30.6% with EGR	33% without EGR 34.1% with EGR	TBD

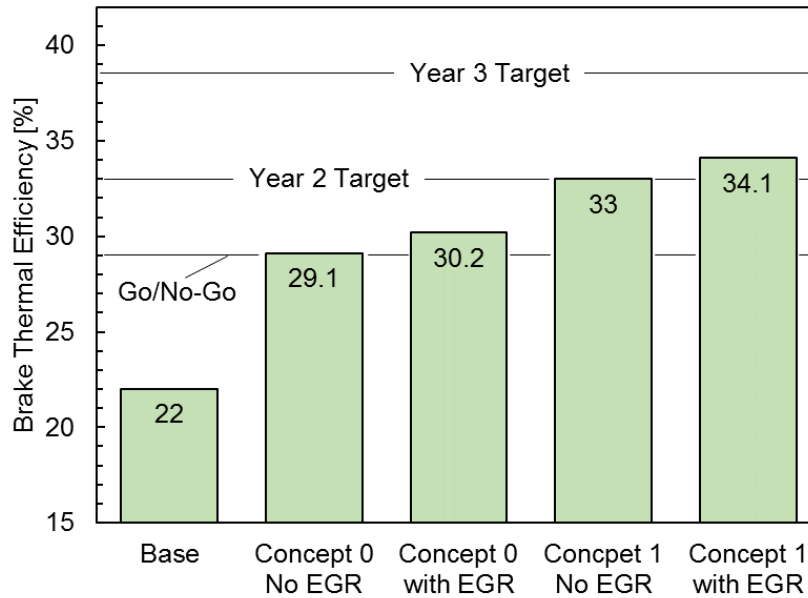


Figure 10.2. Peak brake thermal efficiency (BTE) measured during program.

A three way catalyst (TWC) will be required to achieve the desired emissions targets. To evaluate conversion efficiency at the exhaust temperatures present in the existing test engine, a commercial catalyst that was originally a part of the baseline off-the-shelf engine system was implemented after the engine exhaust and post catalyst emissions were measured. The engine was operated at 2400 RPM with 0% EGR and a stoichiometric equivalence ratio. The engine was tested on neat methane; accordingly, the engine out non-methane, non-ethane VOC emissions are negligible (regulated per CARB 2007 emissions limits for distributed power generation). Accordingly, the present study focuses on CO emissions. The results are shown in Table 10.2. The off the shelf catalyst reduces CO by 99.6% to values below the current program targets. These experiments show that the exhaust temperatures are sufficient for catalyst operation.

Table 10.2. Key Brake Specific Engine Emissions Pre- and Post-Catalyst.

	Pre-Catalyst Emissions [g/kW-hr]	Post-Catalyst Emissions [g/kW-hr]	Target Emissions [g/kW-hr]	Catalyst Conversion Efficiency [%]
Carbon Monoxide	2.85	0.01	0.045	99.58

A custom catalyst is being produced to target the emissions of the Concept 1 and Concept 3 engine. To start, a catalyst support structure was sized using a target space velocity methodology. The space velocity is a function of the engine geometry and flow rates and is used to determine the volume of catalyst structure needed to effectively convert the mass flowing through the catalyst.

The program targets require operation on methane numbers from 70 to 100. The previous experiments were performed with a methane number of 100. To evaluate the methane number effects, experiments were performed using several different blends of propane and methane. The methane number was

varied from 100% to 68% and was calculated using the correlation proposed by Ryan et al. [7]. This correlation was selected because it was specifically developed to cover high concentration propane blends at a stoichiometric equivalence ratio. The engine was operated at 2400 rev/min without EGR. At each methane number, the spark timing was swept to find the knock limited spark advance (KLSA) and the maximum brake torque (MBT). Figure 10.3 shows the spark timing of knock onset and MBT as a function of methane number. The results show that the engine can maintain MBT timing over the entire methane number range. That is, the KLSA is always advanced of MBT timing. Figure 10.4 shows the brake thermal efficiency as a function of methane number. A BTE of over 33% is maintained over the entire span of methane numbers evaluated. In fact, the BTE increases slightly with decreasing methane number. This can be attributed to shorter combustion duration resulting from the higher laminar flame speed of propane.

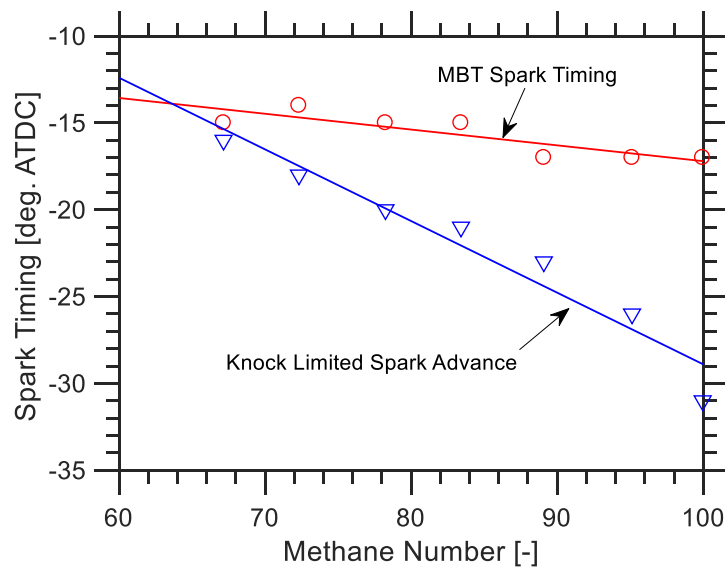


Figure 10.3. Knock limited spark advance and maximum brake torque (MBT) spark timing as a function of methane number.

⁷ Ryan, T.W., Callahan, T.J., and King, S.R., "Engine Knock Rating of Natural Gases – Methane Number", Journal of Engineering for Gas Turbines and Power, 115:769-776, 1993.

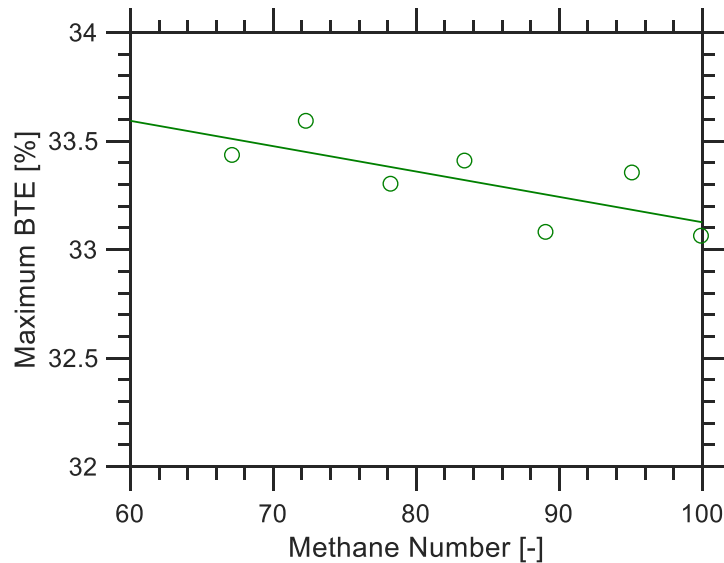


Figure 10.4. BTE as a function of methane number without EGR.

Engine Testing of Thermal Barrier Coatings

Experiments to test the thermal barrier coatings were conducted on a modified version of the Concept P0 engine due to the availability of parts to apply the coatings since the Concept 1 engine had a limited number of prototype pistons and cylinder liner-blocks machined. Domed pistons were installed in the Concept 0 engine to increase the compression ratio from the stock value of 11:1 to 17:1 while maintaining the stock bore and stroke. Increasing the compression ratio to 17:1 allowed the investigation of the coatings at the high compression ratio environment that is expected in the Concept 3 engine and is detailed in SAE Paper 2019-01-0228. Adiabatics Inc applied the PC and the RC coatings to the tops of the high compression ratio pistons at a target thickness of 0.1 mm as seen in Figure 10.5. The RC coating was also applied to the head of the engine. A material properties comparison of the PC and RC coating to the cast head and forged piston is shown in Table 10.3. Both coatings have a significantly lower thermal conductivity and heat capacity than either of the aluminum alloys of the uncoated engine. As a result, the PC and RC coated piston engine configurations should exhibit lower heat loss, higher exhaust loss, and higher thermal efficiency.

Table 10.3. Comparison of engine material and Adiabatic's coating properties.

Coating	Head 319-T5 Al	Piston 4032 Al	PC	RC
Thermal Conductivity [W/m-K]	109	141	0.28	0.28
Heat Capacity@ 300C [kJ/m ³ -K]	2687	2315	650	500
Density [g/cm ³]	2.79	2.68	0.60	0.45
Nominal Coating Thickness [mm]	-	-	0.81	0.91

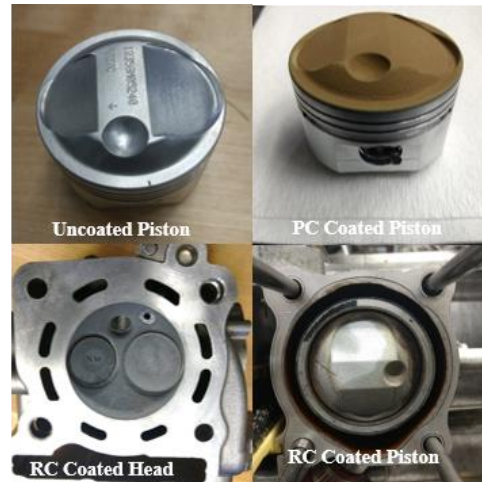


Figure 10.5. Engine operating conditions.

The application of the coatings to the engine parts led to four main experiments: (1) Uncoated engine, (2) PC coated piston, (3) RC coated piston, and (4) RC coated head and piston. For each experiment, the engine was operated at 1400 rev/min. with methane number swept from 100 to 60 using an increasing propane fraction. At each methane number for each experiment, spark timing was swept to find maximum brake torque (MBT) timing and knock limited spark advance (KLSA) timing. Figure 10.6 shows the brake thermal efficiency and energy balance-based heat loss. It can be seen that the coatings improve the overall brake thermal efficiency compared to the uncoated engine and reduce the overall heat loss. The coatings improvements in efficiency follow the predicted trend based on the coating's material properties. Since the RC coating has a 23% lower heat capacity than the PC coating, it was expected to have improved heat retention in the cylinder. The extension of the RC coating to the head further improves brake efficiency while reducing heat losses.

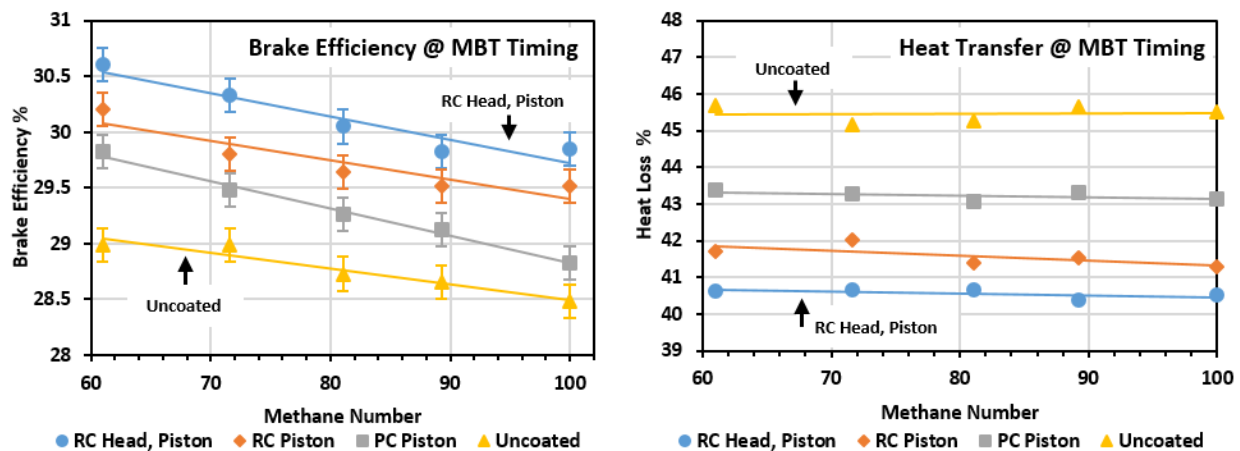


Figure 10.6. MBT timing brake thermal efficiency (left) and heat loss (right) for each coating across methane numbers.

Task 11 - Validated GT-Power model with refined/enhanced sub-models

Summary

The previously developed GT-Power model from Task 3 was updated to match test engine setup and validated to relevant test data (e.g., system pressures, temperatures, flow rates, indicated, and brake performance). The test engine is referred to as Concept1. All metrics except exhaust temperature were validated within 3% of test data. Exhaust temperature validated within 100K of thermocouple reading. This was completed by end of project Q7, satisfying Milestone M11.1, “Validated GT-Power model with refined/enhanced sub-models. All metrics except exhaust temperature will be validated within 3% of test data. Exhaust temperature validated within 100K of thermocouple reading.”.

Details

The GT-Power model from Task 3 was updated to match Concept1 test engine setup at UW-ERC. A picture of the updated GT-Power model is shown in Figure 11.1.

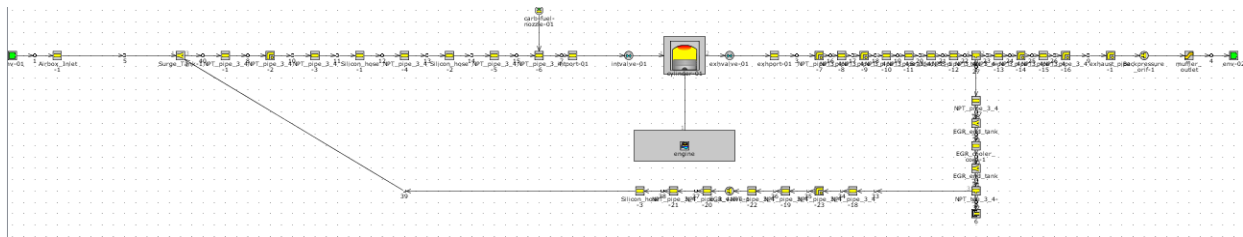


Figure 11.1. GT-Power model of Concept1 test engine.

The GT-Power model includes detailed intake and exhaust system as well as EGR system. Concept1 key engine dimensions are summarized in Table 11.1. Concept2 and Concept3 are included in the table for comparison.

Table 11.1. Engine concept key dimension summary.

	Concept1	Concept 2	Concept3
Displacement	71 cc	70 cc	70 cc
Bore	39.5	40 mm	40 mm
Stroke	57.9	56 mm	56 mm
Bore/Stroke Ratio	0.68	0.71	0.71
CR	17.5:1	18:1	18:1
Intake Seat Dia.	17.0 mm	14.5 mm	14.5 mm
Exhaust Seat Dia.	12.5 mm	12.5 mm	12.5 mm
Valve included angle	43deg	43deg	29deg

The model was validated to relevant test data (e.g., system pressures, temperatures, flow rates, indicated, and brake performance). During the validation process, friction, combustion, flow loss and heat transfer sub-models were adjusted to match test data. Sub-model development is based on the full dataset without individual point "tuning". This generally allows for better future model prediction and extrapolation.

All metrics except exhaust temperature had a validation target of model matching within 3% of test data. Exhaust temperature validation metric is within 100K of thermocouple reading.

The engine was run with methane and the test data provided was a spark sweep at 1800rev/min, Lambda=1, 0% EGR and 1 bar inlet pressure (no boost). Figures 11.2-11.5 compare validated GT-Power model and test data. For test data, 3% error bars have been included except exhaust temperature where a +100°C error bar is included.

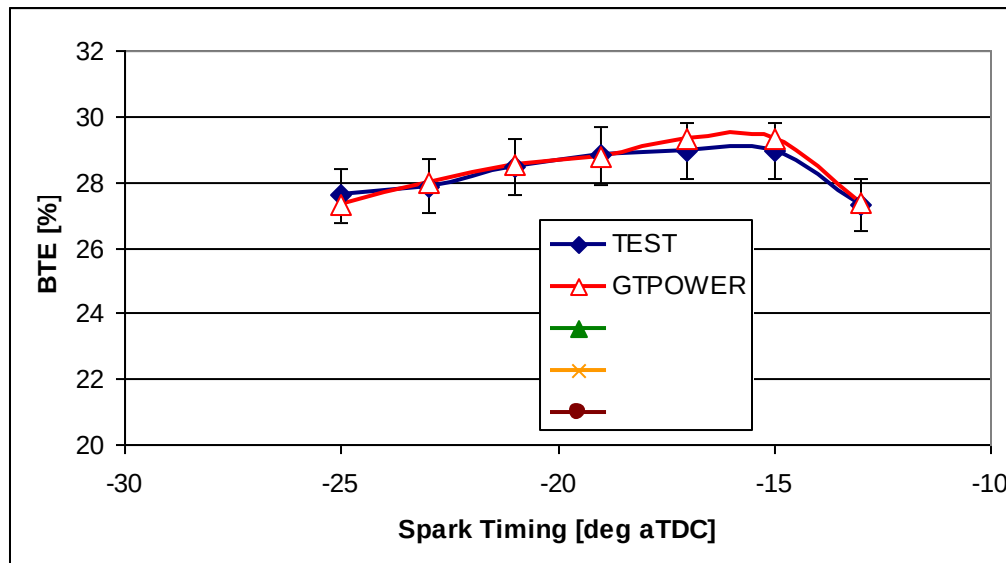


Figure 11.2. BTE results of validated GT-Power model compared to test data.

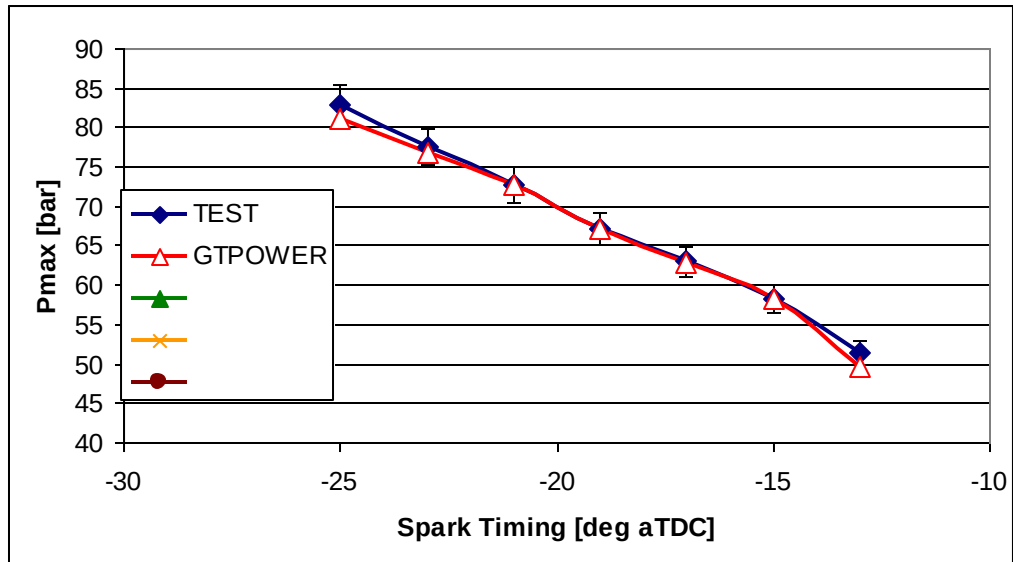


Figure 11.3. Cylinder Pmax results of validated GT-Power model compared to test data.

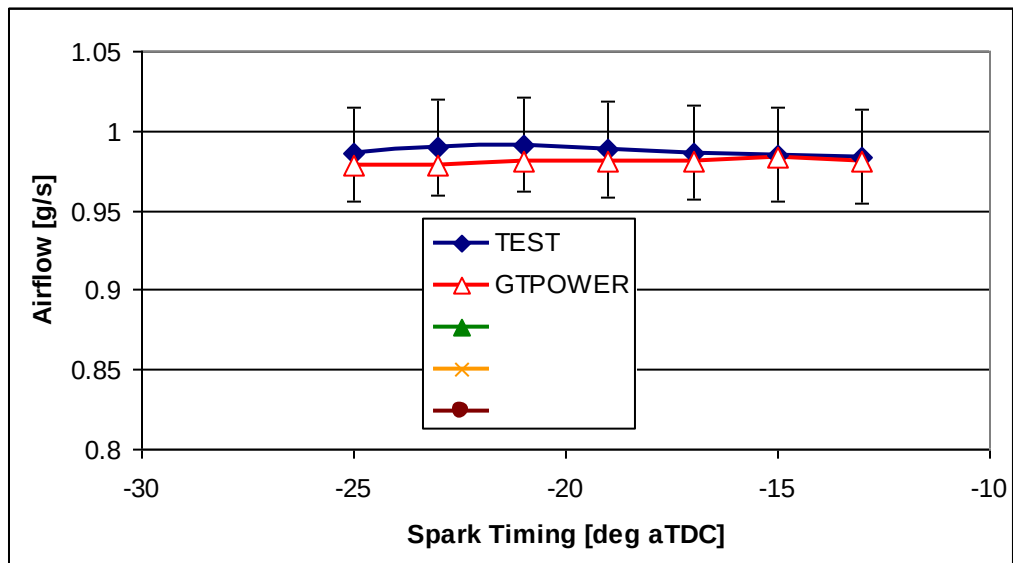


Figure 11.4. Airflow results of validated GT-Power model compared to test data.

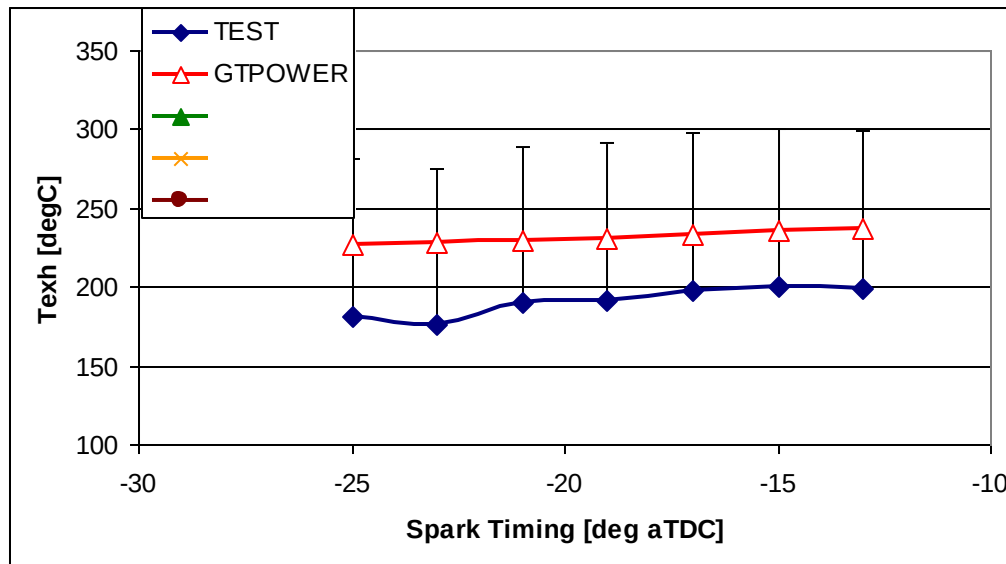


Figure 11.5. Exhaust temperature results of validated GT-Power model compared to test data.

The GT-Power model is matched very well to the test data considered with all validation metrics meeting Milestone 11.1 objectives. Milestone 11.1 is stated below.

Milestone 11.1. Validated GT-Power model with refined/enhanced sub-models. All metrics except exhaust temperature will be validated within 3% of test data. Exhaust temperature validated within 100K of thermocouple reading.

Task 12 - GT-Power Optimization Using Validated Model

Summary

The validated GT-Power model from Task 11 was used to perform a similar design/calibration optimization study as was performed in Task 3.

The optimization based on Concept1 validated GT-Power model did not provide sufficient evidence to deviate from the current proposed design. Further, the latest DoE/Optimization study confirmed the requirement for improved combustion and TBCs. The overwhelming key metric was improvement in combustion system performance as is expected in Concept3 design. With current design path and relaxed calibration requirements compared to previously estimated (1.2bar vs. 1.5bar boost and 25% vs 30% EGR), target engine BTE of 38.5% appears achievable. This work was completed by end of project Q8, satisfying Milestone M12.1, "Updated base engine design and operating parameters to achieve 40% electrical conversion efficiency. The engine design parameters will potentially include engine displacement, bore, stroke, engine speed, compression ratio, valve timings, valve lift profiles, valve diameter, required intake pressure and temperature, EGR level, intake and exhaust geometries, identification of heat loss reduction required from TBCs, boosting system specification, combustion duration and phasing.".

Details

The validated GT-Power model from Task 11 was used to perform a similar design/calibration optimization study as was performed in Task 3.

A GT-Power model was built and validated to test cell data taken at UW-ERC of the Concept1 engine. The first step was to simplify and parameterize the model to facilitate a large scope Design of Experiments (DoE). A comparison of the full and simplified models can be seen in Figure 12.1.

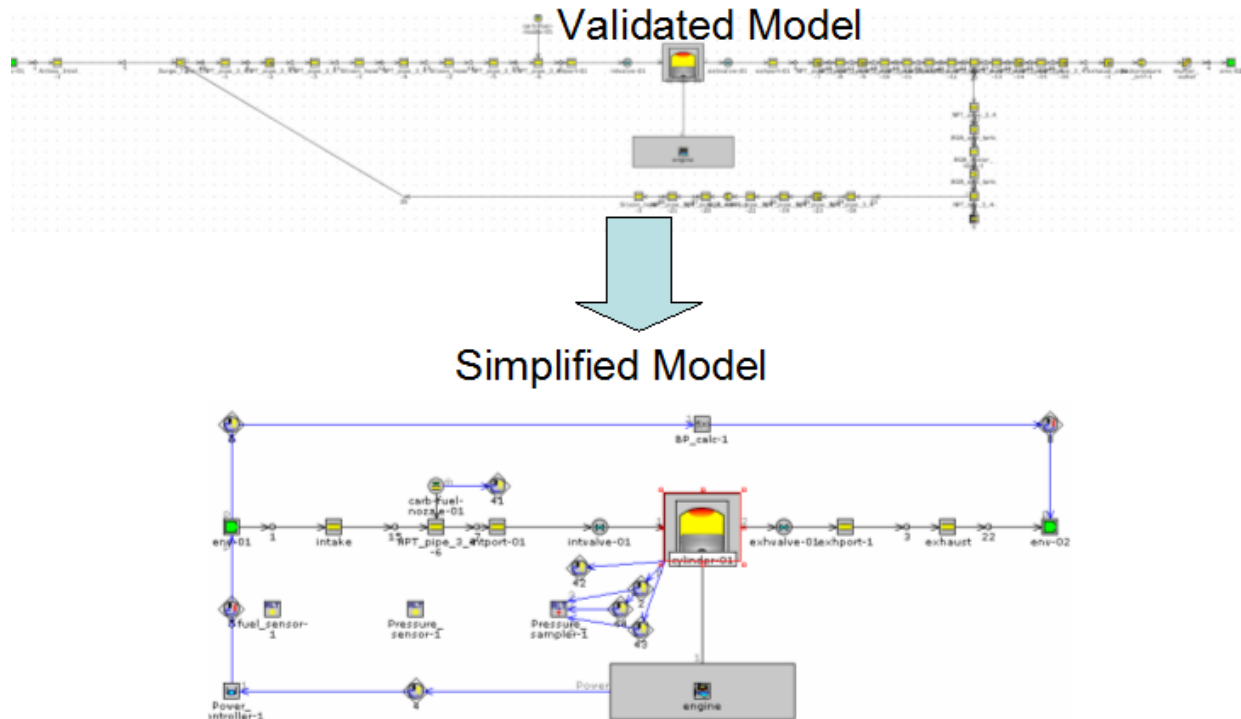


Figure 12.1. Comparison of full and simplified GT-Power models.

Some key features of the simplified model include:

- EGR system removed and replaced with inlet boundary condition input.
- Intake and exhaust simplified. Volume and effective length maintained, but intake and exhaust represented as single component.
- AFR fixed at stoichiometric.
- Chen-Flynn friction correlation based on stabilized Concept1 engine friction measurements.
- Target power achieved by controlling boost pressure.

A broad DoE of design and calibration parameters was performed. This included the variables and ranges shown in Table 12.1. The design space was a Latin Hypercube space-filling DoE where a portion of the design space is shown in Figure 12.2.

Table 12.1. DoE variables and ranges.

Variable	Description	Units	min	max
BS_RATIO	Bore/stroke ratio	-	0.5	0.9
DISP	Displacement	cc	50	90
EVO	Exhaust valve opening phase	deg aTDC	60	110
EDUR	Exhaust cam duration scale	-	0.9	1.1
IVO	Intake valve opening phase	deg aTDC	270	310
IDUR	Intake cam duration scale	-	0.9	1.1
CR	Compression ratio	-	16	18
EGR	Exhaust gas recirculation rate	-	0	0.3
RPM	Engine speed	rev/min	1500	3000
TINLET	Intake gas temperature	degC	40	80
CA50	Combustion 50% burn phasing	deg aTDC	8	16

The DoE described was performed for 3 different development levels. The first, or Base, includes no changes to sub-models derived during Task 11 validation. This includes a combustion model with 90% combustion efficiency (lowest measured) and burn rate model dependent on engine speed and EGR rate. The second, or Comb, includes an assumption that the combustion system will be improved including 98% combustion efficiency and 20deg burn duration. The third development level builds on the Comb level, but adds Development level thermal barrier coatings (TBCs) as previously described. This level will be referred to as TBC for simplicity.

Neural network models were fit to predicted BTE, boost and cylinder pressure from GT-Power model. The neural network model included coefficients for linear, quadratic and interaction effects as well as bias and weighting.

The neural network model was used to optimize various combinations of displacement and B/S ratio. Discrete combinations of displacement and B/S ratio were set and other parameters were optimized for maximum BTE relative to certain constraints. These constraints include.

- Variable boost limit as described below.
- 25% EGR limit. Experience based combustion stability limit.
- 50'C inlet temperature.

Additionally, for each of the 3 development levels, 4 optimization scenarios were considered.

- Opt1. All variables optimized with 1.5bar boost limit.
- Opt2. All variables optimized with 1.2bar boost limit.
- Opt3. Concept1 cam duration/phasings, all other variables optimized with 1.5bar boost limit.
- Opt4. Concept1 cam duration/phasings, all other variables optimized with 1.2bar boost limit.

Table 12.2-12.4 summarize the optimum for each of the scenarios described with Concept1 engine bore/stroke ratio and displacement shown in yellow for comparison.

Table 12.2. Base engine optimization results with comparison to Concept1 bore/stroke ratio and displacement.

	Units	Opt1	Opt1-Con1 Eng	Opt2	Opt2-Con1 Eng	Opt3	Opt3-Con1 Eng	Opt4	Opt4-Con1 Eng
PREDICTED BEFF	%	34.31	34.18	33.65	33.09	33.71	33.62	33.13	33.09
PREDICTED PCYL	bar	85.87	83.16	70.98	76.87	97.21	81.90	80.72	76.49
PREDICTED PINLET	bar abs.	1.5	1.5	1.2	1.2	1.5	1.5	1.2	1.2
INPUTS									
BS_RATIO	-	0.6	0.7	0.6	0.7	0.55	0.7	0.6	0.7
DISP	cc	60	70	70	70	65	70	75	70
EVO	deg aTDC	93	94	94	99	85	85	85	85
EDUR	-	0.90	0.90	0.90	0.90	1.00	1.00	1.00	1.00
IVO	deg aTDC	270	270	270	290	296	296	296	296
IDUR	-	1.04	1.03	1.02	1.00	1.00	1.00	1.00	1.00
CR	-	16.00	16.00	16.00	16.00	18.00	16.13	18.00	17.14
EGR	-	0.14	0.23	0.07	0.15	0.25	0.25	0.18	0.14
RPM	rev/min	2078	2010	2115	2234	2089	2016	2114	2198
TINLET	degC	50	50	50	50	50	50	50	50
CA50	deg aTDC	16.00	16.00	16.00	12.40	16.00	16.00	16.00	16.00

Table 12.3. Improved combustion engine optimization results with comparison to Concept1 bore/stroke ratio and displacement.

	Units	Opt1	Opt1-Con1 Eng	Opt2	Opt2-Con1 Eng	Opt3	Opt3-Con1 Eng	Opt4	Opt4-Con1 Eng
PREDICTED BEFF	%	38.11	37.79	37.52	37.03	38.01	37.46	37.37	37.24
PREDICTED PCYL	bar	92.58	74.74	75.03	67.98	93.13	80.41	72.33	72.07
PREDICTED PINLET	bar abs.	1.5	1.5	1.2	1.2	1.5	1.364581812	1.2	1.2
INPUTS									
BS_RATIO	-	0.5	0.7	0.65	0.7	0.5	0.7	0.65	0.7
DISP	cc	50	70	60	70	50	70	65	70
EVO	deg aTDC	74	93	80	88	85	85	85	85
EDUR	-	0.99	1.00	1.03	1.01	1.00	1.00	1.00	1.00
IVO	deg aTDC	270	310	310	310	296	296	296	296
IDUR	-	1.09	1.10	0.90	1.04	1.00	1.00	1.00	1.00
CR	-	18.00	18.00	18.00	18.00	18.00	18.00	18.00	18.00
EGR	-	0.25	0.25	0.25	0.17	0.25	0.25	0.25	0.23
RPM	rev/min	2519	2116	2445	2260	2390	2023	2381	2208
TINLET	degC	50	50	50	50	50	50	50	50
CA50	deg aTDC	15.18	16.00	16.00	15.51	15.50	16.00	16.00	16.00

Table 12.4. TBC engine optimization results with comparison to Concept1 bore/stroke ratio and displacement.

	Units	Opt1	Opt1-Con1 Eng	Opt2	Opt2-Con1 Eng	Opt3	Opt3-Con1 Eng	Opt4	Opt4-Con1 Eng
PREDICTED BEFF	%	39.47	39.26	38.91	37.84	39.29	38.84	38.66	38.51
PREDICTED PCYL	bar	80.04	76.41	59.72	72.86	94.41	79.05	71.74	74.39
PREDICTED PINLET	bar abs.	1.5	1.5	1.2	1.2	1.5	1.31	1.2	1.2
INPUTS									
BS_RATIO	-	0.7	0.7	0.85	0.7	0.6	0.7	0.75	0.7
DISP	cc	55	70	75	70	50	70	75	70
EVO	deg aTDC	92	99	92	91	85	85	85	85
EDUR	-	0.97	0.97	0.98	0.99	1.00	1.00	1.00	1.00
IVO	deg aTDC	310	310	310	310	296	296	296	296
IDUR	-	1.10	1.10	1.10	1.08	1.00	1.00	1.00	1.00
CR	-	17.74	17.81	17.52	17.00	17.55	17.55	17.29	17.66
EGR	-	0.25	0.25	0.25	0.15	0.25	0.25	0.25	0.21
RPM	rev/min	2423	1990	2326	2250	2289	1987	2014	2072
TINLET	degC	50	50	50	50	50	50	50	50
CA50	deg aTDC	15.94	16.00	16.00	12.00	15.80	16.00	16.00	16.00

The optimization based on Concept1 validated GT-Power model did not provide sufficient evidence to deviate from the current proposed design. Further, the latest DoE/Optimization study confirmed the requirement for improved combustion and TBCs. The overwhelming key metric was improvement in combustion system performance as is expected in Concept3 design.

This work satisfied Milestone 12.1, Updated base engine design and operating parameters to achieve 40% electrical conversion efficiency. The engine design parameters will potentially include engine displacement, bore, stroke, engine speed, compression ratio, valve timings, valve lift profiles, valve diameter, required intake pressure and temperature, EGR level, intake and exhaust geometries, identification of heat loss reduction required from TBCs, boosting system specification, combustion duration and phasing.

Task 13 - CFD Model Validation to Measured Data from Phase I Prototype CHP Engine

Summary

Task 13 was focused on validating the CFD model to Concept1 test data. Key metrics include peak pressure within 10%, IMEP within 5%, combustion efficiency within 3% and NOx within 20%. Three operating conditions, 2000, 2300, and 2400rev/min, representing a speed sweep of Concept1 test data was used as the basis of model validation. The key metrics at the modeled operating conditions were within the SOPO targets. This work was completed by end of project Q9, satisfying Milestone M13.1, “The CFD model capable of reproducing the measured emissions, performance, and knocking characteristics of the phase I prototype CHP engine. Prediction of peak cylinder pressure within 10%, IMEP within 5%, combustion efficiency within 3%, NOx emissions within 20%, onset of knock within 3 crank angle degrees (spark timing).”.

Details

The goal of Task 13 is to use Task 10 data and Task 11 boundary conditions to validate the CFD model used to optimize the combustion system. Comparisons between the measured and predicted emissions, performance (e.g., thermal efficiency), and knock onset will be used to ensure the model is suitable for use in engine concept evaluation and optimization.

Figure 13.1 shows the computational model of the Concept1 engine used for the validation activity.

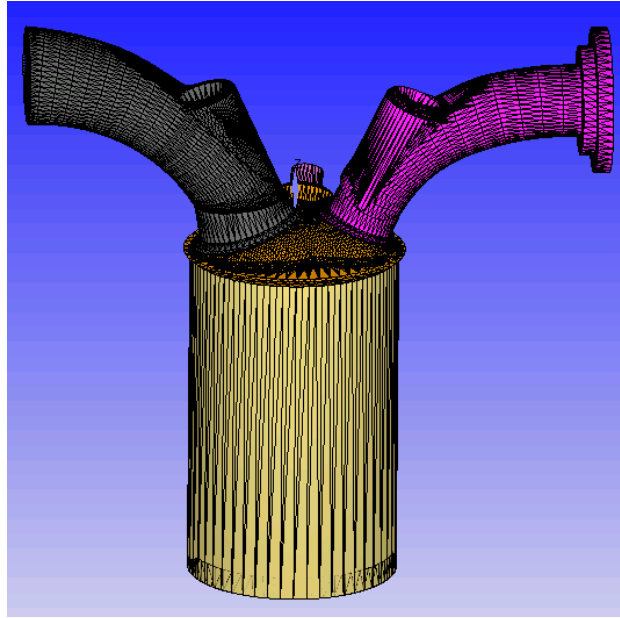


Figure 13.1. Concept1 computational model.

Initial simulations were focused on a hot motoring case. Figure 13.2 shows cylinder pressure comparison of a test and modeled hot motored case.

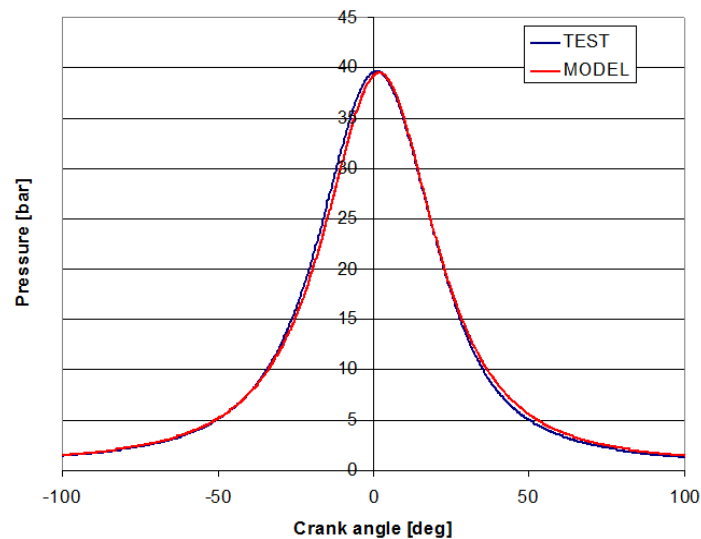


Figure 13.2. Comparison of test and modeled cylinder pressures for hot motored case.

Reasonable match on hot motored case. The model settings were carried forward to use for modeling fired test cases.

The validated GT-Power model previously reported in Task 11 was used to generate boundary conditions for modeling fired test cases. Test data modeled was MBT spark timing across an engine speed range at target brake power. The GT-Power results can be seen in Figures 13.3 and 13.4.

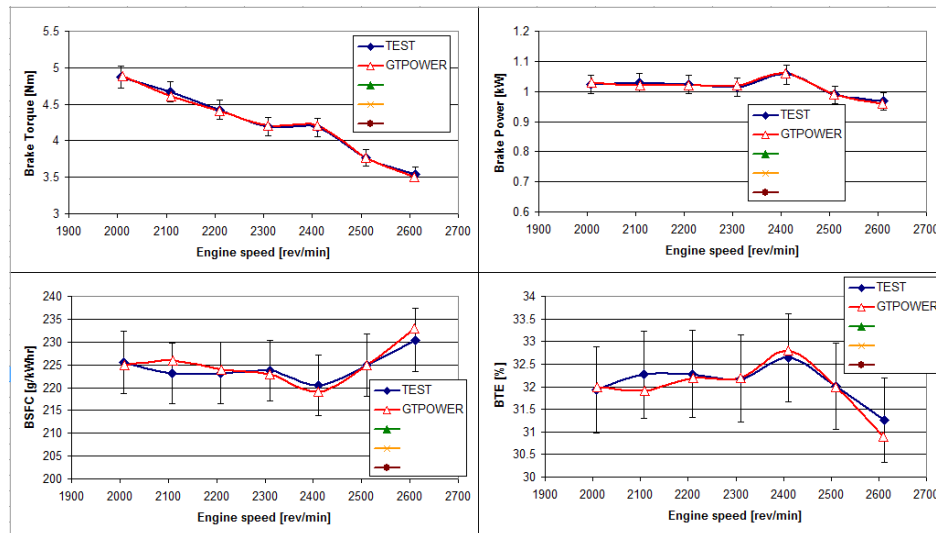


Figure 13.3. GT-Power model brake performance results.

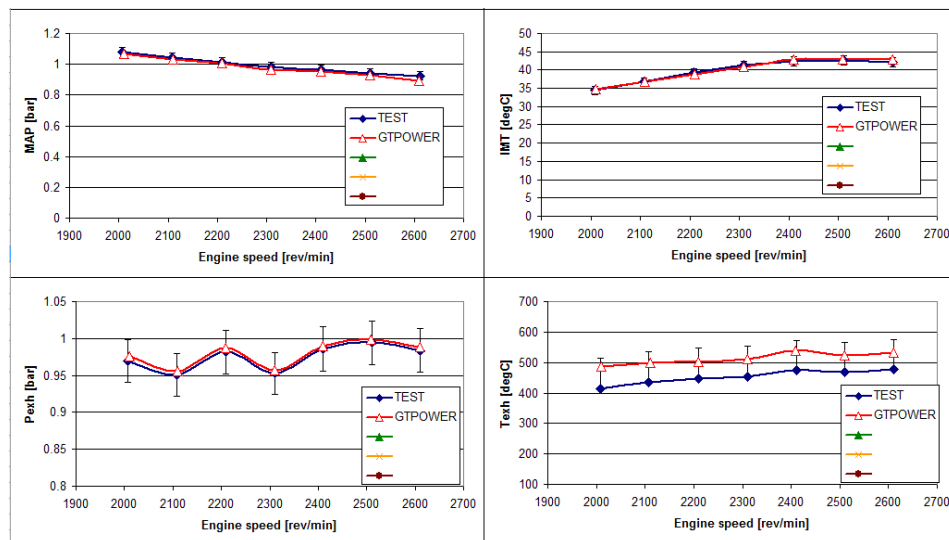


Figure 13.4. GT-Power model pressure and temperature results.

GT-Power predicted port pressure, port temperature and AFR were used as boundary conditions for CFD simulations. Three engine speeds were chosen to model including 2000, 2300 and 2400 rev/min.

The summary results with comparison to SOPO targets for Milestone 13.1 are shown in Table 13.1 and Figure 13.4.

Table 13.1. CFD Results compared to SOPO Milestone 13.1 targets.

	Engine Speed	IMEP Net	Pmax	Comb Eff.	NOx
	rev/min	bar	bar	%	g/kW-hr
TEST	2000	9.46	76.12	89.99	25.20
	2300	8.31	61.12	92.40	20.96
	2400	8.20	57.40	93.59	17.14
MODEL	2000	9.93	76.46	89.38	27.49
	2300	8.57	60.97	91.96	22.24
	2400	8.34	57.42	90.92	16.42
ERROR	2000	4.70%	0.44%	0.69%	8.32%
	2300	2.94%	0.25%	0.48%	5.77%
	2400	1.66%	0.02%	2.94%	4.40%
	Error Target	5.00%	10.00%	3.00%	20.00%

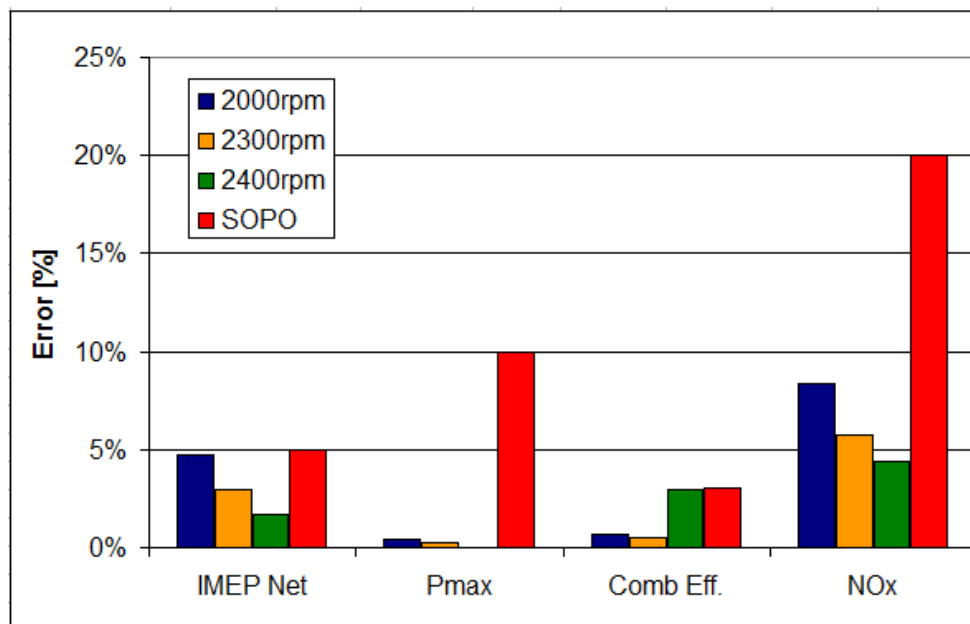


Figure 13.4. CFD results bar chart compared to SOPO Milestone 13.1 targets.

CFD results are reasonably matched to test data and satisfy Milestone M13.1, The CFD model capable of reproducing the measured emissions, performance, and knocking characteristics of the phase I prototype CHP engine. Prediction of peak cylinder pressure within 10%, IMEP within 5%, combustion efficiency within 3%, NOx emissions within 20%, onset of knock within 3 crank angle degrees (spark timing).

Task 14 - Computational Optimization of First Generation Prototype CHP Engine

Summary

Task 14 was focused on optimization of the prototype CHP engine through CFD. The validated model from Task 13 was used to investigate spark pre-chamber impact on performance. Simulations show issue with 0.5mm diameter nozzle holes where ignition seems to occur every other engine cycle. Simulations with 0.8mm nozzle holes ignited every cycle. Comparison of trapped residuals in pre-chamber showed similar pre-chamber filling for both designs. Comparison of TKE in pre-chamber showed higher velocities in 0.5mm design at ignition timing. Visualization of combustion process seems to indicate initial flame kernel is being "blown" out in 0.5mm design.

Pre-chamber and conventional sparkplug models were validated to hot motored test data. A model with ring leakage was implemented to help match expansion pressure. Excellent agreement was achieved for both pre-chamber and conventional sparkplug models.

Investigation into port development was performed with focus on TKE at spark timing. Three design variants were considered with none outperforming the baseline design. Qualitative analysis was performed on flow structure and compared to TKE numbers. Flow features likely responsible for undesirable TKE destruction were identified.

Computational analysis with conjugate heat transfer of the conventional spark plug configuration was performed with and without RC thermal barrier coating. Results indicate that addition of the RC coating allowed spark timing to be advanced at the same or lower knock intensity. Improved ISFC was found with RC TBC coated piston and head. Piston and head wall temperatures were shown to track more closely with bulk gas temperature compared to non-coated parts resulting in reduced heat transfer. A pathway to improved efficiency was found with TBC's. This work was completed by end of project Q14, satisfying Milestone 14.1, "Computational analysis of the First-Generation Prototype CHP engine is complete. Pathways to further engine performance improvement will be identified."

Details

In support of testing development, focus was put on modeling spark pre-chamber ignition. Testing and modeling considered passive pre-chamber with surface gap spark plug. Figure 14.1 shows the computational model with spark pre-chamber.

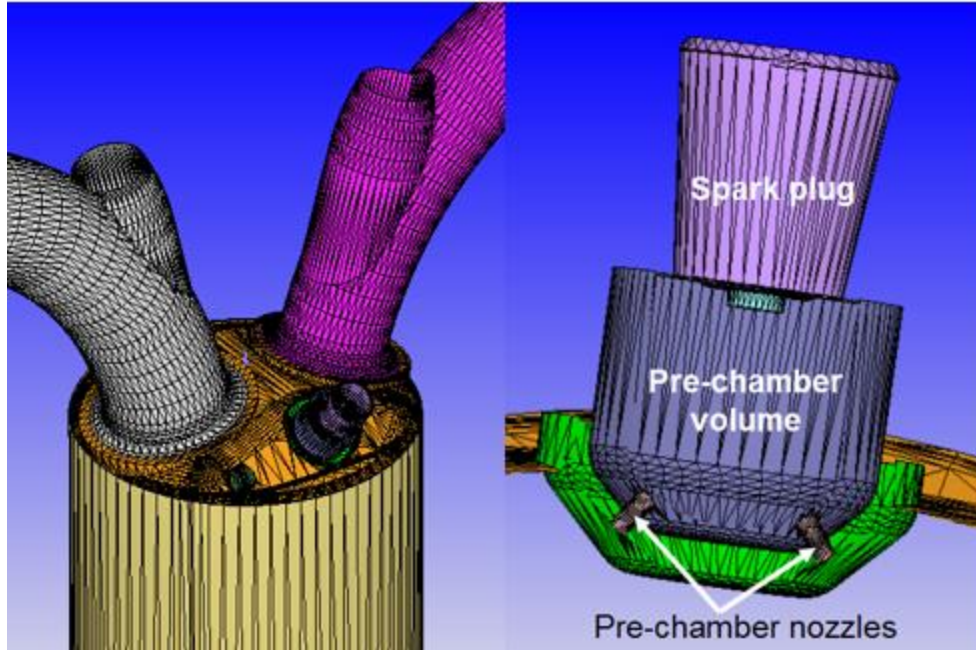


Figure 14.1. CFD model with pre-chamber.

Simulations were performed at 2400rev/min and -17aTDC ignition timing with two different pre-chamber nozzle diameters, 0.5 and 0.8mm. Figure 14.2 shows a comparison of pre-chamber heat release rate (HRR) for 3 consecutive engine cycles.

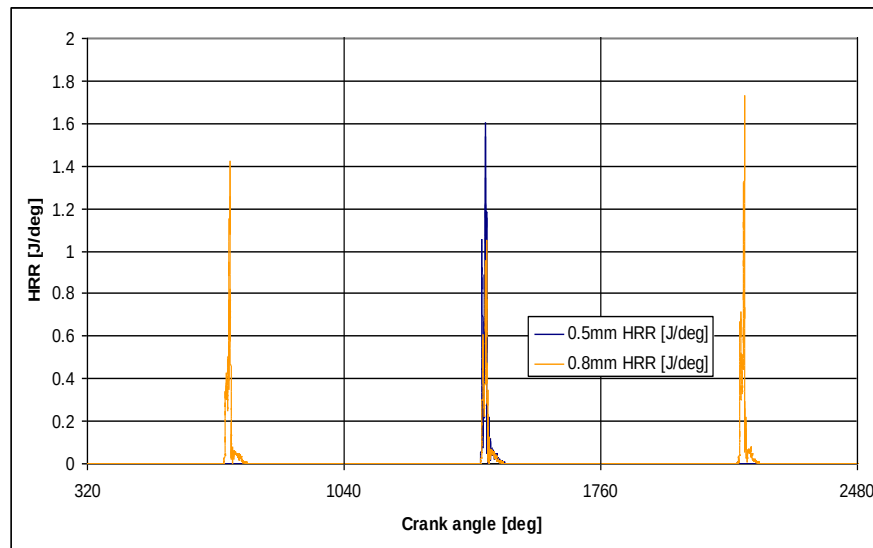


Figure 14.2. 3 cycle heat release rate comparison 0.5 and 0.8mm pre-chamber nozzle holes.

As can be seen in Figure 14.2, the 0.5mm hole pre-chamber only ignites every other engine cycle while the 0.8mm hole ignites every cycle. Analysis of TKE at ignition shows high TKE corresponds to non-ignition cycles as shown in Table 14.1.

Table 14.1. Summary TKE at ignition time for 3 engine cycles.

	TKE [m ² /s ²]	
Cycle	0.5mm	0.8mm
1	29.9	13.6
2	26.7	22.1
3	42.6	15.2

This seems to indicate that the initial flame kernel is being "blown" out by high velocity flow near the spark plug. Figure 14.3 shows a side by side comparison of 0.5 and 0.8mm pre-chambers after 3rd cycle ignition at several crank angles. The figure is a velocity cut-plane through the center of the pre-chamber aligned through one nozzle hole. Also included is a 2300K temperature iso-surface to model the flame front. In each crank angle snapshot, the 0.5mm hole pre-chamber is on the left and the 0.8mm is on the right.

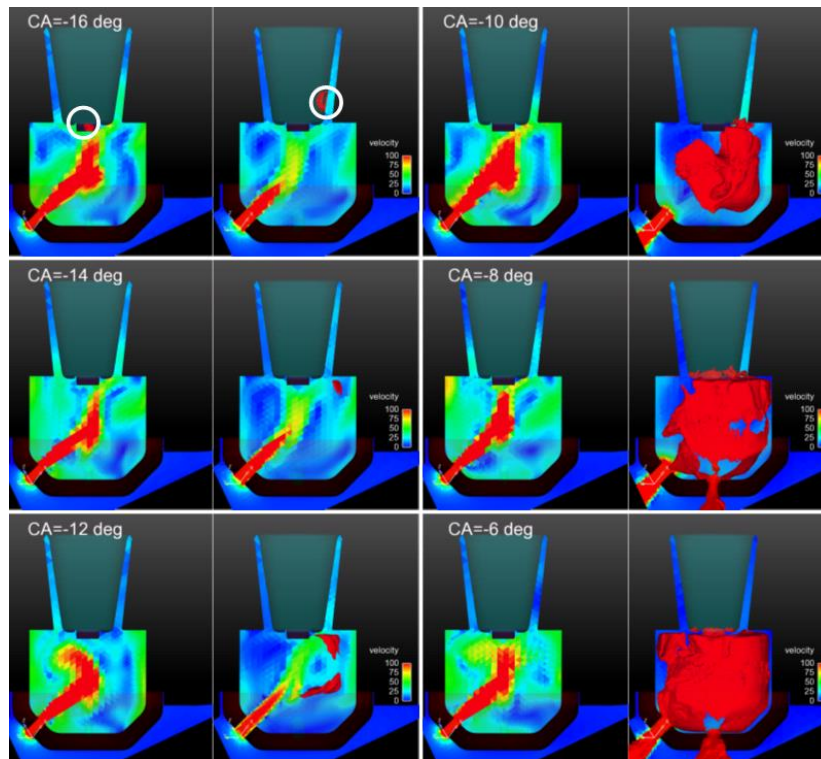


Figure 14.3. Velocity cut-planes with temperature isosurface for pre-chamber combustion.

As can be seen in the second pane, -14deg aTDC, the 0.5mm flame has extinguished explaining simulation results that were verified in test.

Intake port design investigations were conducted where three design concepts were considered with focus on turbulent kinetic energy (TKE) at ~spark timing. Figure 14.4 shows a cutaway view through the center of the intake port for the baseline port and 3 modified versions.

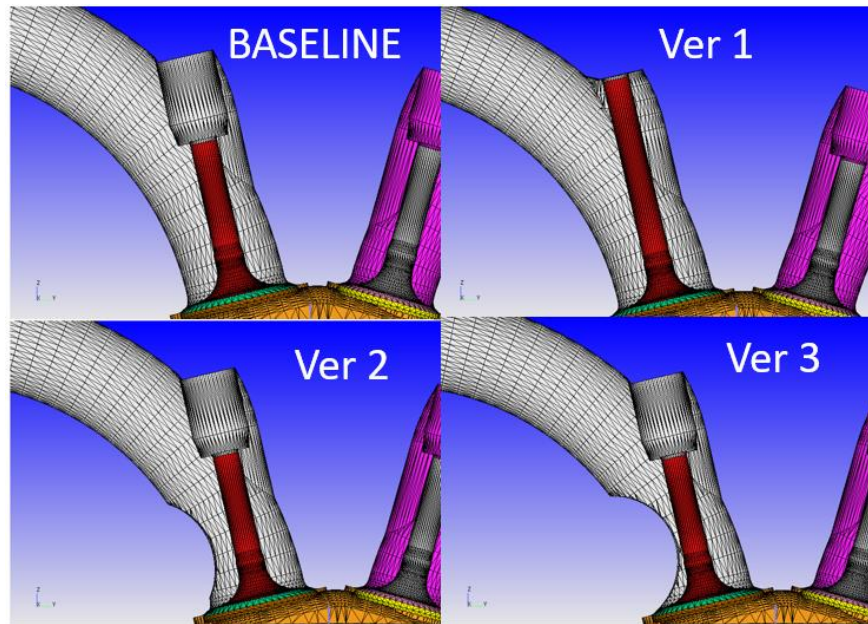


Figure 14.4. CFD model intake ports cutaway view.

For all designs, two complete engine cycles were simulated. The TKE of the second cycle was compared as can be seen in Figure 14.5.

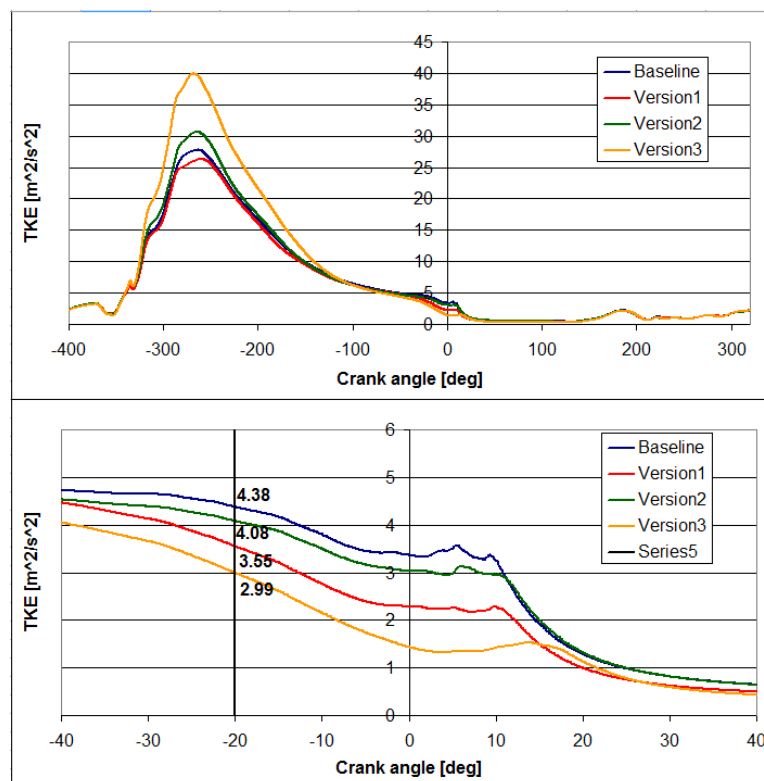


Figure 14.5. TKE comparison of baseline and Versions1-3 port designs.

Figure 14.11 shows TKE over the entire cycle in the upper graph and zoomed in TKE in the lower graph. Peak TKE values for Version2 and Version3 (small and large scallops) were higher than the baseline. However, both designs had more TKE destruction during compression ultimately having lower TKE at spark timing. Version1 (valve guide removed) had similar TKE over the engine cycle when compared to the baseline. However, this design also had lower TKE at spark timing. No ports considered performed better than the baseline.

Spark sweeps (-24 to -18 deg aTDC) were performed with and without TBC. The simulations were performed with stoichiometric methane at 1bar MAP and 3000rev/min. Conjugate heat transfer on the piston and head were used for all simulations. For the coated simulations, .05mm RC coating on the head and piston face were modeled. Figures 14.6 and 14.7 compare wall temperature and heat transfer results.

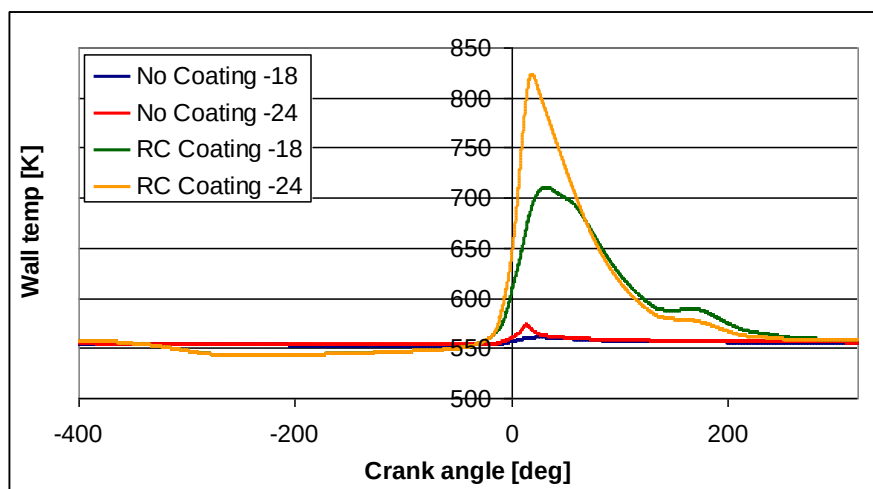


Figure 14.6. Comparison of head wall temperatures.

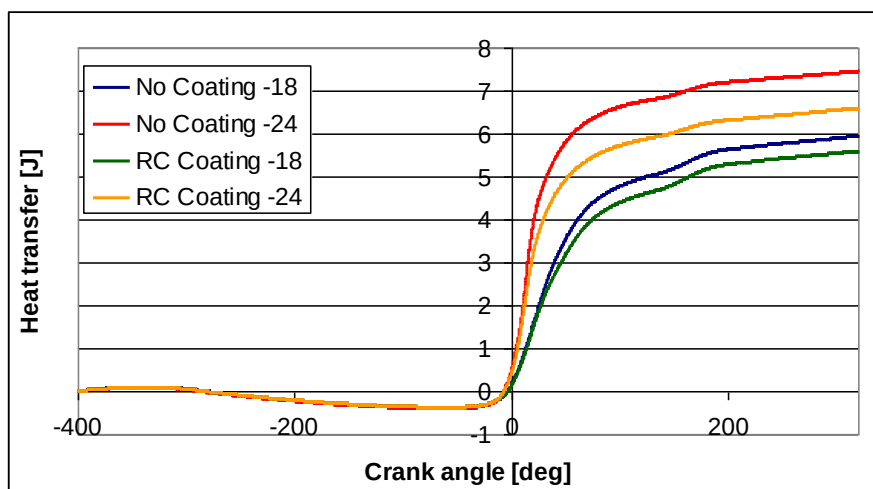


Figure 14.7. Comparison of head heat transfer.

As expected, the coated piston and head surface temperatures track better with the gas temperature while the non-coated parts temperatures are comparatively constant over the engine cycle. This results in decreased heat transfer with comparatively less heat transfer as spark timing is advanced. This explains ISFC that improves with advancing spark timing.

Improved knock sensitivity was seen in test and verified analytically through modeling numerical pressure transducers. Figure 14.8 shows comparative knocking intensity at 4 locations in the head.

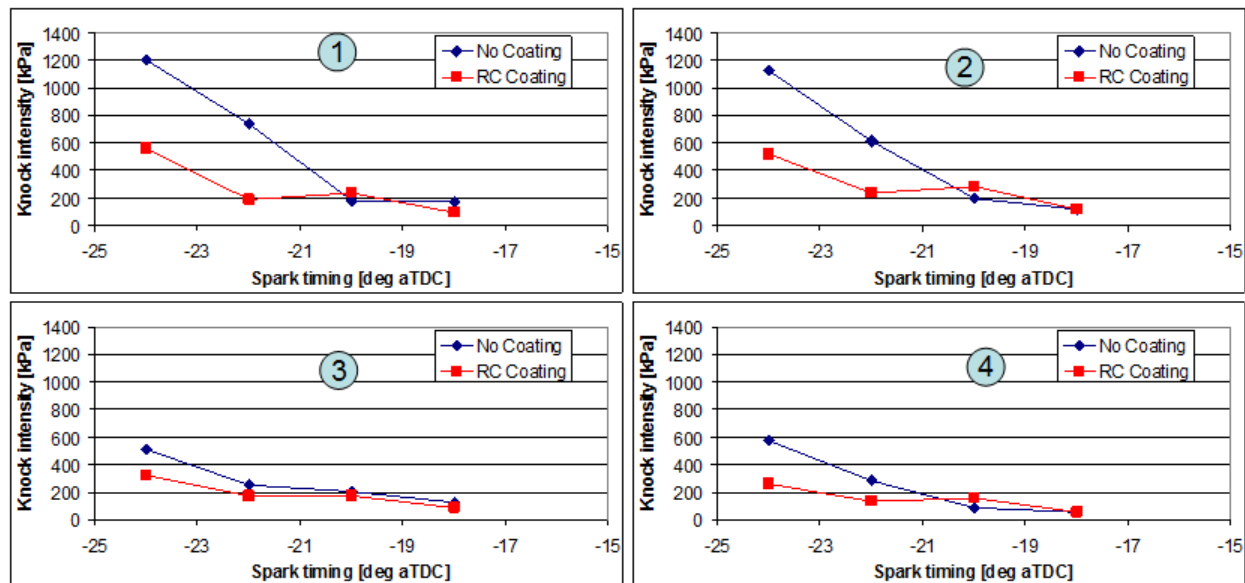


Figure 14.8. Knocking intensity comparison.

Figure 14.8 shows that the predicted knock intensity varies significantly depending on where the transducer is located. Transducers 3 and 4 (test location) had significantly lower knock intensity than the other 2 locations. However, all show that simulations with RC coated parts had lower knock intensity at advanced spark timings that translate to ~2deg additional spark advance before onset of knock.

Design variations as well as TBCs were assessed showing improvement with TBC and magnitude of design change impact on performance satisfying Milestone M14.1, "Computational analysis of the First-Generation Prototype CHP engine is complete. Pathways to further engine performance improvement will be identified."

Task 15 – Durability Testing of Phase 1 Prototype

Summary

The objective of this task is to perform durability testing on at least one engine to evaluate initial concept durability and reliability. At completion of at least 500 hours of operation or until a major engine failure, perform a system tear-down and inspection to measure wear and estimate system life.

Details

Assessment of engine components at assembly and after initial testing yielded several areas that required design improvement. The revisions were thought necessary to improve engine performance (efficiency, emissions) as well as engine durability. Components for the Briggs & Stratton endurance engine and the UW-ERC Phase II performance engine that procured in Q11 are detailed below.

Piston

Piston skirt profile optimization continues to be an iterative process in the engine industry. A third iteration (I3), based on the results from an inspection of the UW-ERC engine (I2 piston), is compared in Figure 15.1. Sample pistons were made available for the endurance engine as well as the UW-ERC performance engine.

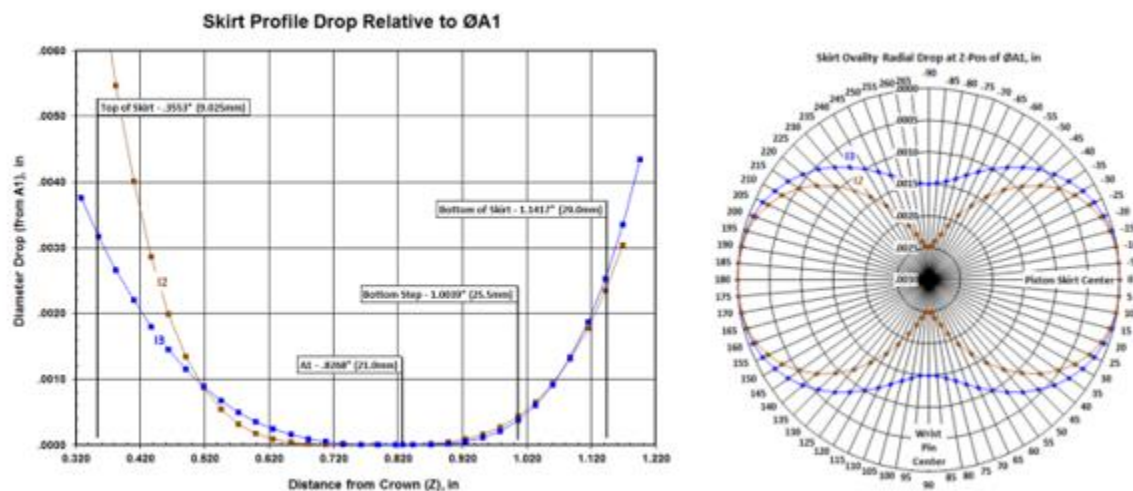


Figure 15.1. Proposed Skirt Profile Changes, I2 to I3.

Camshaft

During the initial engine build, it was discovered that cam lobe locations were not updated after finalizing cam follower designs. This caused intake lobes to be offset from the cam roller followers and a narrow exhaust lobe with overhanging roller followers. Additionally, the cam lobe profile was improved, smoothing out a small concave section at the opening and closing profiles. Cam lobe locations and shapes were corrected and sample cams procured.

Chain Guide

As observed through the Briggs & Stratton windowed engine, the camshaft chain drive system showed good stability but with higher than desired wear on the fixed guide near the crankshaft sprocket. Also, analysis showed that the tensioner adjustment was at the end of travel. The movable guide was redesigned to better utilize the available tensioner travel while the profile was modified to have less contact (lower friction) with the chain. The fixed guide profile was also modified to reduce the contact

“hot spot” near the crankshaft sprocket. Revised guides were 3D printed and an improved material procured for the chain-tensioner wear interface.

Crankshaft Lubrication

During initial motoring tests, an excessive crankcase-crankshaft oil jet was observed (using the windowed crankcase), overflowing the crankshaft slinger system for oiling the main and rod bearings. Significant crankshaft oil entrainment resulted, thought to cause an increase in crankshaft oil windage with resultant increase in engine friction. Based on orifice flow calculations, flow area was reduced by 70% by plugging the original 0.160” slot with a pin and milling a 0.040” wide slot. Flow observation testing confirmed a reduced, yet adequate, oil flow amount.

Crankshaft Design

Engine assembly and initial dynamometer testing showed poor flywheel and encoder shaft alignment to the crankshaft. This was due to the use of flange joints for these shafts and higher than required tolerance levels on mating surfaces. Re-machining of components and select fit assembly provided workable engine assemblies for UW-ERC and Briggs & Stratton. Redesign of the encoder shaft and flywheel joints to a more traditional small engine taper fit was accomplished. Revised crankshaft, flywheel, and encoder shaft components were procured.

Cast Iron Liner

Initial experience with DLC coated cylinder sleeves was not acceptable with thin ID coating and areas of coating delamination. Further analysis and discussion with DLC suppliers provided insight into these issues. Duralar, a DLC suggested vendor from Federal Mogul, stressed that an ID-generated plasma needed to be used for proper adhesion and improved thickness uniformity. As these issues are being worked through, the decision was made to utilize more standard cast iron sleeve material for the Briggs & Stratton endurance engine and the UW-ERC Phase II performance engine.

With the above Phase II upgrades having been made, endurance testing commenced on the Briggs & Stratton dynamometer test engine in July 2018. Operational issues during the test period, and corrective action for these issues, are described below.

Head Gasket

An early hour (26 hours) head gasket combustion leak was discovered after combustion gases were noted in the cooling system. Head removal showed a leakage path between the cylinder and a cylinder head coolant passage, as shown in Figure 15.2. Cylinder assembly measurements showed that the cylinder sleeve was actually below the cylinder deck surface, compromising clamping load in this critical area. Correcting this geometry issue by adjusting the assembly to have the sleeve slightly above the cylinder surface corrected the leakage issue. No replication of head gasket leakage, was witnessed after the sleeve geometry correction.



Figure 15.2. Head Gasket Leakage.

Spark Plug Insulator Erosion

Spark plug insulator erosion was discovered at 132 hours, at 280 hours (148 hours on a new plug), and again at 400 hours (120 hours on a new plug). See Figure 15.3. Radial erosion grooves in the ceramic insulator, possibly consistent with the spark arc were observed. Federal Mogul's Champion Spark Plug group commented that the material loss is caused by the spark arc traveling across the insulator surface to the spark plug shell. Increasing the protrusion of the center electrode, such as available with plug G54V (+0.5mm protrusion), was thought to correct this issue. A comparison of center electrode design is shown in Figure 15.4. Samples were not made available in time to evaluate for improvement.

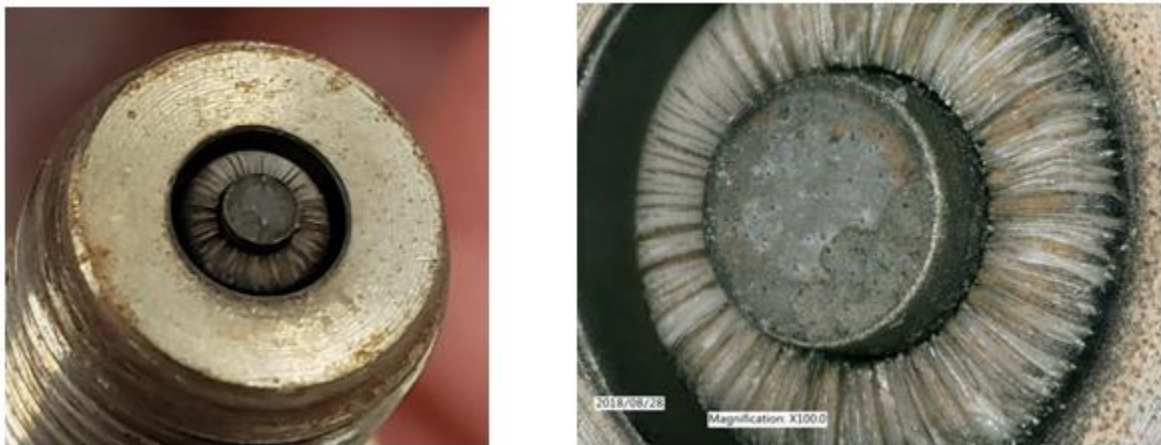


Figure 15.3. Spark Plug Insulator Erosion.

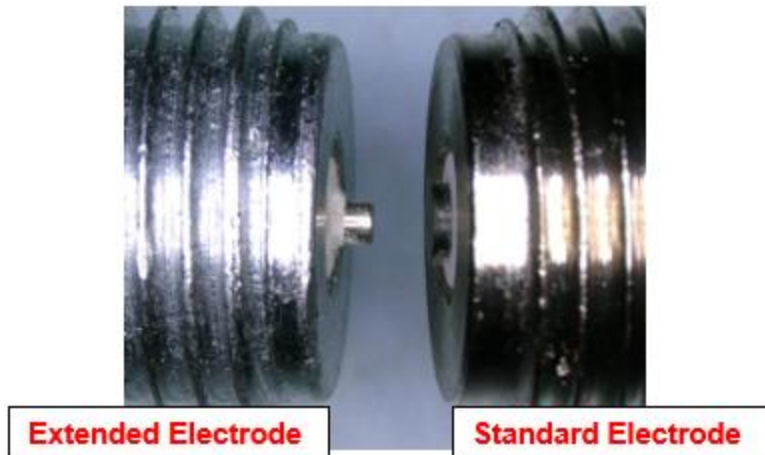


Figure 15.4. Spark Plug Center Electrode Comparison.

Engine Noise Investigation

An increase in engine noise was noted beginning at about 370 hours. Investigation (borescope inspection) led to the discovery of light scuffing on the cylinder sleeve and the observation of reduced oil flow from the crankcase oil jet (observed through the windowed crankcase). Engine oil viscosity was thought to be significantly increased; unfortunately an oil sample was not saved for analysis. An oil change improved observed oil flow with the decision made to closely monitor oil consumption, blowby, and cylinder leakage for the remainder of the endurance test. Oil consumption increased, from 0.005 oz/hr to a still low 0.010 oz/hr. Blowby, cylinder leakage, and power levels were maintained through the remaining test time.

At the end of the test period careful engine teardown and inspection was accomplished, with detailed photos and wear measurements summarized in a report published as part of the Q15 update (Q3 2019). Generally, the engine was in good condition with minimal wear in critical areas. Minimal wear components include: piston rings, valve guides and stems, camshaft lobes, chain and chain sprockets, and roller bearings. An area of concern includes the carbon deposits in the ring belt of the piston, including the ring grooves, ring lands, and piston rings themselves (See Figure 15.5 and 15.6). Additional deposits with increased run hours could lead to ring sticking, causing power loss and high oil consumption. This result is clouded by the lubrication issue with resulting bore/piston scuffing damage discovered at 370 hours that at least contributed to the depositing issue. The lubricating oil used was AMSOIL Synthetic Natural Gas Oil (SAE 40wt), a low ash (0.46%), high corrosion protection (5.7 TBN) formula. It was discovered that a high ash SAE 30 weight engine oil was used after the bore scuffing event, potentially contributing to ring belt deposits.



Figure 15.5. Piston Ring Belt Carbon.

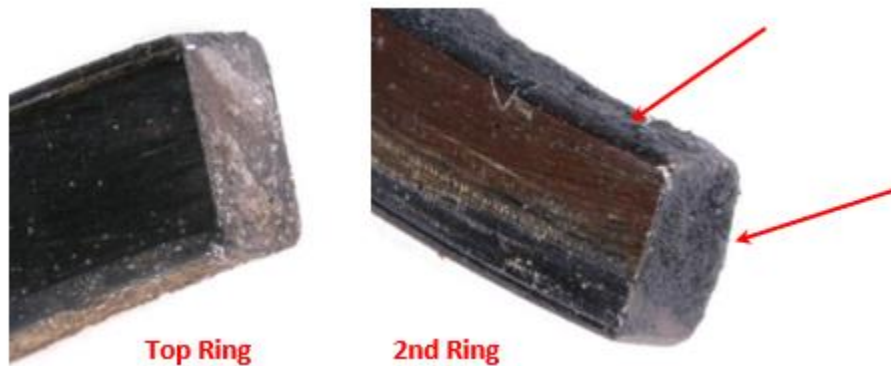


Figure 15.6. Top and 2nd Ring Gap & Backside Carbon Deposits.

Another area of concern is the condition of the valve seat inserts and valve sealing faces after 500+ hours. Intake and exhaust valve faces showed wear for the limited endurance test time. The exhaust valve seat insert exhibited a “peened” appearance due to significant carbon impact damage and/or adhesive wear. The exhaust valve face was highly oxidized and/or corroded. Intake and exhaust valve materials were both ANSI Silchrome-1 (9% Chrome, 3% Silicon) while the valve seat inserts were cast Triballoy 400 (high hot wear resistance due to high cobalt content). An improved hot oxidation/corrosion resistant material may be needed for the valve faces. Another issue with valve and seat contact was the wide and non-concentric face contact. This was the result of not having proper “topping” and “throat” cutters during prototyping. A third potential contributor to the poor wear performance of the valve seats and faces was the misalignment of the cam followers to the hydraulic lash adjusters. Unusual valve dynamics, especially during valve seating, could cause high seating velocity and force.

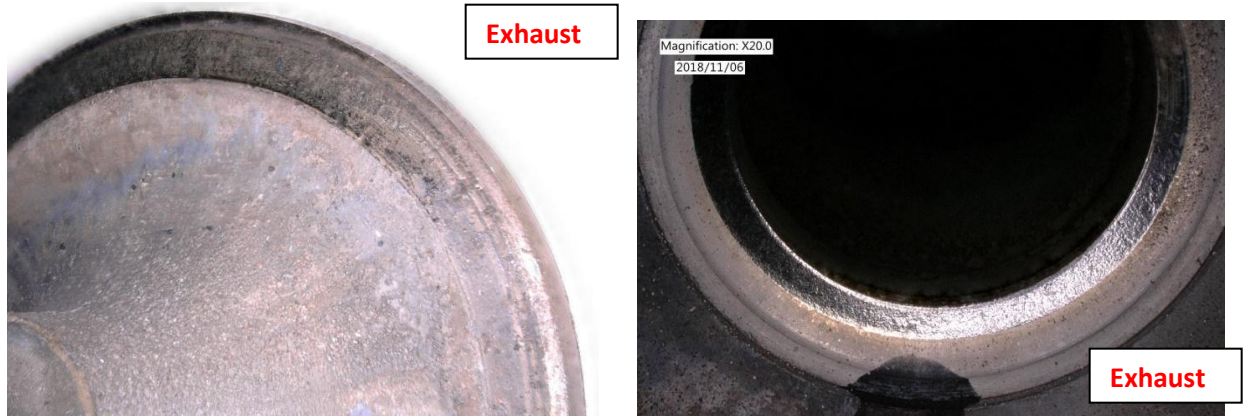


Figure 15.7. Exhaust Sealing Surfaces.



Figure 15.8. Intake Sealing Surfaces.

Recommendations for Continued Endurance Development

Several areas of corrective action are recommended before commissioning a second endurance test of the Phase 2 GENSETS engine. These actions are as follows:

1. Review cylinder/piston lubrication to assure proper piston skirt lubrication over an expected range of oil viscosities. Perform piston ring dynamic analysis to confirm proper inter-ring pressures over the engine cycle. Adjust piston ring gaps and/or ring land volumes to provide proper ring motion.
2. Confirm proper valve dynamics after correcting the misalignment of the cam follower sockets to the hydraulic lash adjusters. This has been done in the past using laser displacement measurement techniques.
3. Machine valve inserts with “topping” and “throat” cutters to provide a concentric seating surface of proper width.

4. Review valve face and seat conditions with Federal Mogul and Winsert, potentially procuring intake and exhaust valves with Triballoy 400 hard facing. The combination of a Triballoy hard faced valve and a cast Triballoy seat insert provided very good wear results in a previous gaseous fueled long life engine developed by Briggs & Stratton. At the time of prototype procurement for the GENSETS engine, an appropriate supplier could not be identified for Triballoy hard faced valves. If unavailable, consider other valve hard facings with improved hot oxidation/corrosion resistance.

Thermal Barrier Coatings (TBC's) should be applied to the combustion chamber components (head, valves, and piston crown) to evaluate durability of the latest coating formulation.

Conclusions

Based on initial development testing, the Briggs & Stratton dynamometer test engine was rebuilt with necessary Phase II upgraded components and endurance tested for 535 hours. End of test engine inspection and wear measurements were accomplished and summarized in a report included in the project Q15 update (Q3 2019). Engine wear on critical components was minimal, but piston ring belt deposits and valve seat insert and valve sealing faces exhibited need for improvement. Task 15 was successfully completed.

Task 16 - Development of Improved TBCs with Very Low Thermal or Heat Capacitance and Minimum Porosity

Summary

Figure 16.1 shows the timeline for Task 16: Development of Improved TBCs with Very Low Thermal Heat Capacitance and Minimum Porosity. Milestone 16.1 was completed at the end of Q4 of 2016, Milestone 16.2 was completed at the end of Q4 of 2017, and Milestone 16.3: Production Variation Testing of New Coatings was completed at the end of Q2 of 2019.

In Task 16, the main objective was to develop new coatings that possessed better thermal properties than the original two coatings tested in Task 1, while also improving durability. This was done through the use of in-house bench testing as well as 400 hour genset durability tests. Throughout this project, our measuring procedures were greatly improved upon and genset durability tests were completed on several improved iterations of coatings. The current TBCs have shown improvement in every key thermal property (thermal conductivity, density, specific heat capacity, bond strength) along with positive test results from the WERC and Briggs & Stratton.

A patent was submitted in Q2 2018 entitled "Thermal Barrier Coatings Containing Aluminosilicate Particles" and a SAE technical paper 2019-01-0228 entitled "Low Heat Capacitance Thermal Barrier Coatings for Internal Combustion Engines" on our coating development and WERC engine testing results was published and presented at the WCX SAE conference in 2019.



Figure 16.1. Timeline of Task 16, Improved TBCs.

Details

The purpose of newly developed TBCs within an engine is to increase the wall temperature swing between the intake and power strokes. Low thermal conductivity, low specific heat capacity, and low density are the main properties that allow a TBC to do this as they reject heat while taking more time to increase in temperature. Our target for this work was to achieve maximum heat rejection in the cylinder without retaining significant levels of heat energy stored in the cylinder wall. Engine simulation programs showed this will yield optimal thermal efficiency gains from the use of TBCs.

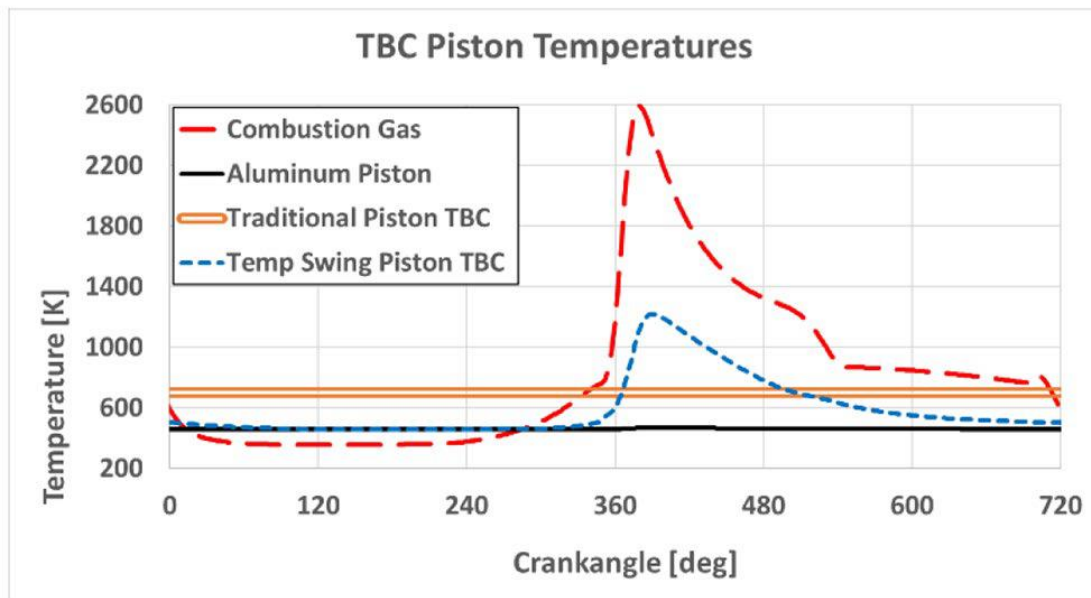


Figure 16.2. Wall temperature swings of different coatings within a Honda 125 PCX engine.

Thermal diffusivity is calculated using thermal conductivity, density, and specific heat,

Where: Thermal Diffusivity or $\alpha = \frac{k}{\rho c_p}$

k = thermal conductivity

ρ = density

c_p = specific heat

By developing coatings with low thermal conductivity, low density, and low specific heat, we looked to increase the temperature wall swing within the engine. Detailed below is an illustration and explanation on the procedure and how each of these thermal properties was measured throughout the project:

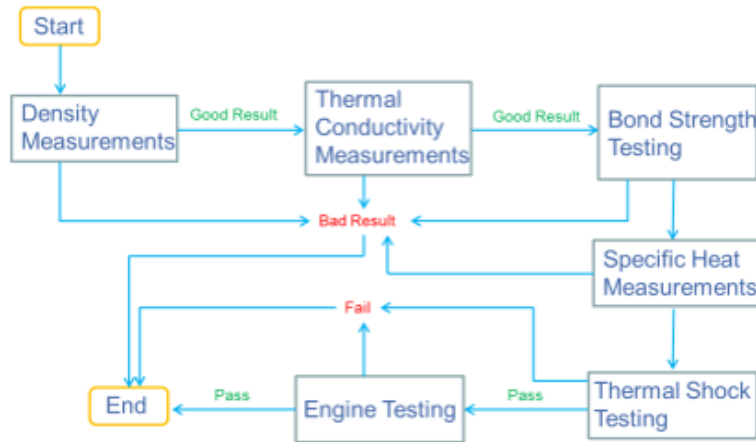


Figure 16.3. Flowchart illustrating the testing procedure for TBCs which are documented in SAE Paper 2019-01-0228.

In order to meet Milestone 16.1, at the end of Q4 in 2016, three improved coatings were selected as the most promising. These three coatings all possess improved thermal properties over the baseline coatings as well as demonstrated better durability through standard cycle oxy-acetylene thermal shock rig testing. Shown below is a comparison of the three coatings and their properties.

Material Characterization

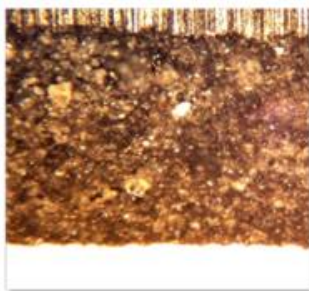
Bond Strength (mPA) 26.9
Thermal Cond. (W/m-K) 0.30
Density (g/cm³) 0.75

Material Characterization

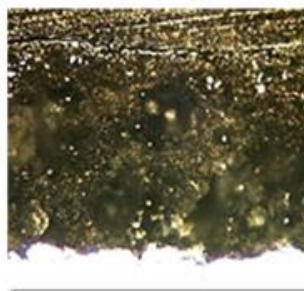
Bond Strength (mPA) 27.6
Thermal Cond. (W/m-K) 0.32
Density (g/cm³) 0.89

Material Characterization

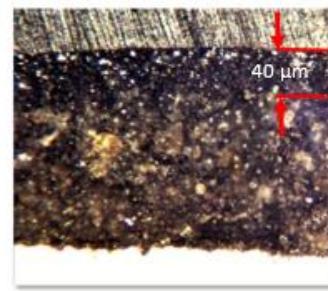
Bond Strength (mPA) 22.8
Thermal Cond. (W/m-K) 0.28
Density (g/cm³) 0.60



DZ Single TBC Layer



DZ Single Layer TBC with Sealer



Dual Layer Concept TBC

Figure 16.4. The three improved coatings selected in Q4 of 2016.

Milestone 16.2: Validation and Durability Testing of New Coatings Completed

Multiple iterations of coatings were subjected to standard oxy-acetylene thermal shock rig testing as well as 400 hour genset durability testing. The set chosen for the first coated 400 hour genset test were the DZ Coating and the Dual Layer Coating with the results as follows:

DZ Coating

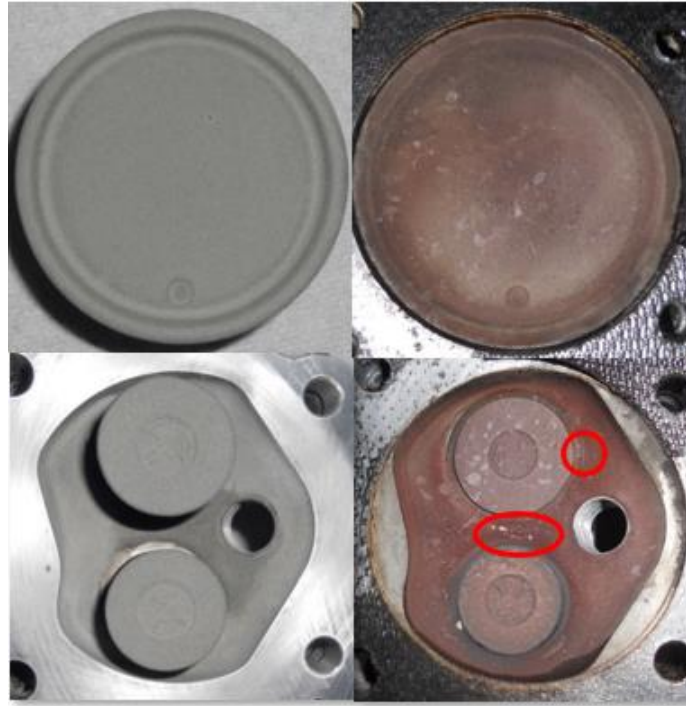


Figure 16.5. Piston and Head coated with the DZ Coating Before (Left) and After (Right) 400 hours of genset durability testing. The coating remained in fair condition due to a couple of small areas on the head that flaked off.

Dual Layer Coating

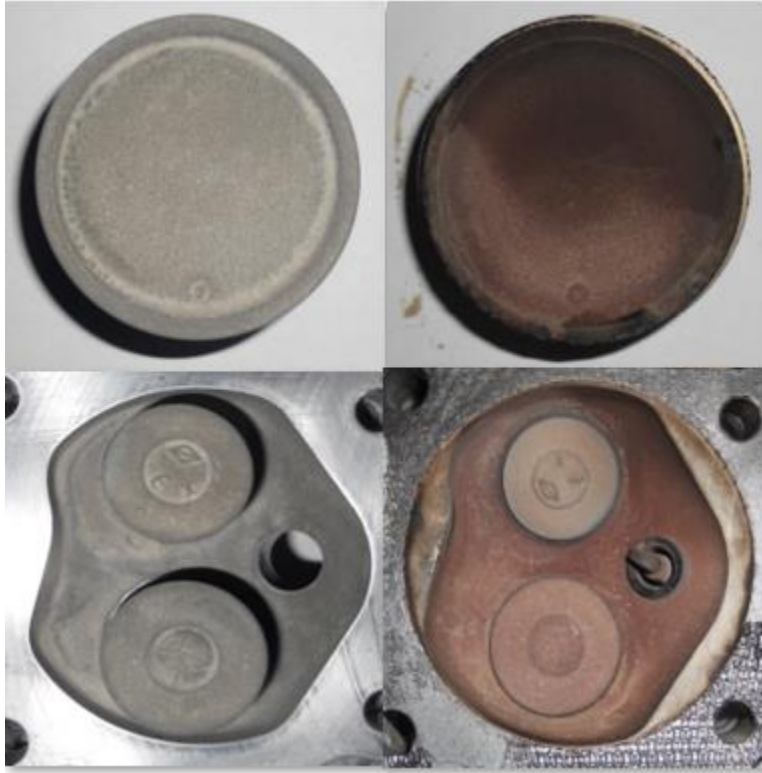


Figure 16.6. Piston and Head coated with the Dual Layer Coating Before (Left) and After (Right) 400 hours of genset durability testing. The coating remained in good condition on both the piston and head with no signs of flaking.

After the first coated genset iteration (near the end of Q3 in 2017) the team made the decision to use Inconel-751 valves rather than standard natural gas valves in the final engine design. Thermal shock rig testing had shown that the DZ, Dual Layer, RC, and PC coatings failed to create a strong bond to this substrate due to its high composition of nickel and chromium, rendering them unusable on the exhaust valve. However, it was found that applying a surface treatment to the surface of the exhaust valves would allow the RC coating to endure thermal shock testing which was later verified during a 400 hour genset test.

While the 400 hours of genset testing on the DZ and Dual Layer coatings was ongoing, two coatings, RC and PC, were developed with properties listed below. These two coatings were chosen for the second coated iteration test.

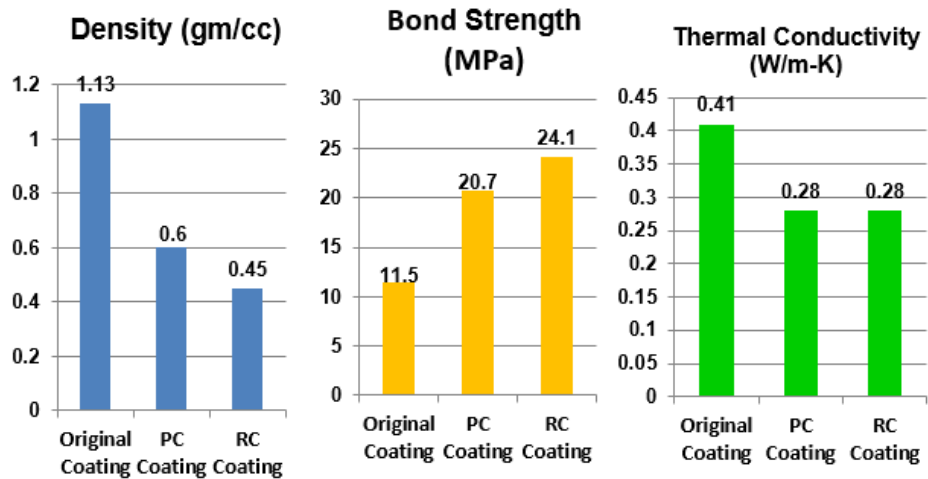


Figure 16.7. Three charts that compare properties of the RC and PC coatings to the original baseline coating.

RC Coating



Figure 16.8. Piston and Head coated with the RC Coating Before (Left) and After (Right) 400 hours of genset durability testing. Although the coating remained in good condition on the piston, a large amount of carbon deposits formed on its surface. The head remained in fair condition as there were two areas where minor flaking occurred.

PC Coating



Figure 16.9. Piston and Head coated with the PC Coating Before (Left) and After (Right) 400 hours of genset durability testing. The coating on both remained in good condition throughout testing without any notable deterioration.

Application techniques were investigated for the effect on thickness, bulk density, volume density, and porosity, and bond strength. Taguchi Method partial factorial testing on the RC coating was used for the first part. Below are the graphed results.

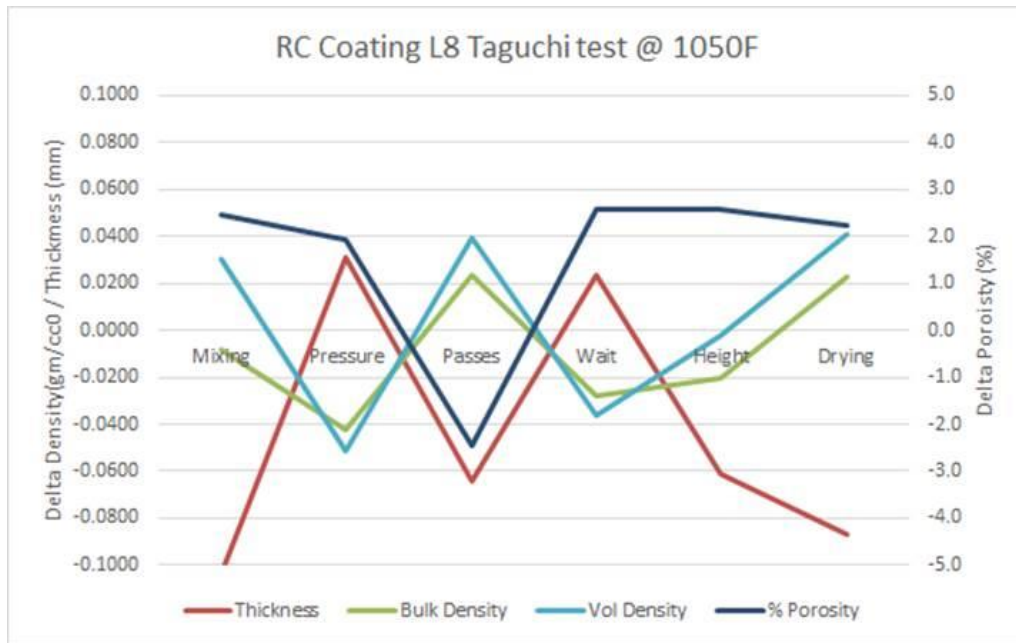


Figure 16.10. Taguchi Method results on how application variables affect thickness, bulk density, volume density, and porosity of RC coating after heating to 1050 °F.

As the results show, density has a strong inverse relationship to thickness. Whenever one of the application variables cause the thickness of the coating to increase or decrease, bulk density and volume density does the opposite. This is important as density is one of the main factors in the specific heat capacity equation.

Bond strength testing and density measurements on varying thicknesses of coating were used for the second part, determining how thickness affects bond strength and density. It was found that as coating thickness increases, both bond strength and density decrease. The four graphs below show the results for the RC and PC coatings from these tests.

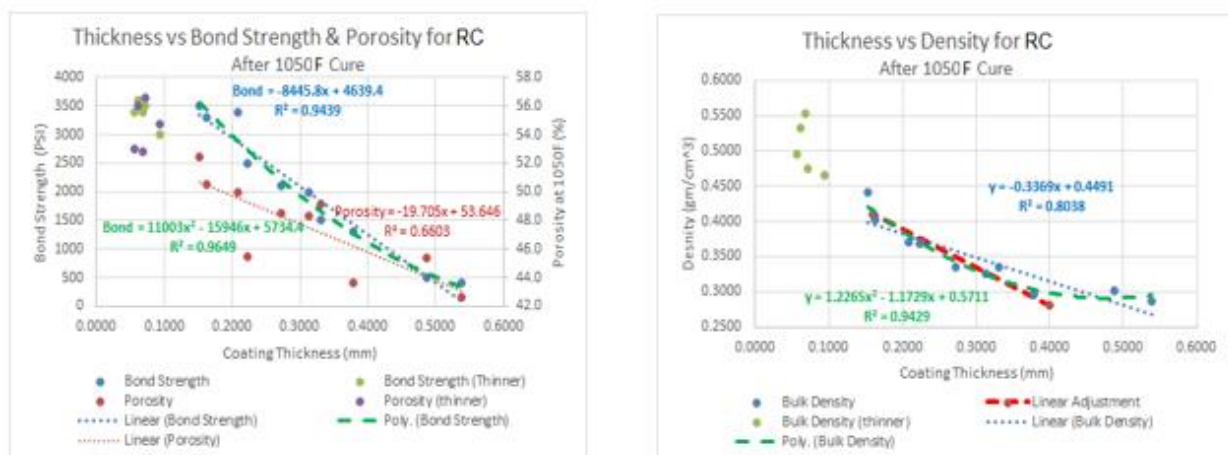


Figure 16.11. How thickness affects bond strength and density of the RC coating after curing at 1050 °F.

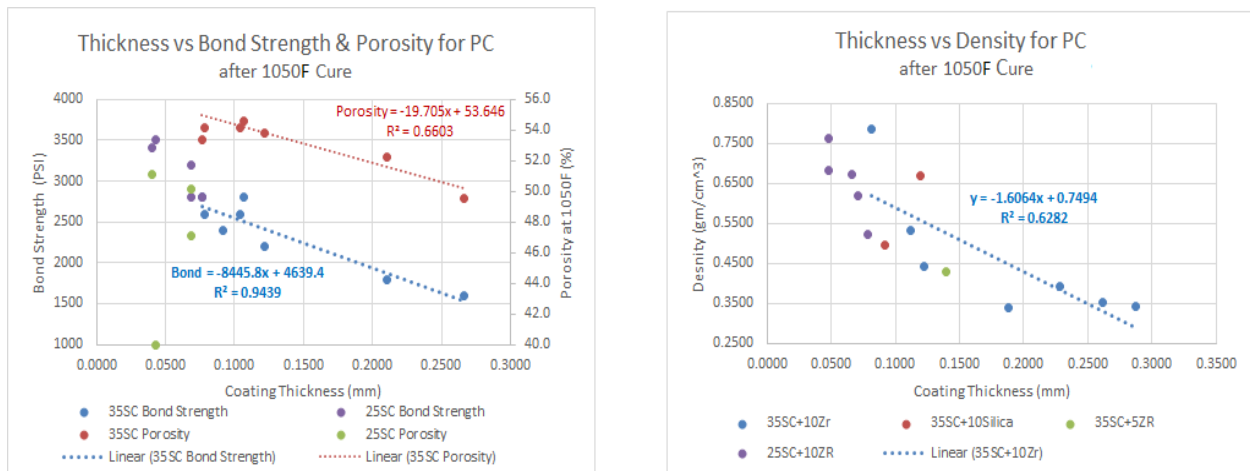


Figure 16.12. How thickness affects bond strength and density of the PC coating after curing at 1050 °F.

After the 400 hour genset durability test was completed for the RC and PC coatings, temperatures were measured at several different points on the engine

In all locations, the TBCs reduced the maximum temperatures and were especially effective on the exhaust valves as the temperature decreased by 64°C on exhaust valve 1 and 26°C on exhaust valve 2. The decrease in temperature on the pistons was also significantly reduced on the exhaust side by up to 24°C.

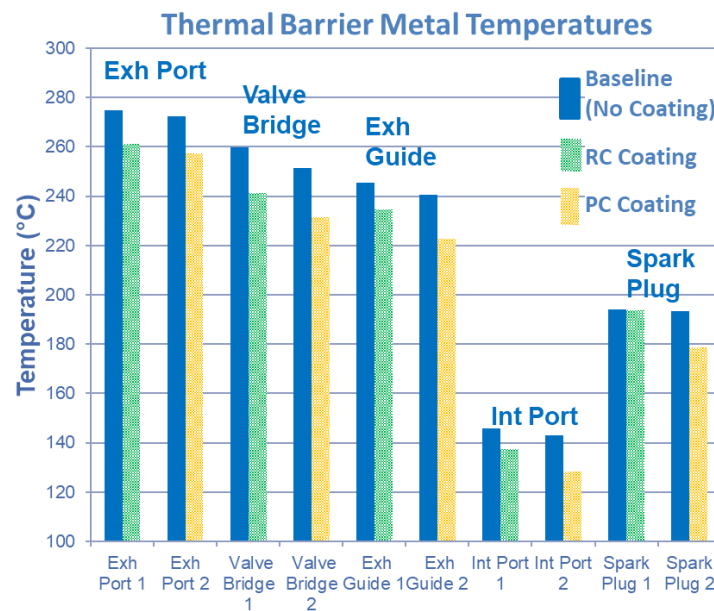


Figure 16.13. Comparison of baseline vs TBC coated cylinder head metal temperatures.

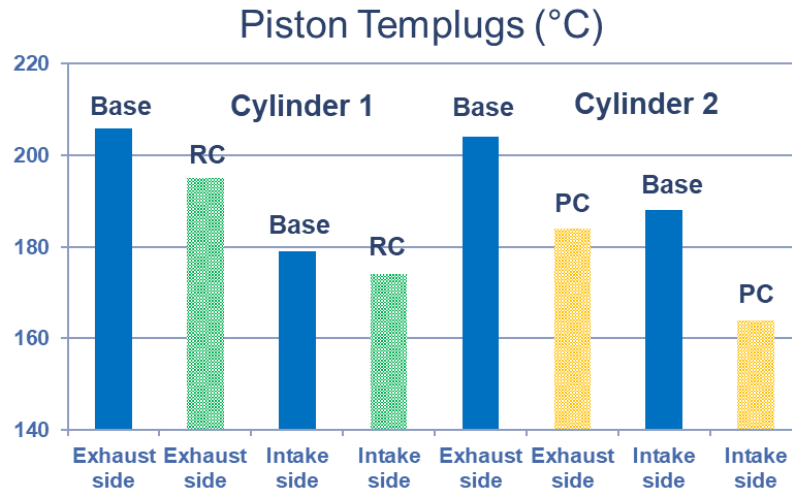


Figure 16.14. Comparison of baseline vs TBC coated templug piston metal temperatures.

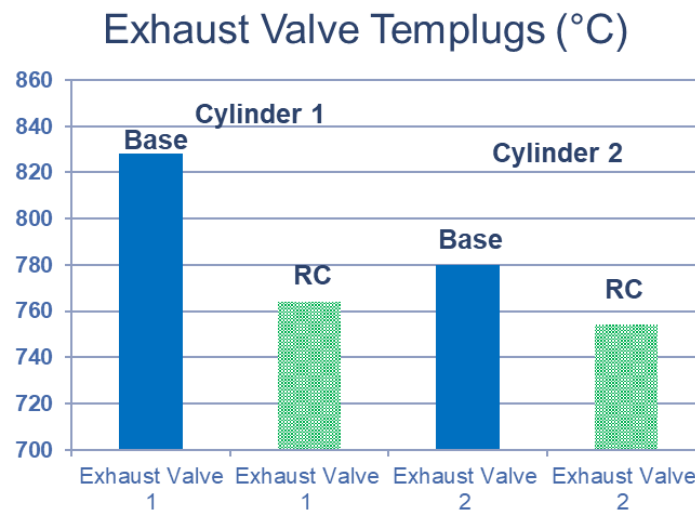


Figure 16.15. Comparison of baseline vs TBC coated templug exhaust valve metal temperatures.

During the UW-ERC testing, a failure of the RC piston coating was observed after 20-30 hours of knock testing. Shown in Figure 16.16 are before and after pictures after testing at UW-ERC.

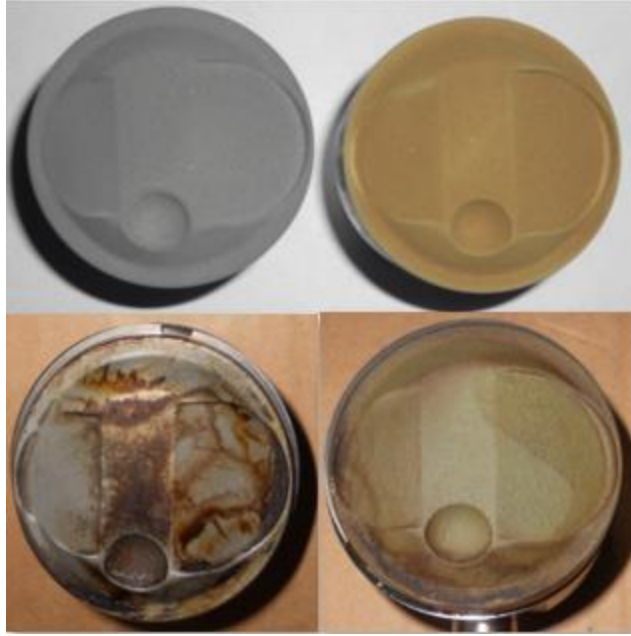


Figure 16.16. RC and PC coatings before (Top) and after UW-ERC testing. The RC coating failed after 20-30 hours of knock testing, while the PC piston remains in good condition.

After an intensive investigation, bond strength results showed that the root cause of this failure was determined to be the coating curing process. When the RC coating was applied to the piston, the coating was only cured for one hour at 350F to prevent weakening of the aluminum material. One-hour cure bond strength results below show that the RC coating fully cures when it reaches 850F and below 600F has no substrate adhesion strength. Because of these results, a longer duration cure cycle was investigated. With longer duration, increased bond strength at the lower temperatures was found, and therefore, the RC coating is now cured at 390F for 12 hours (390F is a recommended maximum piston aluminum temperature without change to its integrity); note the strength is still below the internal coating strength. The PC coating was also tested and found to fully cure at lower temperatures and it is believed to be the reason why it did not fail during ERC engine testing. Therefore, it was determined that the PC Coating is more effective for aluminum or low cure temperature substrates. Figure 16.17 shows how curing temperature and duration affects TBC bond strength.

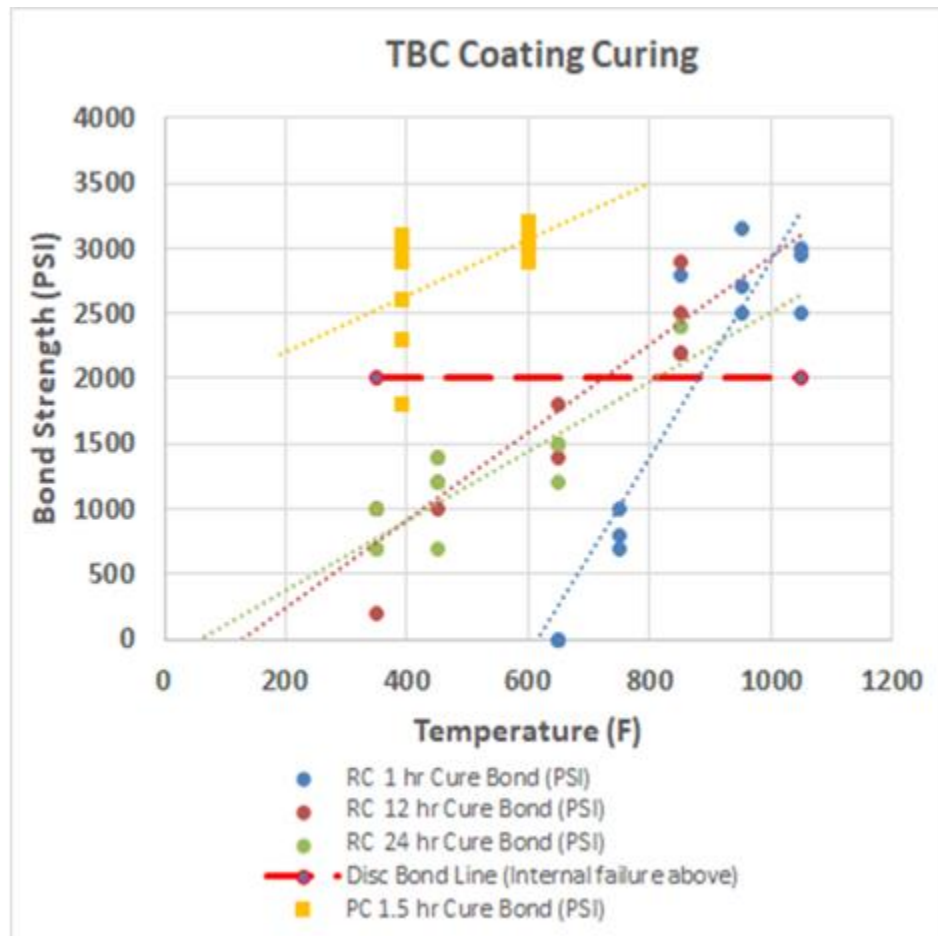


Figure 16.17. RC and PC coating bond strength results using various cure cycle temperatures and durations. Longer durations at lower temperatures significantly increase RC coating bond strength although it is still lower than the PC coating.

Because the PC Coating adheres to aluminum better but its thermal properties compared to the RC Coating are poorer, a new PC formula was developed that had a density and heat capacity very similar to the RC coating while maintaining the low curing temperature and durability of the PC Coating. The new PC Coating, referred to as LD PC Coating, performed just as well as the old PC Coating on durability bench tests and had similar thermal properties of the RC Coating. Figure 16.18 compares the LD PC Coating to that of the old PC Coating and the RC Coating.

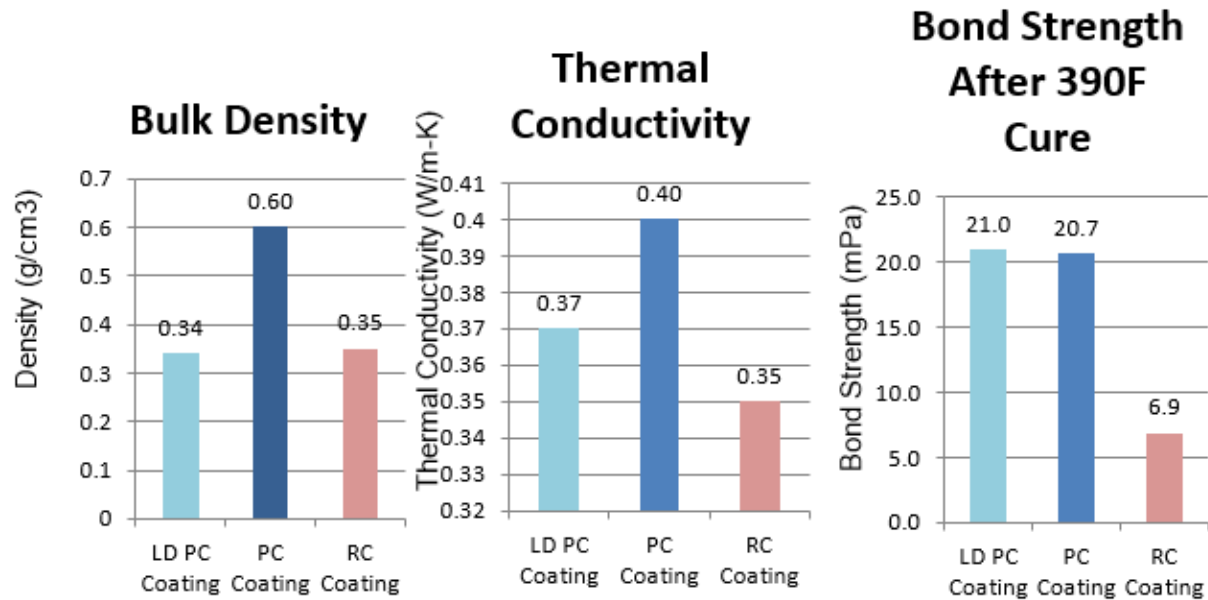


Figure 16.18. LD PC, PC, and RC coating bulk density, thermal conductivity, and bond strength results. The LD PC possesses better thermal properties than PC and very similar to that of the RC. It also has a much better bond strength at low temperature cure (390°F).

After the LD PC was optimized for thermal properties and bond strength, it underwent durability testing on the oxy-acetylene thermal shock rig. The LD PC coating passed both thermal property and bench durability testing so it was tested for 400 hours on the Briggs & Stratton 10kW 0.25L/cyl twin natural gas genset. The RC Coating was applied to one side of the twin cylinder engine and the LD PC Coating was applied to the other side with the results as follows:

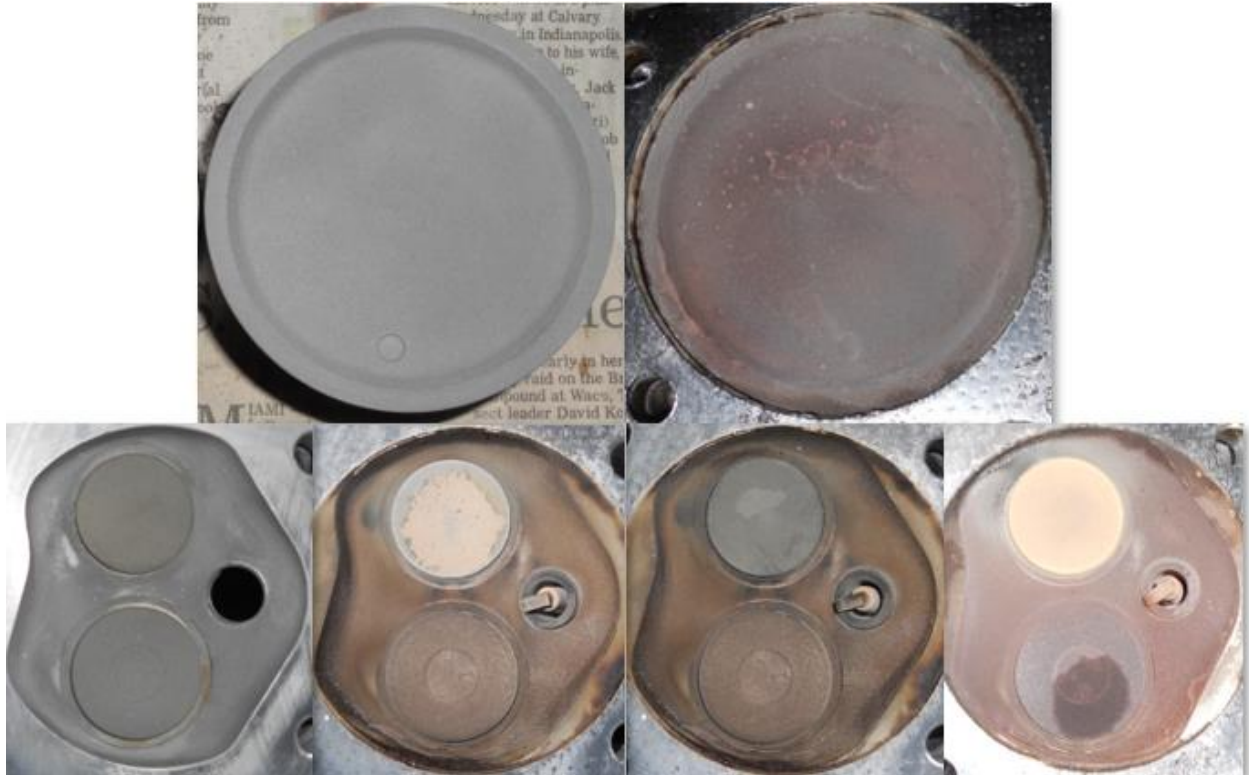


Figure 16.19. Piston: RC Coating before (Top Left) and after (Top Right) 400 hours of genset testing. Head: Before (Bottom Left), after 120 hours (with EX coating failure, Bottom Middle Left), after 120 (with new EX valve, Bottom Middle Right), and after 400 hours (Bottom Right). Coatings remain in good condition.

The coating on the piston held up well during testing and the coating on the cylinder head remained in good condition after 400 hours. However, the exhaust valve failed after 120 hours. The exhaust valve coating was then replaced with a thinner coating and made it through the remaining 280 hours without any failure and remains in good condition. The cylinder head and intake valve coatings do not appear to have experienced any significant degradation during testing and remain in good condition.

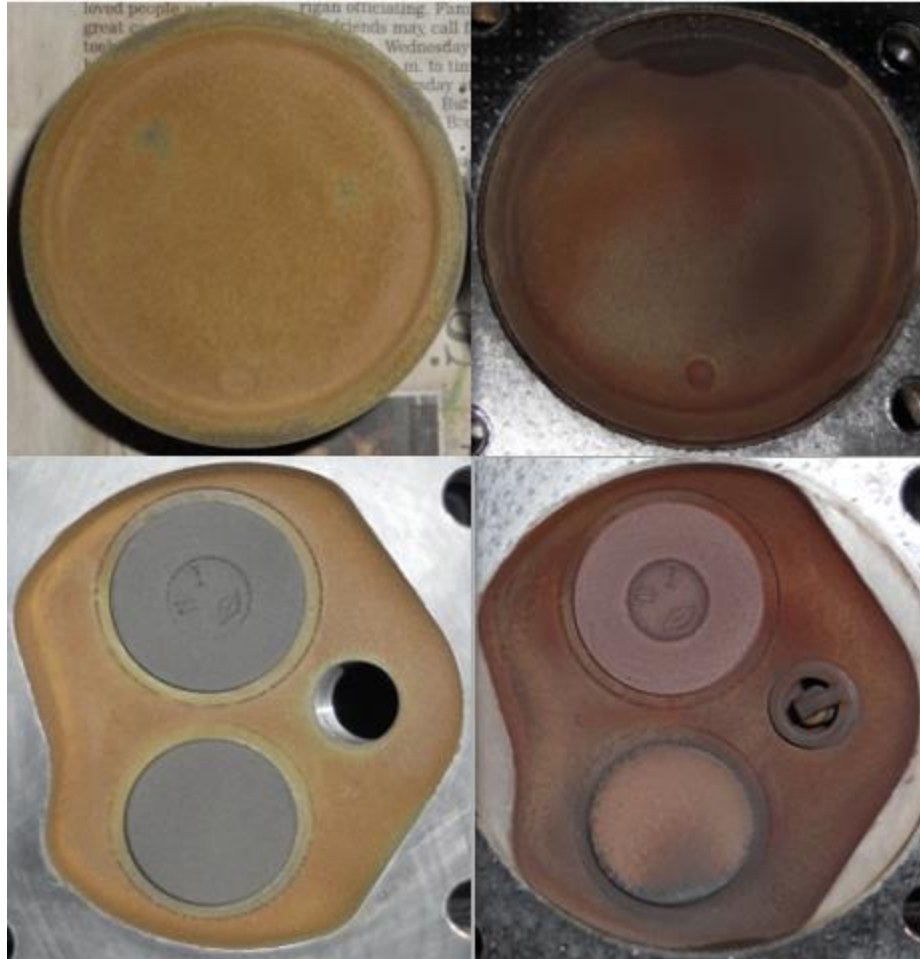


Figure 16.20. Piston: LD PC Coating before (Top Left) and after (Top Right) 400 hours of genset testing. Head: LD PC Coating on head, RC Coating on valves before (Bottom Left) and after (Bottom Right) 400 hours of genset testing. Coating on piston and head remains in good condition with minor flaking around EX valve.

In Q1 of 2019, Briggs & Stratton conducted performance testing of the TBCs on an 810 V-Twin gasoline engine. Results were mixed as the coating improved knock resistance dramatically but also increased unburned hydrocarbons substantially. This increase in unburned hydrocarbons wasn't seen during testing with a gaseous fuel, so it was theorized that the porosity of the coating is absorbing some of the liquid fuel and preventing it from combusting, causing the increase in unburned hydrocarbons. For this reason, more emphasis was placed on developing a less porous coating either through the use of a sealer or through the formulation of the coating. Therefore, a new smoother and less porous coating, SPC, was developed in Q2 of 2019. This coating is similar to the PC and LD PC coatings in that it uses the same binder, but it also exhibits better thermal properties.

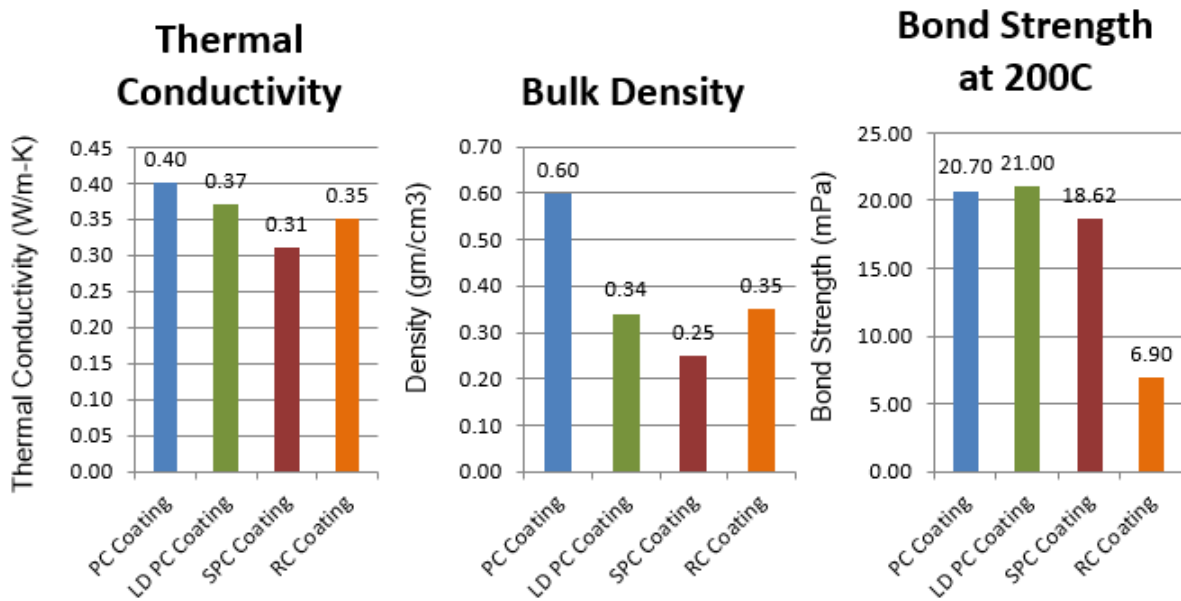


Figure 16.21. Thermal properties of new SPC Coating as compared to previous PC, RC, and LD PC coatings.

PSZ samples were also measured in Q2 of 2019 This was done in order to assess the accuracy of our density measurement procedure as well as the procedure for thermal conductivity and specific heat capacity by comparing results to that our developed TBCs. Shown below in Figures 16.22 and 16.23 are the results from the bench testing.

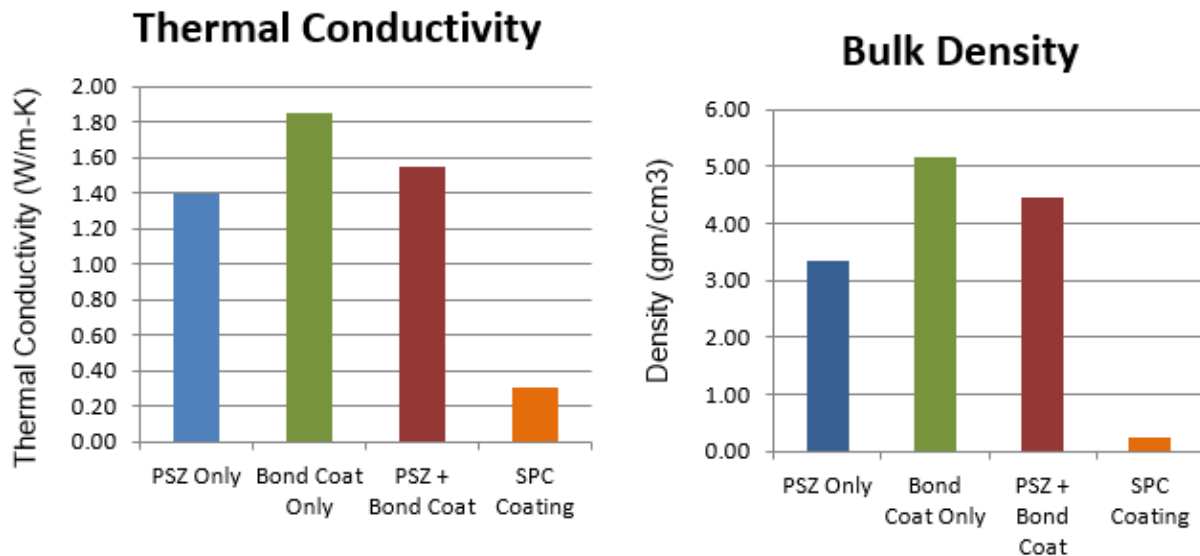


Figure 16.22. Thermal conductivity and density measurements of PSZ as compared the new SPC Coating.

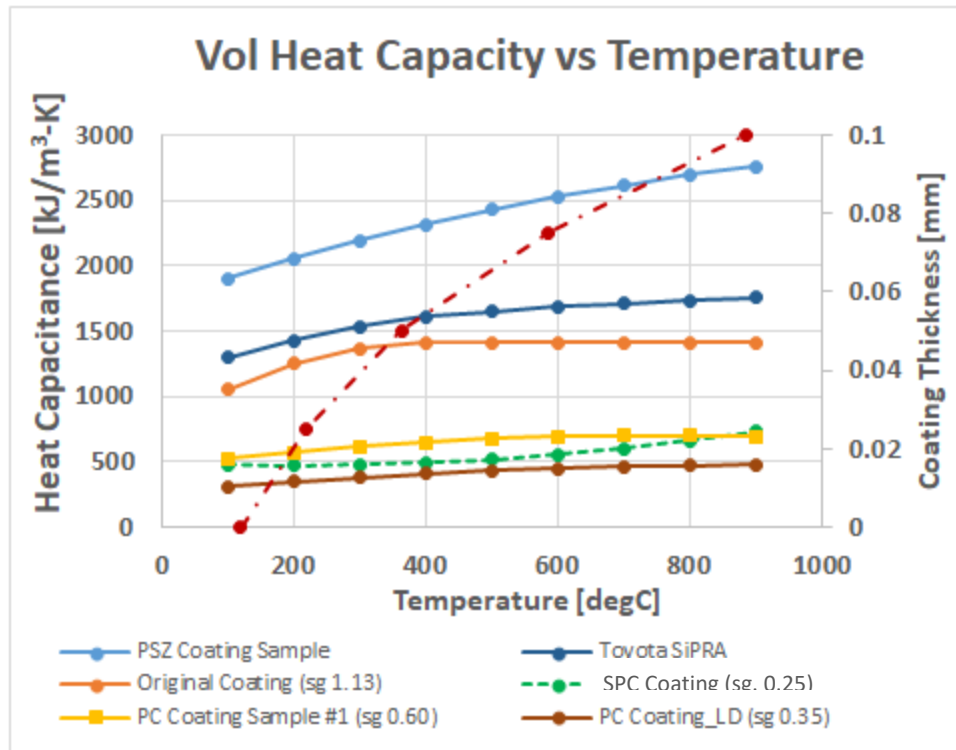


Figure 16.23. Heat capacitance of Adiabatics' original, PC, LD PC, and SPC coatings as compared to PSZ and Toyota's SiRPA Coating.

The third and last deliverable of Task 16 was also completed in Q2: Production Variation of New Coatings. The original milestone stated that 1,000 thermal cycle tests would be run on the coatings; however, the thermal shock test was re-designed earlier in the project so that the torch was switched from natural gas to oxy-acetylene. It was found that a 50 cycle test on the oxy-acetylene rig was comparable to 1,000 cycles on the natural gas rig. Therefore, the coatings were run for 50 cycles at 14 seconds rather than 1,000 cycles at 24 seconds. The RC Coating, which was developed for exhaust valves, was run on a high nickel/chromium exhaust valve, while the PC Coating was run on 6061 aluminum samples which is the same substrate as the B&S pistons.

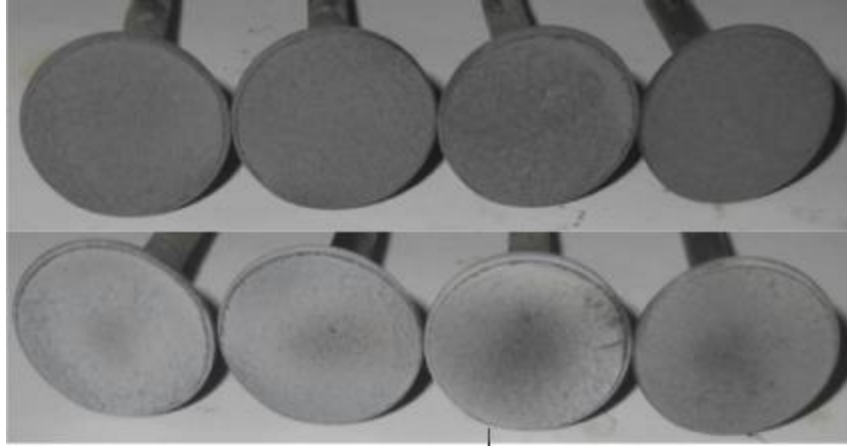


Figure 16.24. RC coatings on EX valves before (Top) and after (Bottom) 50 cycles of thermal shock rig testing. Coating thicknesses from left: 0.002", 0.003", 0.004", and 0.008". All valves passed.

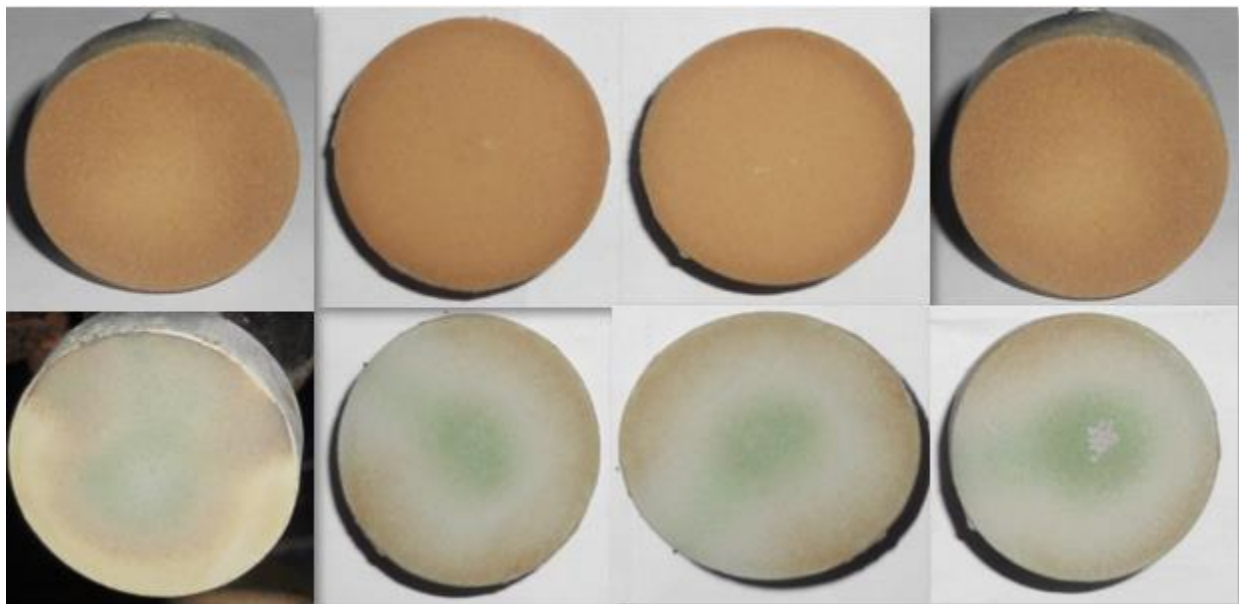


Figure 16.25. PC coatings on 6061 aluminum samples before (Top) and after (Bottom) 50 cycles of thermal shock rig testing. Coating thicknesses from left: 0.002", 0.003", 0.004", and 0.008". All coatings except for the 0.008" passed.

Conclusions

Milestone 16.1: Demonstration of at Least Three Improved TBCs was completed on time at the end of Q4 in 2016 when three improved TBCs were selected as they showed improvements in thermal properties (density, thermal conductivity, and specific heat) and durability (bond strength and thermal shock) over that of the baseline TBC. The first coating was a heat cured silica alumina micronized boron carbide sprayed coating with a high temperature (1480°C) polymer ceramic resin. The second coating

was a similar heat cured silica alumina micronized boron carbide sprayed coating with a high temperature (1480°C) polymer ceramic resin. However, it was sealed in order to decrease porosity. The third coating was a dual layer coating with the bottom layer being a sprayed ambient cured silica alumina micronized boron carbide coating and the top layer being a sprayed ambient cure low specific heat silica carbon coating.

Milestone 16.2: Validation and Durability Testing of Improved TBCs was completed on time at the end of Q4 of 2017 after multiple iterations of coatings were subjected to standard oxy-acetylene thermal shock rig testing as well as 400 hour genset durability testing. The set chosen for the first coated 400 hour genset test was the DZ Coating and the Dual Layer Coating. However, the RC, PC, and LD PC coatings were later subjected to thermal property and durability testing of which all of them passed and outperformed the baseline TBC.

Milestone 16.3: Production Variation of New Coatings was initially expected to be completed at the end of Q4 of 2018, but was completed at the end of Q2 of 2019 after a project extension was granted. For this milestone, the RC and PC coatings were applied at different thicknesses and subjected to thermal shock rig testing. As the thermal shock rig was re-designed earlier in the project, the coatings were run for 50 cycles at 14 seconds rather than 1,000 cycles at 24 seconds. The RC Coating, which was developed for exhaust valves, was run on a high nickel/chromium exhaust valve, while the PC Coating was run on 6061 aluminum samples which is the same substrate as the B&S pistons. All of the coatings passed except for the PC Coating applied at 0.008”.

Task 17 - Computational Optimization of Low Cost Boosting System

Summary

Task 17 was focused on optimizing boosting systems that is able to achieve 38.5% engine BTE. Four options are presented that include 2 PWS based systems and 2 intake/exhaust tuned systems. All options showed potential to achieve target BTE. The PWS systems showed optimum BTE in the 39-40% range while the tuned intake/exhaust systems showed optimum BTE in the 38-39% range. All of the systems rely on optimized combustion systems and TBCs. This work was completed by end of project Q9, satisfying Milestone M17.1, “Design of the low cost boosting system meeting the targets specified in Task 3 is complete. Expected minimum pressure ratio of 1.2 at the engine operating condition specified / determined in M3.1.”.

Details

Task 17 focused on computationally optimizing a low cost boosting system. Preliminary analysis has shown that a boosting system is an important enabler for achieving the mechanical efficiency and system size targets. The validated GT-Power model was used to optimize a low cost boosting system that is able to achieve the boost required to achieve 40% electrical conversion efficiency. Four options have been studied including 2 PWS systems, an induction/exhaust tuning system and an induction system with resonator.

Two PWS systems are shown in Figures 17.1 and 17.2. The first system has external cooled EGR. The second system has no external EGR circuit, but rather recirculates exhaust gas through the PWS device.

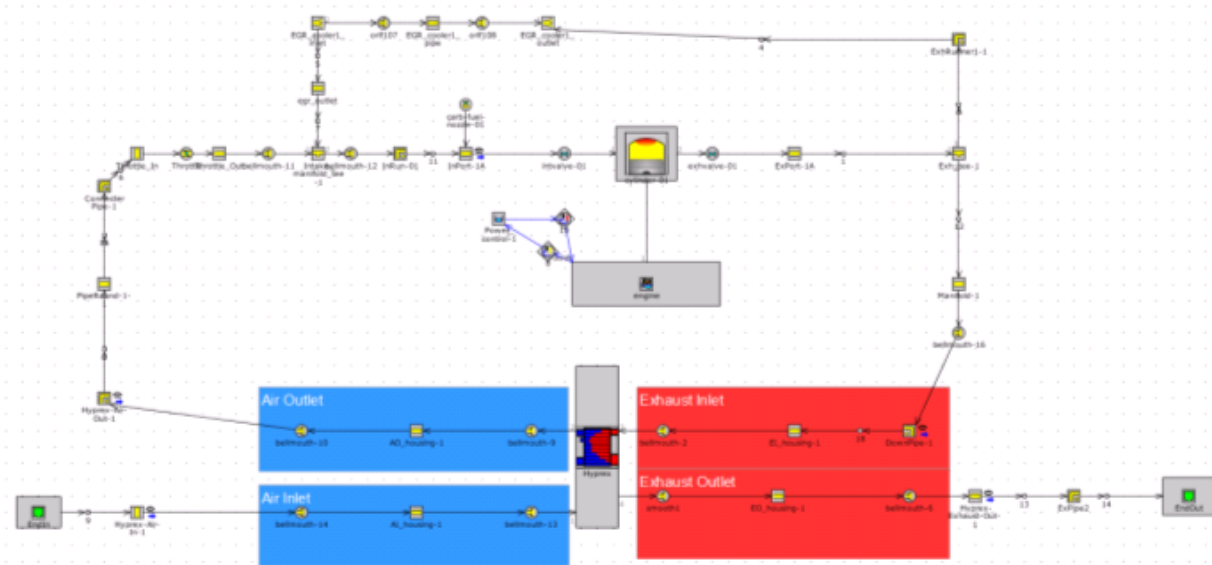


Figure 17.1. PWS system with external EGR.

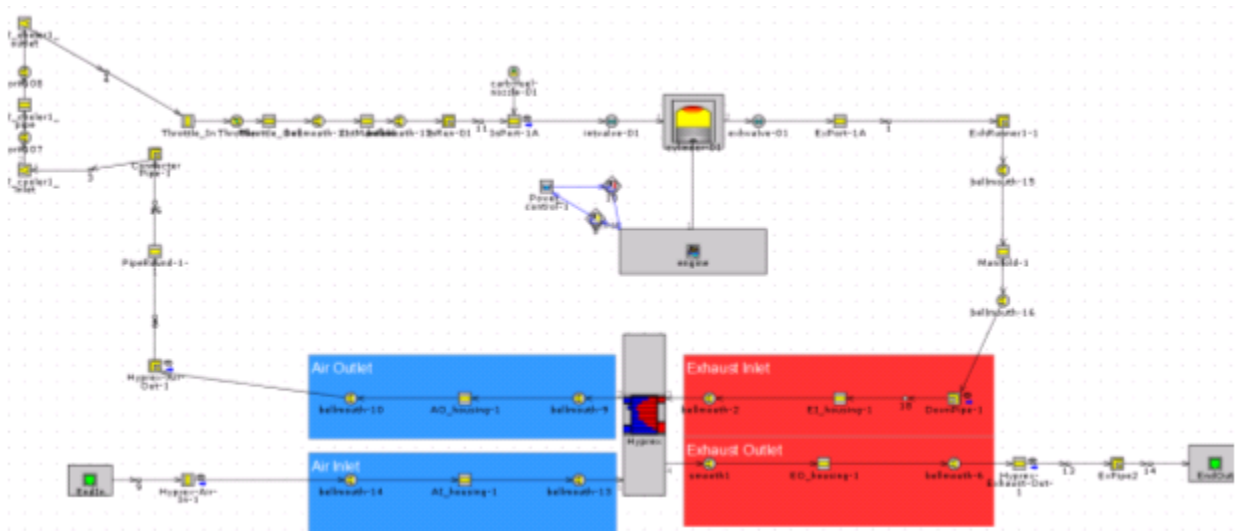


Figure 17.2. System with exhaust gas recirculation through PWS.

DoE study of key PWS design parameters including rotor speed, length, port phasings and port open times was performed.

Best case achieved 39.4% BTE with 1.26bar boost, 24.4% EGR and TBC. DoE study repeated on system with external cooled EGR circuit. Similar best case results found: 40.4% BTE with 1.25bar boost, 30.6% EGR and TBC.

Intake and exhaust tuning was investigated. Several DoE's of varying levels of complexity were performed including:

1. Fixed diameter, intake and exhaust runner length.
2. Intake resonator (neck dia, length and volume), intake runner length.
3. Intake and exhaust dia and length with current cam.
4. Intake and exhaust dia and length, intake and exhaust cam phasing and duration.

Each DoE included EGR rate as a variable as well as TBCs. The EGR outlet temperature was fixed at 90°C. Additionally, each DoE was repeated with different levels of combustion efficiency. The baseline had combustion efficiency of 95%, which is comparable to efficiency demonstrated in test. The repeat DoE was run with 98% combustion efficiency, which although not yet demonstrated on this engine development, is typical of larger spark ignited engines. For all the studies conducted, the target engine brake power was maintained by adjusting engine speed, limited to 5000rev/min. Finally, the intake and exhaust were divided into primary and secondary sections. The distinction is that the primary runners are the sections from the port to EGR inlet/outlet. The intake secondary runner is the section from the airbox to the EGR inlet and the exhaust secondary runner is from the EGR takeoff to the muffler.

The results show BTE improves with mid-to-short exhaust and mid length intake. Best BTE predicted of 38.13%. The improved combustion efficiency runs showed similar trends with a best BTE of 39%. This indicates that simple runner length optimization coupled with optimized combustion could achieve overall program goal of 38.5% BTE.

For DoE 2, a Helmholtz style resonator was included in the intake system. A Helmholtz resonator is a device that can be used to amplify or negate pressure pulses in an internal combustion engine intake system. It has an open end attached at a favorable point in the intake ducting, a neck attached to a closed volume. A simple sketch of the style of resonator used in this study can be seen in Figure 17.3.

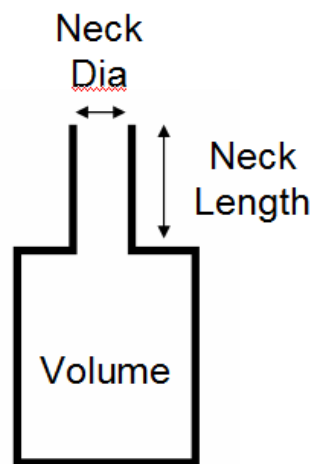


Figure 17.3. Helmholtz resonator with variables considered in DoE.

In addition to previously mentioned variables, DoE 2 included intake primary and secondary lengths and EGR rate. In addition to repeating DoE 2 with two levels of combustion efficiency, the study was repeated with resonator placement, either in the primary or secondary duct. This effectively varies placement of the resonator in the intake system.

Results can be seen in Figure 17.4.

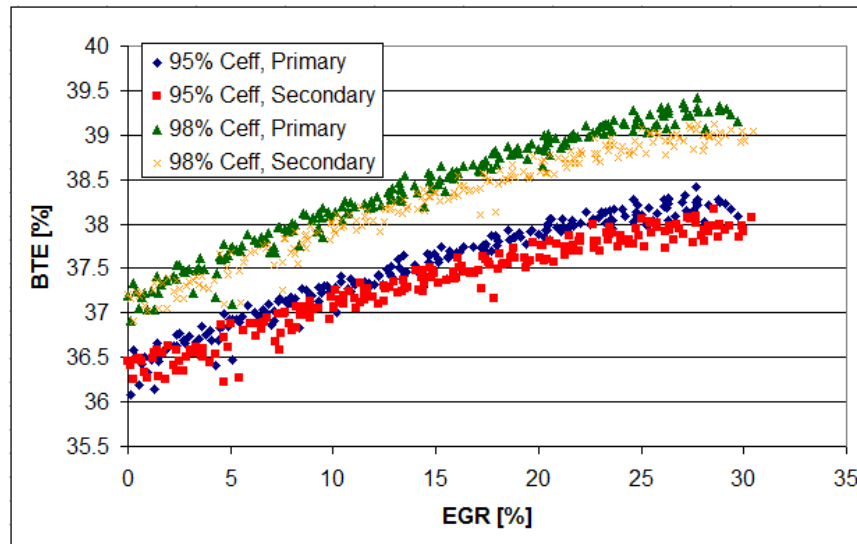


Figure 17.4. Helmholtz resonator DoE results, BTE as a function of EGR.

No variables other than EGR showed as clear of a trend with respect to BTE. As can be seen, improvement in combustion efficiency resulted in roughly 1% improvement in BTE. Placement of the resonator on the primary pipe (closer to the engine, after EGR inlet) had better results compared to placement closer to the air system inlet. Considering a combustion stability limit of 20-25% EGR, it appears that program target BTE of 38.5% is achievable with base or slightly improved combustion efficiency.

For DoE 3, the intake and exhaust primary and secondary ducts were varied independently. This included diameter and length. Additionally, EGR was varied. Duct diameters were varied from 1" to 2" and the duct lengths were varied from 50mm to 1000mm. Similar to previous studies, EGR was varied from 0-30%.

Results are shown in Figure 17.5 where, similarly to DoE 2, they are presented as BTE as a function of EGR.

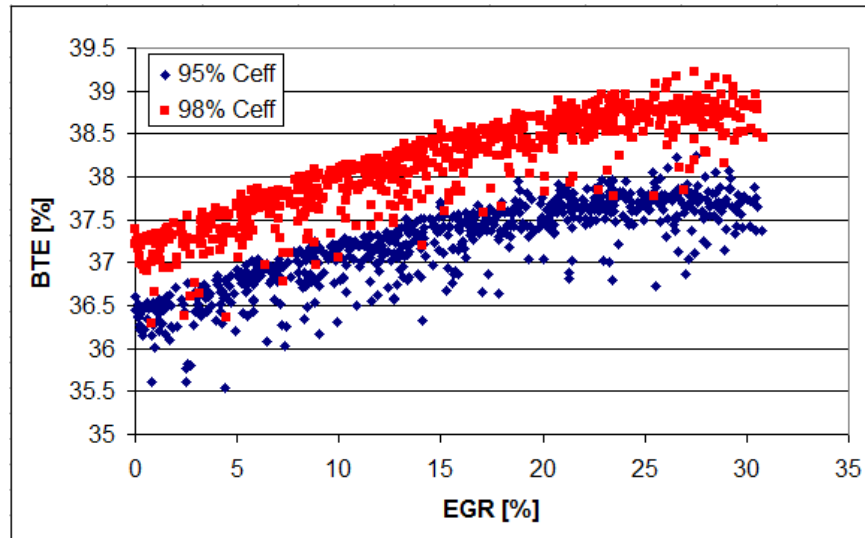


Figure 17.5. Intake/Exhaust with current cam DoE results, BTE as a function of EGR.

The results show, similar to previous studies, that improved combustion results in ~1% BTE increase. Also, similar to the resonator study, target BTE of 38.5% appears achievable at 20-25% EGR with baseline to slightly improved combustion.

The final study, DoE 4, included DoE 3 variables for intake and exhaust ducting as well as intake and exhaust cam duration and phasing. BTE as a function of EGR results are shown in Figure 17.6.

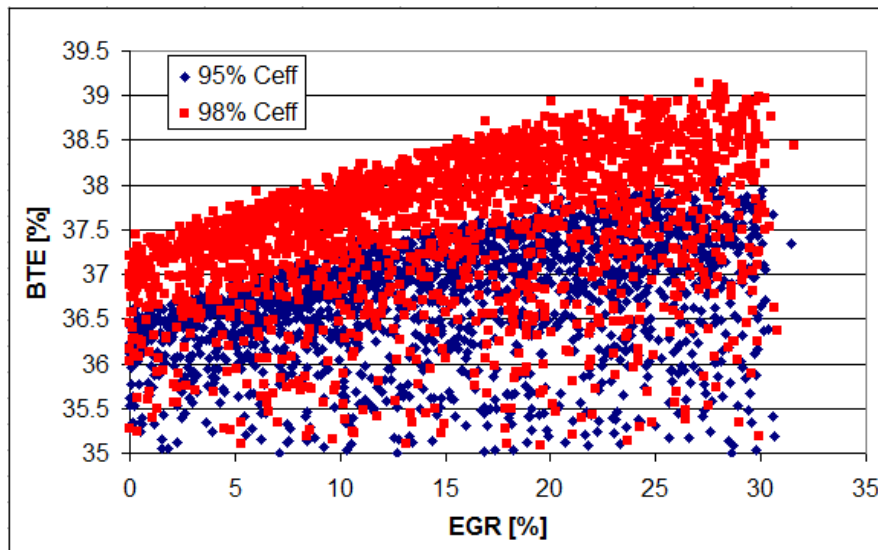


Figure 17.6. Intake/Exhaust ducting and cam DoE results, BTE as a function of EGR.

The results are very similar to DoE 3 where improved combustion results in ~1%BTE increase and target BTE of 38.5% seems achievable at 20-25% EGR with base to slightly improved combustion. The results

from DoE 4 are very similar to DoE 3 indicating that the cam developed as part of the original optimization is still at or near the optimum.

The boosting system optimization study has shown 4 feasible options for achieving target BTE, thereby fulfilling Milestone M17.1, "Design of the low cost boosting system meeting the targets specified in Task 3 is complete. Expected minimum pressure ratio of 1.2 at the engine operating condition specified / determined in M3.1." Those include PWS with no external EGR, PWS with cooled EGR, optimized intake and exhaust runners with cooled EGR, with or without intake resonator. All of the studies have shown that efficient combustion will be a key to achieving the program BTE goal. The PWS systems have shown more promise to achieve high BTE although at a complexity cost. To address this, a drum charger system has been developed. This is expected to offer reduced capability of the PWS systems but at much less complexity.

Task 18 – Design and Construction of Phase II Prototype Engines

Summary

The objective of this task is to perform layout and 3-D modeling of refined engine design informed by Phase I concept prototypes experimental and simulation results. Drawings will be created and a sourcing plan for Phase II prototypes will be accomplished. Sourced or manufacture components necessary to build updated prototypes and additional parts to support development testing will be provided.

Details

An engine was provided to UW-ERC with Phase II upgraded components and the Briggs & Stratton endurance engine was rebuilt with these updated components (see Task 15). Additional efforts under this task focused on components that could increase the thermal efficiency of the engine through combustion improvements and mechanical friction reduction. Components included parts with thermal barrier coating, a DLC coated cylinder sleeve, a reduced tension oil ring assembly, and several iterations of pre-chamber spark plugs.

UW-ERC evaluation of the Phase II engine has shown a significant reduction in friction over the Phase I engine with a corresponding improvement in engine efficiency. See Task 19. Several component modifications are thought to contribute to this reduce friction including the improved piston profile, a more optimized crankshaft lubrication supply (reduced oil windage), and the improved flywheel/coupling alignment.

For further friction reduction, a stainless steel cylinder sleeve coated with DLC (Diamond Like Carbon) by Duralar was installed in the UW-ERC engine along with a new ring set that included an oil ring assembly with 20% lower tangential tension. After a 2 hour break-in, a bore scope was used to evaluate the coating. Surprisingly, the DLC was found removed from the upper section of piston ring travel with evidence of piston skirt scuffing. Teardown yielded the damage shown in Figure 18.1. The cause of the coating delamination was thought to be due using an exterior generated plasma instead of an Inside diameter generated beam. The engine was rebuilt with a cast iron sleeve, a 20% lower tension oil ring

assembly, and TBC treated components. The test schedule did not allow the evaluation of a low friction Iron-Titanium oxide coated sleeve (provided by Adiabatics, Inc).

The UW-ERC Phase II engine was fitted with thermal barrier coatings, as developed by Adiabatics, Inc. The cylinder head, intake valve, and piston were coated using improved thermal property, low density “PC” material. The exhaust valve was coated with the latest “RC” coating. Cylinder head intake and exhaust ports were not treated. Testing results are reported under Task 19.



Figure 18.1. DLC Coated Sleeve Failure and Piston Damage.

UW-ERC analysis has shown that combustion speed improvement should improve engine efficiency. An in-cylinder turbulence increase is one method to improve combustion speed. Methods to improve in-cylinder turbulence with intake port or combustion chamber modifications were evaluated using WERC CFD modeling capabilities. As reported in Task 14, modifications did not show a significant change in turbulence; no hardware was fabricated for engine testing.

Another method to improve combustion speed is with a passive pre-chamber spark plug. Combustion is initiated in the plug pre-chamber creating jets that pass through orifices into the main chamber. These jets create rapid combustion in the main chamber. An initial design, shown in Figure 18.7 and 18.8, included a pre-chamber volume of 200mm³ with two (2) 0.5mm diameter orifices. After initial unsuccessful operation at UW-ERC, the orifice holes were increased to 0.8mm diameter with impressive

combustion speed improvement. Still, engine efficiency was not improved, thought due to increased heat transfer.

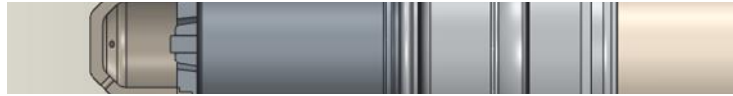


Figure 18.2. Pre-Chamber Spark Plug (200mm³ volume).

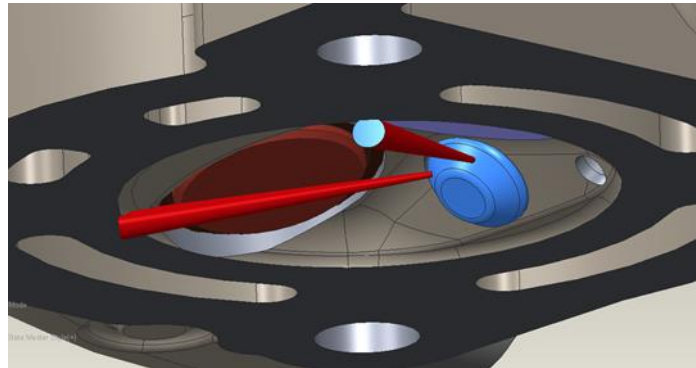


Figure 18.3. Simulated Orifice Jets – PC Spark Plug.

A WERC CFD model was developed to analyze pre-chamber performance and evaluate volume and orifice diameter(s) into the main chamber (see Task 14). Modeling results confirmed the need for the larger orifice holes in the original design and also showed that charge motion into the pre-chamber (during the compression stroke) was directed across the spark gap, potentially affecting the early flame kernel. It was also thought that the pre-chamber volume at 0.2cc (5% of combustion chamber clearance volume) with 2 orifice holes might be causing excessively fast burn rates and high heat transfer. Single orifice hole designs were configured with chamber volumes of 200mm³, 120mm³, and 80mm³ at inlet angles away from the spark gap. A tangential orifice hole configuration was also designed to create chamber swirl and prevent flame kernel interaction. A matrix of prototyped pre-chambers, with volume to orifice size ratios, is provided in Table 18.1. Sample geometries are shown Figure 18.4 and 18.5. Test results continued to show combustion speed improvements with most pre-chamber configurations, but without thermal efficiency gain. See Task 19 for performance results with pre-chamber spark plugs. Further optimization of passive pre-chamber spark plug designs would require significant CFD simulation, outside the scope of the present effort.

Table 18.1. Pre-Chamber Volume to Orifice Area Ratio.

	Volume/Orifice Area 100-160 Range	Volume/Orifice Area 200-250 Range	Volume/Orifice Area 500+ Range
Pre-Chamber Volume			
200 mm ³ (~5% of TDC volume) Two hole (initial prototypes)	114 (1.0mm holes)	200 (0.8mm holes)	510 (0.5mm holes)
120 mm ³ (~3% of TDC volume) One hole (modified angle orifice)		239 (0.8mm hole)	
120 mm ³ (~3% of TDC volume) One hole (tangential orifice)		239 (0.8mm hole)	
80 mm ³ (~2% of TDC volume) One hole (tangential orifice)	159 (0.8mm hole)		

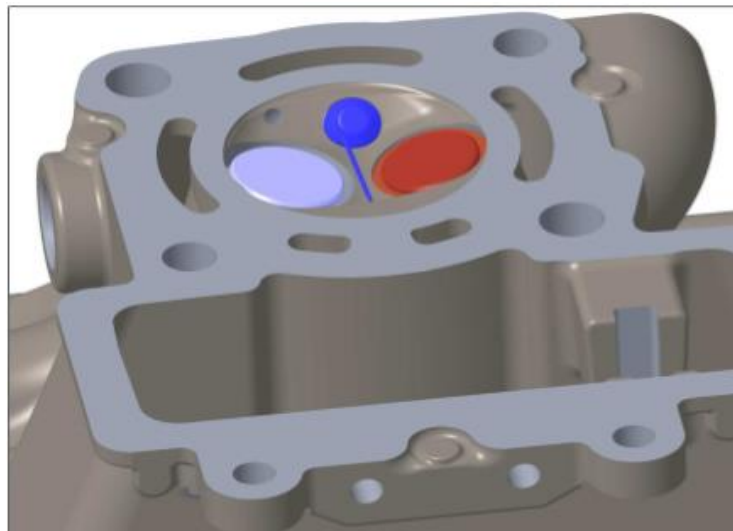


Figure 18.4. Tangential Orifice Pre-Chamber 120mm³ Volume.

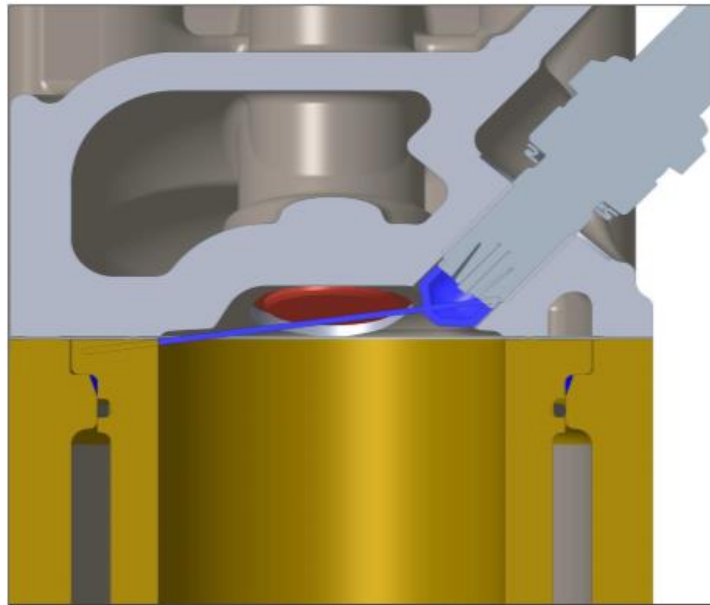


Figure 18.5. Angled Orifice Pre-Chamber 120mm³.

Conclusions

An engine with Phase II upgrades was provided to UW-ERC for continued performance testing. Upgrades included an improved piston profile, a reduced tension oil ring assembly, aligned camshaft lobes, a refinement of the camshaft chain guide geometry, optimized crankshaft/rod lubrication, and an improved crankshaft to flywheel joint design. Testing showed a significant reduction in engine friction and improvement in thermal efficiency. Additional improvement was seen with thermal barrier coating (TBC's) on the piston, cylinder head, and valves. Spark plug passive pre-chamber evaluations yielded a significant increase in combustion speed, but due to heat transfer increases, thermal efficiency improvement was not realized. Performance results are provided under Task 19. Task 18 was successfully completed.

Task 19 Performance Testing of Phase II Prototype

Pathway to Meet Technical Performance Targets (Milestone M19.1)

The results from the preliminary concept 3 testing were combined with the testing results from the concept 0 and concept 1 testing to provide a pathway to achieve 40% electrical efficiency. Figure 19.1 shows the pathway and Table 19.1 details each step. The current engine achieves ~32% BTE. Reducing friction to the concept 1 engine values through improved piston design and higher oil temperatures is expected to increase the BTE to 33.1%. Increasing coolant temperature and reducing heat loss is expected to increase BTE to 33.8%. The concept 0 and 1 engines showed that adding EGR increased BTE by 1.1% absolute, this increases the BTE to 34.9%. Note that this is slightly higher than the peak BTE observed in the concept 1 configuration (34.5%) due to the shortened burn duration and increased combustion efficiency. Adding thermal barrier coatings (TBCs) was found to increase BTE by 1.2% in the concept 0 configuration. If similar results can be found on the concept 3 engine, the BTE would increase

to 36.1%. Further design optimization after continued engine testing (i.e., evaluation of wear areas) is expected to enable further friction reduction. If the mechanical efficiency can be increased to 94% (a typical automotive level at this load), the BTE would increase to 37.4%. Finally, although the burn duration is shortened compared to the concept 1 engine, the combustion duration is still longer than desired. Further CFD analysis will refine the combustion chamber based on the current testing results with a focus on shortening the burn duration. This is expected to enable an improvement in GIE and enable achievement of the program brake thermal efficiency target of 38.5%.

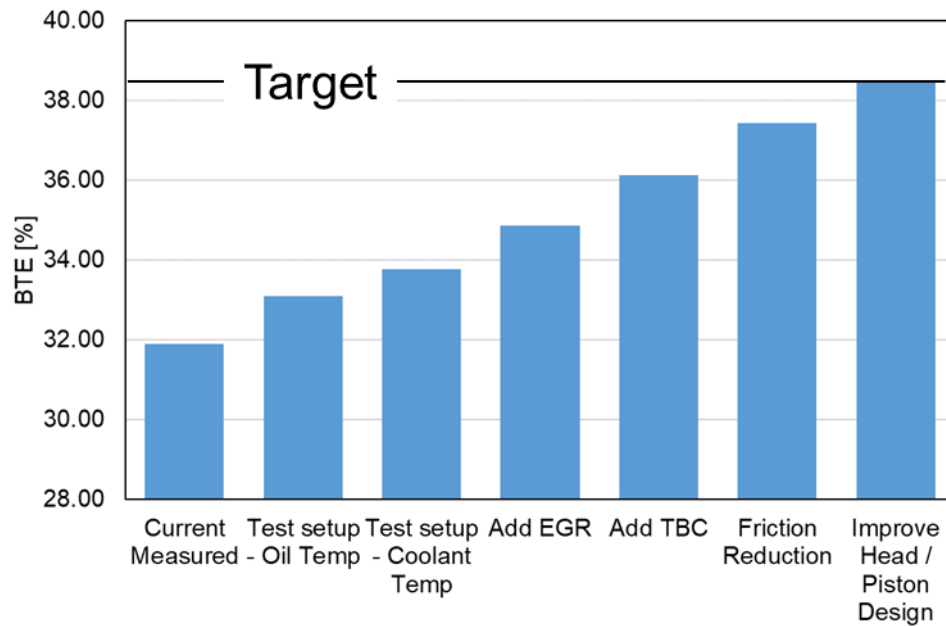


Figure 19.1. Pathway to achieve 40% electrical efficiency. Note that the BTE target is 38.5% per the program specifications that allow consideration of 3% recovery from a thermoelectric device.

Table 19.1. Step-by-step pathway to achieve 40% electrical efficiency.

	BTE	Description	Justification
Test setup - Oil Temp	33.1	Increase oil temp to increase mechanical efficiency to concept 1 levels	Measured values on concept 1 engine ($\eta_{\text{mech}}=90.7\%$)
Test setup - Coolant Temp	33.8	Increase coolant temp to reduce heat transfer to concept 1 levels	Measured 2% lower incomplete combustion, but equal GIE à GIE should increase by $2 \times 0.365 = 0.73\%$
Add EGR	34.9	Observed increase on concept 0 and concept 1 engines	Measured values on concept 0 and 1 engines
Add TBC	36.1	Observed increase on concept 0 engine	Measured values on concept 0 engine
Friction Reduction	37.4	Component analysis after running to reduce losses: increase mechanical efficiency from 0.907 to 0.94	Expected improvement in piston skirt design and liner friction coatings
Improve Head / Piston Design	38.5	Refine combustion chamber based on current test data	Mechanical analysis may allow further reduction in crevice volume; CFD analysis should focus on further reducing burn duration

In addition to performance testing, the concept 3 engine was coupled with a three-way catalyst to demonstrate a pathway to reach the program emissions targets. The catalyst was a platinum, palladium, rhodium formulation at a ratio of 6:20:1 with a loading of 75 g/ft³. An equivalence ratio sweep at a fixed speed of 2400 rev/min was performed and emissions sampling was performed before and after the catalyst. Figure 19.2 shows the pre- and post-catalyst emissions and Figure 19.3 shows the measured and required catalyst conversion efficiency. The catalyst shows substantial NO_x reduction, providing NO_x levels near the program target. The measured NO_x conversion is 99.6% and 99.8% is required to reach the program targets. This value is expected to be within the current measurement uncertainty and we expect meeting NO_x targets is feasible with the current formulation. CO is reduced to 0 ppm post catalyst, easily meeting the CO targets. VOC emissions appear to require additional effort; however, this may be related to the measurement approach. VOC emissions operating on neat methane are expected to be low and additional effort is needed to quantify the uncertainty in the measured VOC. CH₄ emissions are not regulated, but should be monitored throughout the project. Overall, the present testing shows that meeting current CARB emissions targets is feasible.

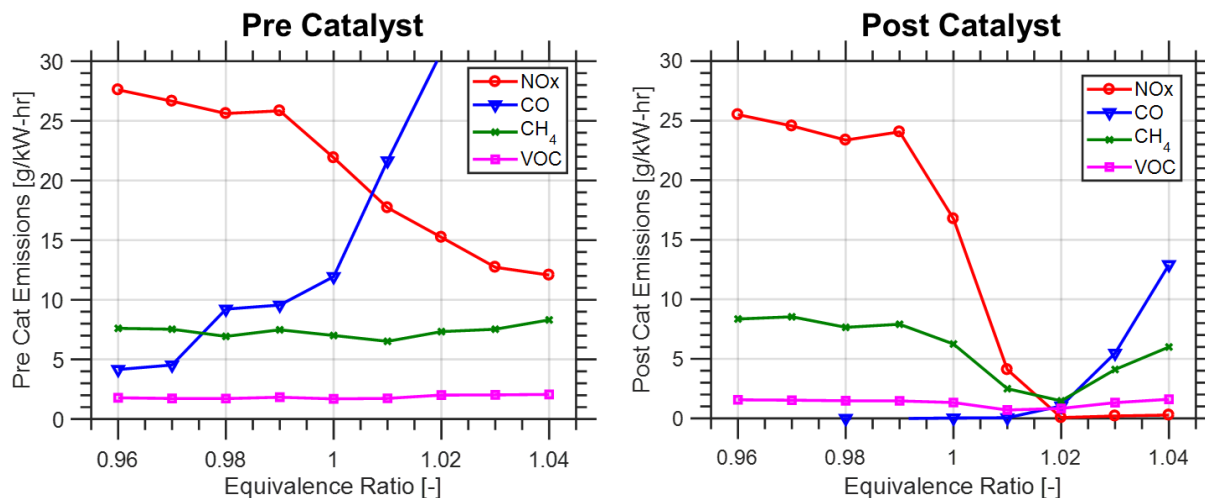


Figure 19.2. (left) Pre and (right) post catalyst emissions from the concept 3 configuration.

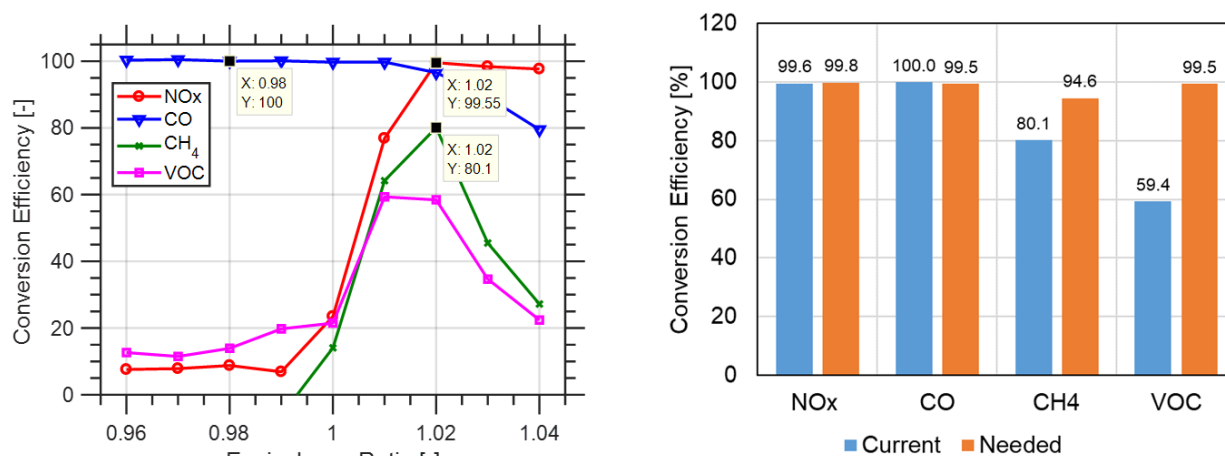


Figure 19.3. (left) Measured catalyst conversion efficiency and (right) comparison of measured conversion efficiency and the efficiency required to meet the current emissions targets.

Other technical performance targets (TPT) from the funding opportunity announcement (FOA) are met based on the engine design and are shown in Table 19.2. Table 19.2 shows full compliance and demonstrates successful achievement with Milestone M19.1 on schedule.

Table 19.2. Summary of technical performance targets (TPT) from the FOA, current status, and pathway to meet the target.

TPT#	Property	Primary Target	Status	Pathway to meet
1.1	Electric power generation capacity	1 kWe	Met	Current Approach
1.2	Fuel to electricity conversion efficiency	≥40%	35.7 with concept 1	Provided pathway with concept 3+
1.3	Useful heat energy output (> 80°C)	>1 kW/kWe	Met: Exhaust > 0.5 kW/kWe at 350°C, Coolant > 1 kW/kWe at 100°C	Current Approach
1.4	Capacity factor	≥99.9%	Met	Current Approach
1.5	Complete system up front cost (not including balance of plant and installation)	≤\$3000/kWe	TBD	Manufacturing and market assessment
1.6	Lifetime	≥10 years	Met by Design	Current Approach
1.7 1.8 1.9	Emissions (NO _x , CO, and VOC)	≤0.07 lb/MW-hr	NO _x and CO are near or exceeding targets. VOC requires increase in catalyst efficiency	Expected increase with increased catalyst inlet temperature
1.10	Total system-out CO ₂ eq (CO ₂ + CH ₄ only)	≤1100 lb/MW-hr	Requires 40% electric efficiency and 94.6% Efficient catalyst – currently measured 80.1%	Expected increase with increased catalyst inlet temperature
1.11	Particulate matter (PM)	≤0.4 g/kW-hr	N/A	Current Approach
1.12	System noise	≤55 dB(A) (3-feet away)	Requires enclosure	Expected to be met with enclosure
1.13	Methane number for operation	≥70	Met	Current Approach

1.14	Number of regular maintenance services	$\leq 1/\text{year}$	Met by design	Current Approach
1.15	Operation and maintenance cost	$\leq \$0.005/\text{kWh}$	Met by design	Current Approach
1.16	Time for regular maintenance	≤ 60 minutes/service	Met by design	Current Approach
1.17	System mass	≤ 150 kg	Met	Current Approach

Summary of Peak Brake Thermal Efficiency from Engine Testing

The project efforts have resulted in substantial increases in BTE compared to the baseline engine. To illustrate this, Figure 19.4 shows the evolution of BTE from the baseline engine to the final design. The project baseline small engine had a BTE of 24.4%. This engine had a displacement of over 160 cc, resulting in a relatively low brake mean effective pressure (BMEP) at 1 kW operation. The low BMEP increased the contribution of friction losses, and substantially reduced the BTE. The engine used a square configuration with a nearly equal bore and stroke. WERC led design work showed that the engine could be downsized to ~70 cc while maintaining the ability to operate at over 1 kW. Further, heat transfer losses could be reduced by increasing the stroke to bore ratio to over 1.5. These changes resulted in a peak BTE of 29.1%. The BTE was further increased to 33% by increasing the compression ratio (CR) to ~17:1. The second round of design iterations was used to optimize the intake geometry to maximize tumble and shorten the burn duration. This resulted in an increase in BTE to 35.1%. Adding thermal barrier coatings resulted in an increase in BTE to 35.5% and increased the exhaust temperature to maintain adequate catalyst performance for methane reduction. Finally, optimizing the EGR rate resulted in an increase in BTE to 35.9% at a stoichiometric air-fuel ratio, enabling the team to meet the stringent NO_x targets.

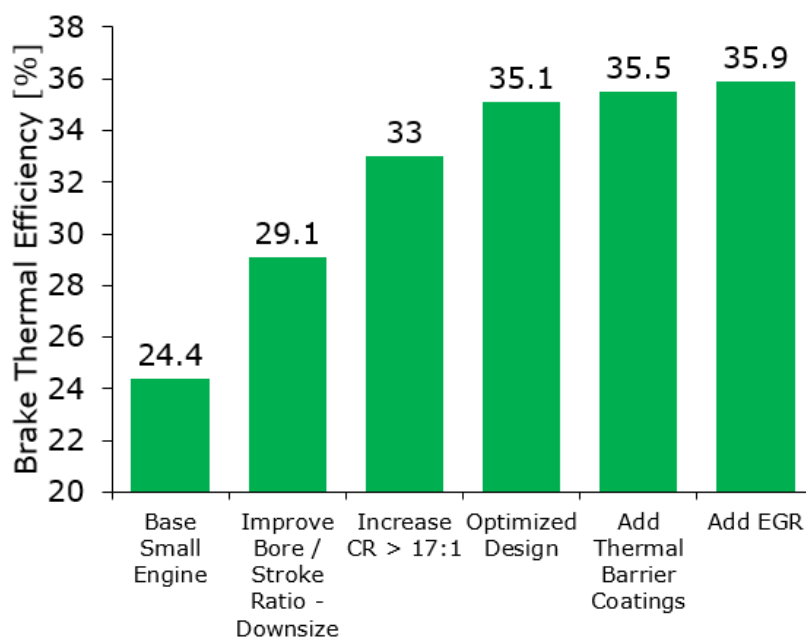


Figure 19.4. Evolution of brake thermal efficiency throughout the project.

Over the course of the project, the BTE was increased by 47%. This was achieved through a systematic reduction in losses in all of the major categories. Figure 19.5 summarizes the loss reductions achieved in this program. Notice that the effort demonstrates that substantial opportunity exists to improve efficiency of small engines, while meeting future emissions targets.

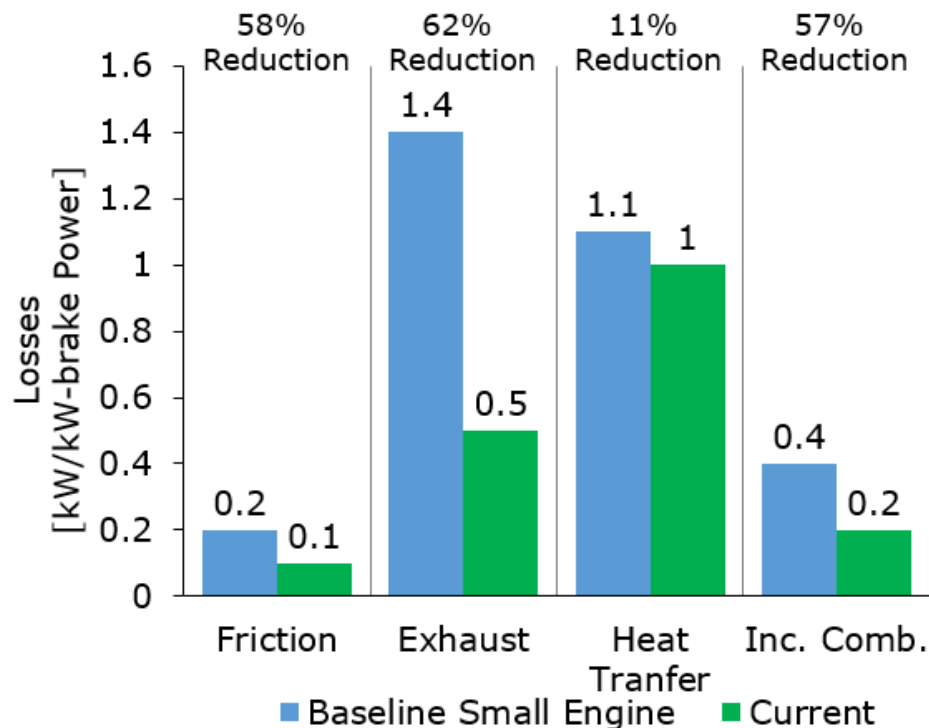


Figure 19.5. Loss reduction achieved over the course of the project.

Task 20 – Evaluation of project developed technology

Summary

The objective of this task is to evaluate project developed technologies (TBCs, friction and wear reduction coatings). In lieu of further endurance testing of a Phase II version of the GENSETS engine, an evaluation of project developed technology on a current production Briggs & Stratton engine was accomplished. Outcomes in reducing engine heat rejection with the use of thermal barrier coatings (TBCs) could have important implications in the performance and durability of Briggs & Stratton engines. Likewise, friction and wear reducing coatings, such as diamond-like carbon (DLCs), could provide similar benefits.

Performance attributes including component operating temperature, exhaust emissions, fuel consumption, and engine power will be evaluated with standard and “treated” engine assemblies. If results are significantly positive, endurance testing will be carried out on a targeted engine for a period of not less than 500 hours or until a major engine failure occurs.

Details

Testing was accomplished on the Briggs & Stratton Vanguard 810cc EFI V-twin engine with TBC treated combustion chamber components (heads, valves, and piston crowns). Emissions testing, temperature testing and knock tendency testing were accomplished.

Figure 20.1 compares HC results on an emission mode weighted basis. The fully treated engine showed hydrocarbon (HC) emissions about 2X the baseline untreated engine assembly. In an effort to better understand the HC increase, the TBC cylinder head assemblies were replaced with standard uncoated components. Less TBC surface area showed a reduced HC impact of only 20% over the baseline.

Changes in critical cylinder head temperatures with the addition of TBC's were evaluated at 3600 rpm, full load. Figure 20.2 shows a nominal temperature reduction of 20° - 40°F with the coated components. This temperature decrease is encouraging, with further reduction thought possible with coating optimization.

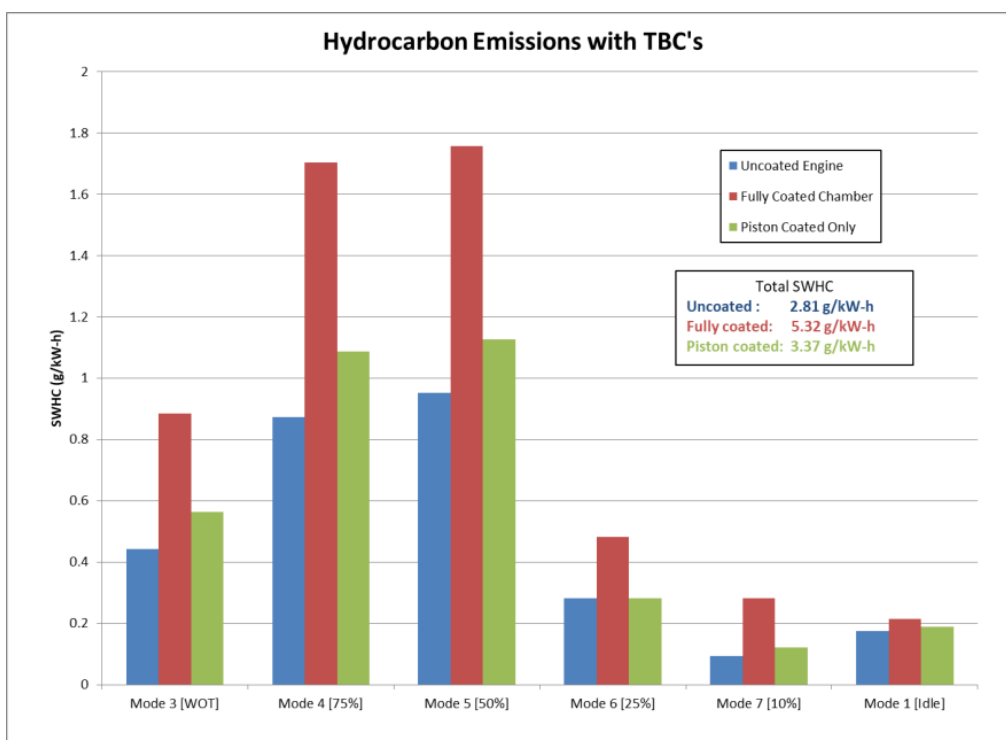


Figure 20.1. Hydrocarbon Emissions Comparison.

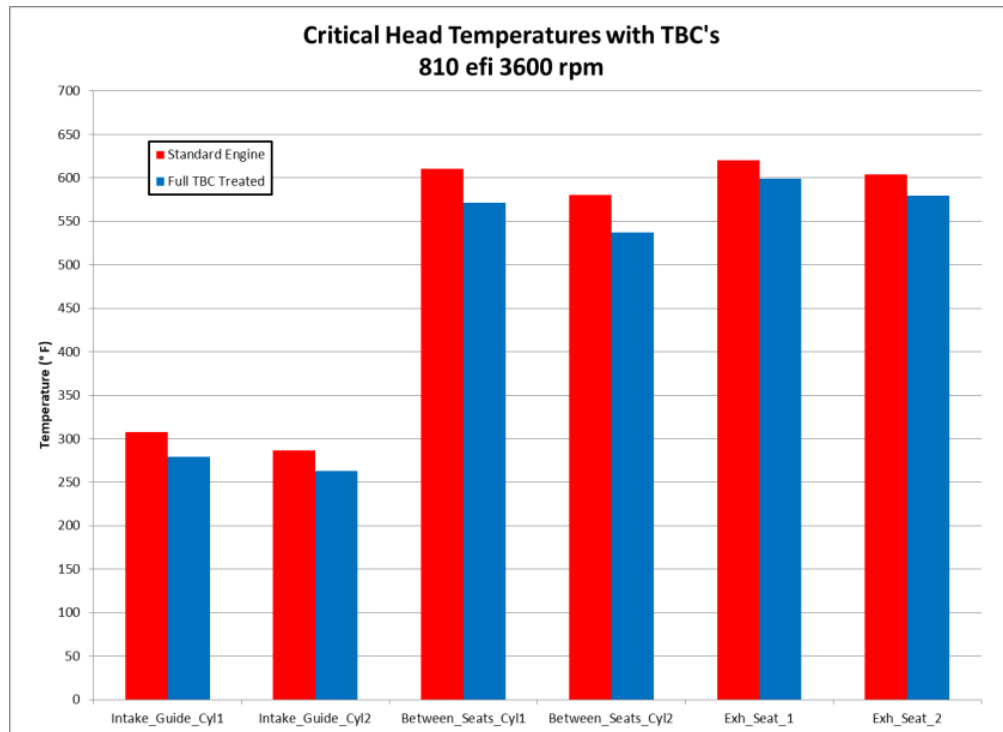


Figure 20.2. Engine Temperature Comparison.

Hot knock evaluations were accomplished with the standard engine and with fully TBC treated combustion chamber components. This comparison was also repeated at a leaner fuel mixture, increasing combustion temperatures and knock tendency. Figures 20.3 and 20.4 provide typical comparisons with advancing ignition timing (lower CA50% values).

Seen from these plots, the TBC's are effective in suppressing knock at normal fuel mixture levels (12:1 AFR) and less effective with lean mixtures (13:1 AFR) where knock tendency increases quickly with lower CA50% values. This is very evident with the higher operating temperature #2 cylinder.

Complete mapping of performance and combustion parameters was postponed because of the increase in HC emissions when using the initial coated combustion chamber components. Adiabatics, Inc. is developing an improved TBC coating with less surface porosity and roughness using sieved particles for a smaller, uniform size as well as an improved top coat sealer. Briggs & Stratton plans to sample and evaluate combustion chamber components with this improved TBC as part of the continuing Spark Ignition TBC task (Task 26).

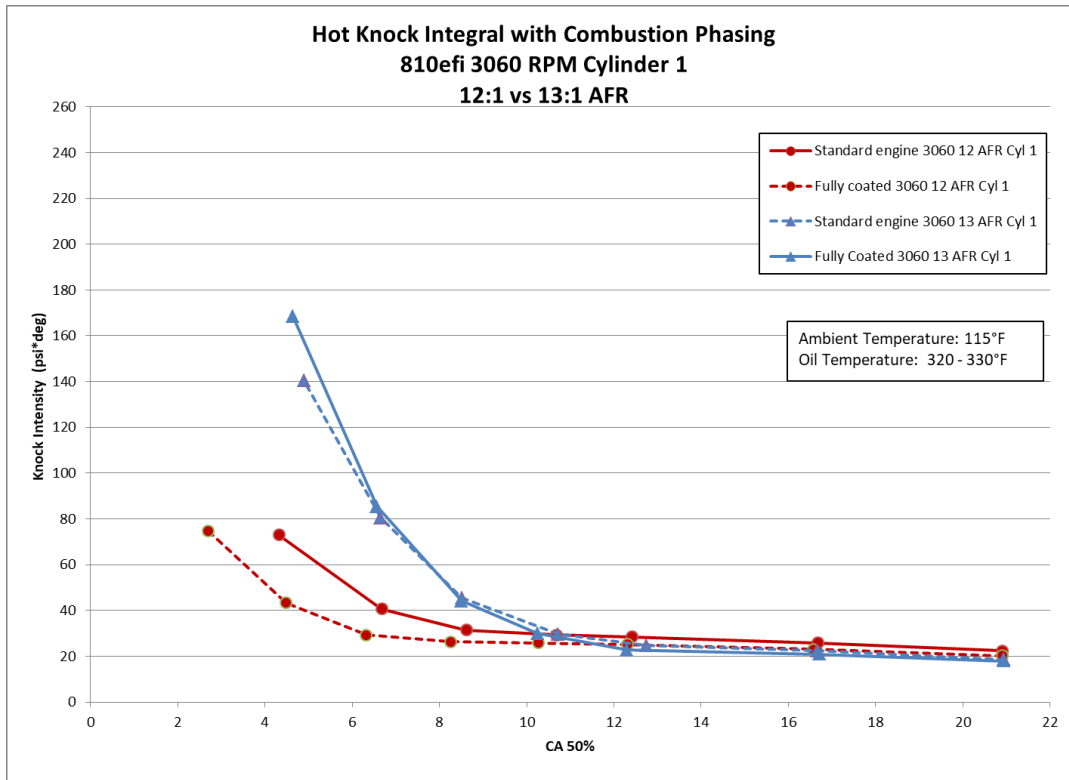


Figure 20.3. Hot Knock Comparisons @ 3060 RPM (Cylinder #1).

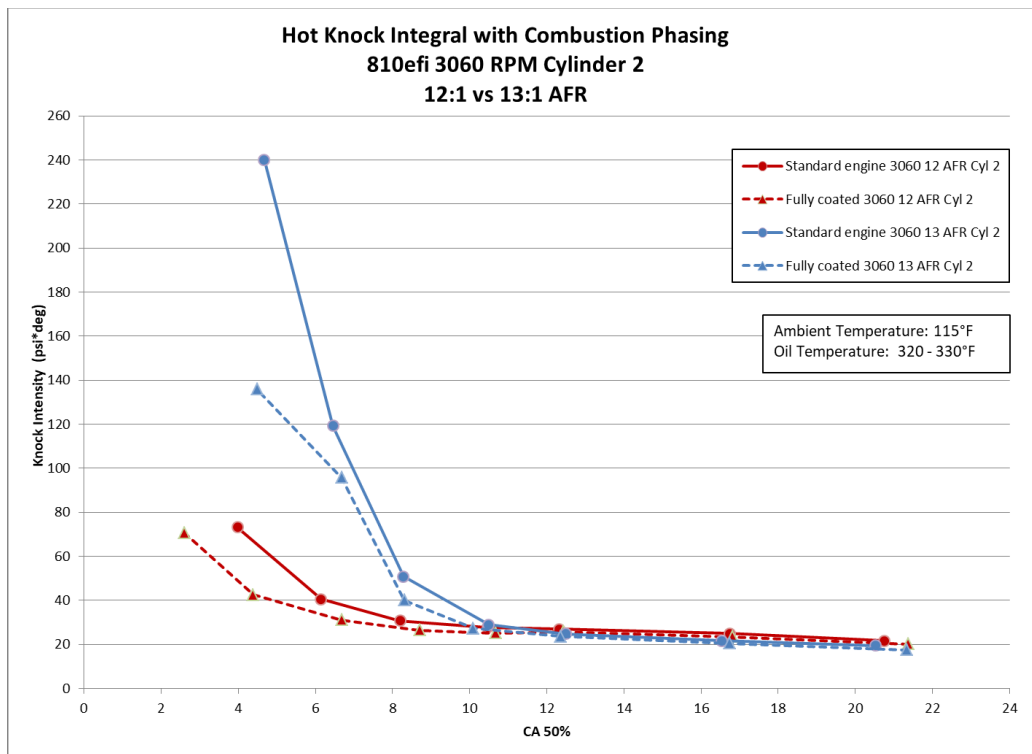


Figure 20.4. Hot Knock Comparisons @ 3060 RPM (Cylinder #2).

Methods of treating the cylinder bores of the engine with low friction coatings continue to be evaluated. A method of coating the present cast-in-place sleeve is being investigated. High fixture cost (large V-twin cylinder casting) and the problem of coating grey cast iron remain significant barriers. Briggs & Stratton continues to evaluate the use of low friction coatings on small single cylinder aluminum sleeve engines.

Conclusions

Thermal Barrier Coating (TBC's) could have important implications in the performance and durability of Briggs & Stratton engines. Initial results from the Briggs & Stratton Vanguard 810cc EFI V-twin engine with PC treated pistons and heads and RC coated valves showed encouraging reductions in critical head temperatures and engine knock tendency. However, a significant increase in exhaust hydrocarbons needs to be understood and corrected for TBC's to be considered on current production engines. TBC formula research to reduce surface porosity and roughness and enhance temperature and knock tendency reduction is incorporated in the continuing Spark Ignition TBC task (Task 26). Task 20 was successfully completed.

Task 21 - Manufacturing and Cost Study

Summary

The objective of this task is to develop bill of materials for Phase II prototype system with refinements identified in Phase II testing and simulation activities. Develop production plan including capital, fixed, and variable costs to manufacture target system.

Details

A focused effort on developing a moderate fidelity cost for the Phase II GENSETS engine was undertaken towards the end of the project timeframe. With Briggs & Stratton purchasing support, study of current production engines, and engineering experience, manufacturing costs were assembled for prototype (~10 engines), pilot (~1000 engines), and early production (~10,000 engine) volumes. See Table 24.1 for a summary of these costs.

Technology readiness level (TRL) is a method of estimating [technology maturity](#) during the product development process. Originated by NASA in the 1970's, TRL's are based on a scale from 1 to 9 with 1 being the exploration of a technology/product and 9 being the commercialization of the product. Briggs & Stratton uses this tool to manage risk in making decisions concerning transitioning products under development into production. The GENSETS engine (engine only, not the complete CHP system application) is thought to be in the TRL 4 to 5 range. That is, the technology has been first demonstrated (TRL 4) and is ready for further development in the lab and in an application (TRL 5).

At the TRL 4 - 5 level, engine production costing is typically of low fidelity as manufacturing details are quite immature. High volume production study and planning is typically not done at this level as significant design changes are expected. With the GENSETS engine, a prototype cost (less than 10 engines) is known based on actual project expenses. A pilot production cost (1000 pieces) has been estimated based on more dedicated fixturing and significant economy of scale. The 10,000 annual

production volume was estimated based on similar engine programs as well as quotations from trusted, purchasing sources.

For the WERC-Briggs & Stratton GENSETS engine, prototyping costs for the engine cast components have been documented. For machining of these parts, as well as numerous billet machined parts, both outside machining sources and the Briggs & Stratton Prototyping Center have provided invoices or labor cost estimates. Finally, engine technician assembly labor costs were estimated. Total engine cost has been detailed at about \$21,200.

For the development of an engine cost estimate at an initial pilot production of 1,000 pieces, volume production tooling was not considered. However, major engine casting tooling and machining fixtures will be more dedicated than at the prototyping level. Some billet machined prototype components would be moved to either forgings or castings. Cost “place holders” were provided for the engine lube and crankcase ventilation systems as designs have not yet matured in these groups. The significant increase in volume is seen to provide component cost reductions in the 60% range. Engines would still be “bench built” by assembly technicians. A short engine test period before CHP assembly is included in the estimate. Total engine cost is estimated at \$5,600. Cost drivers include the crankcase group, the cylinder group, the cylinder head group, and the electrical group. Additional study is suggested to explore alternate sourcing and cost reductions in these groups.

For the 10,000 annual volume cost estimate, several component costs from the current production 810efi engine were used as a starting point. Costs were adjusted higher or lower based on material differences (aluminum versus cast iron), fabrication process (die cast versus cored sand casting), and design features (one piece crankshaft with journal bearings versus an assembled crankshaft with roller bearings). Quotations from known, trusted sources were also developed to supplement the estimation process. Significant casting and machining tooling investments were considered to reduce piece price at this volume level. Cost “place holders” for the engine lube and crankcase ventilation systems were still included, although at a lower level. Assembly and test costs for the 810efi engine (still “bench built”) were used for this volume level. Total engine cost for the 10,000 piece annual volume is estimated at \$1,050 with cost drivers including the crankcase, cylinder head, and electrical groups as well as assembly and test costs.

A 100,000 annual engine volume cost was not estimated at this time as design and manufacturing details are considered too immature and likely to change to deserve the significant advanced manufacturing resources required.

Conclusions

Engine prototyping costs have been documented at \$21,200 for builds of less than 10 engines. For pilot production volumes around 1,000 pieces, engine cost of \$5,600 has been estimated. For initial production volumes of about 10,000 engines, a cost of \$1,050 has been estimated. Supplied quotations, comparison to current production parts, and engineering judgement was used to assemble the estimates. Higher volumes and manufacturing tooling requirements were not estimated as significant design changes were expected with additional engine development. Task 21 was successfully completed.

Table 21.1. Engine Concept #3 Manufacturing Cost Summary.

ARPA-E Engine Program - Residential CHP			
	Prototype Cost (<10)	Cost 1,000/yr	Cost 10k/yr
Crankshaft / Rod Group	\$ 5,789	\$ 407	\$ 81
Piston Group	\$ 1,573	\$ 509	\$ 89
Crankcase Group	\$ 2,172	\$ 674	\$ 116
Sump Group	\$ 451	\$ 10	\$ 9
Cylinder Group	\$ 2,560	\$ 1,026	\$ 54
Timing Group	\$ 512	\$ 199	\$ 33
Cylinder Head Group	\$ 4,139	\$ 1,397	\$ 248
Rocker Cover & Gasket Group	\$ 502	\$ 28	\$ 8
Flywheel Group	\$ 343	\$ 137	\$ 18
Electrical Group	\$ 1,338	\$ 656	\$ 117
Fuel System Group	\$ 258	\$ 108	\$ 43
Breather Group	\$ -	\$ 30	\$ 15
Oil Pump / Reservoir Group	\$ 960	\$ 50	\$ 25
Exhaust Group	\$ 14	\$ 14	\$ 14
Labor and Overhead Group	\$ 600	\$ 320	\$ 160
Cooling System Group	not included, part of CHP System		
Air Cleaner Group	not included, part of CHP System		
TOTAL	\$ 21,210	\$ 5,566	\$ 1,032

Task 22 - Commercialization Assessment

Summary

Develop a commercialization plan that includes further technical development, manufacturing development, sourcing, system costing, and marketing channel considerations. Move project through first two Gates (Discovery and Concept Analysis) of six gate process.

Details

Briggs & Stratton provided a presentation on the company's perspective for commercialization of GENSETS technology at the March 2018 ARPA-e GENSETS annual meeting. After careful review of the various elements related to successful commercialization of new technology in an established market, it

was concluded that there was a low probability of a sufficient market penetration that would support the economies of scale necessary for mass market adoption. The challenges of retrofitting mCHP systems to the installed base and with payback periods well beyond the consumer tolerance of less than 2 years suggested that a residential mCHP system would be limited to early-adopters and eco-focused new construction. For this reason, B&S opted not to pursue development of a residential mCHP appliance.

In addition to the pure economic analysis, the decision not to pursue development of the mCHP appliance was buoyed by the significant challenges associated with sales channel development, and after-sale service and maintenance support. The necessary long-term investment was not compatible with the needs of the broader organization.

Focus was shifted to potential applications for the technology developed in the prime-mover engine project. B&S continues to be interested in the application of TBC's and low-friction strategies in future engine designs as a means to improve power, fuel efficiency, and durability. Task 20 summarizes development and challenges discovered to date.

Conclusions

Briggs & Stratton provided a presentation of various elements related to successful commercialization of new technology in an established market. The challenges of retrofitting traditional systems and providing an attractive payback period would limit the mCHP market acceptance. Sales channel development and after-sales service support were additional concerns. For these reasons, B&S opted not to pursue development of a residential mCHP appliance. Task 22 was successfully completed.

Task 23 – Manufacturing and Cost Study of TBCs

Summary

This task objective was focused on determining the best method for commercializing the TBCs and took into account the raw materials cost, quality control, finishing costs, coating surface preparation, application technique (spraying, HVLP, brushing, dipping, etc.), and curing method. Figure 23.1 shows the timeline for Task 23: Manufacturing and Cost Study of TBCs. Milestone 23: Demonstration of a Quality Control Plan for Manufacturing Engine Components with TBC Applied at the Component Factory is the only milestone for this task, so all work was focused on the deliverable at the end of Q2 2019. Task 23 was completed at the end of Q2 2019.

In Task 23, different iterations of a re-circulating pump/spray mixer were designed and tested in order to combat density discrepancy. These pumps/mixers allowed for much more consistent density data, although improvements are still being considered.

In order to verify Adiabatics methods for measuring density and porosity, samples were sent in to Particle Technology Labs for porosimetry testing which was not successful.

Because the RC Coating was found to fully cure at a higher temperature than initially thought, an alternative laser method for curing was investigated for the RC coating which requires higher cure

A Process Failure Modes and Effects Analysis (PFMEA) was created and revised several times before it was finalized at the end of Q2 2019.

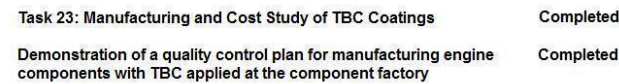


Figure 23.1. Timeline of Task 23, Manufacturing and Cost Study of TBCs.

Details

Density variation of the TBC coatings had been identified as a major problem all throughout the project. Several theories were hypothesized and tested, including humidity, spray parameters, and measuring procedures, but none of them proved to be the cause. Variation was determined to be caused by a mixing issue. Although coatings were completely mixed before spraying and manually mixed in-between each spray pass, the variation was traced to time of sample spraying. It was determined that the coatings (both RC and PC) must be continuously mixed while in the spray gun, otherwise, the resin to additive ratio fluctuates. In order to prove this density variation was a result of sample spray order, a test was conducted in which three sets of five samples were labeled and sprayed in numerical order using a new spray gun for each set. The samples were then weighed and densities were calculated using Archimedes' principle with results showing a very consistent density variation pattern as shown in Figure 23.2. To reduce variation, a prototype re-circulating spray system was created with results showing more consistent density. Adjustments were implemented later in order to reduce variation further.

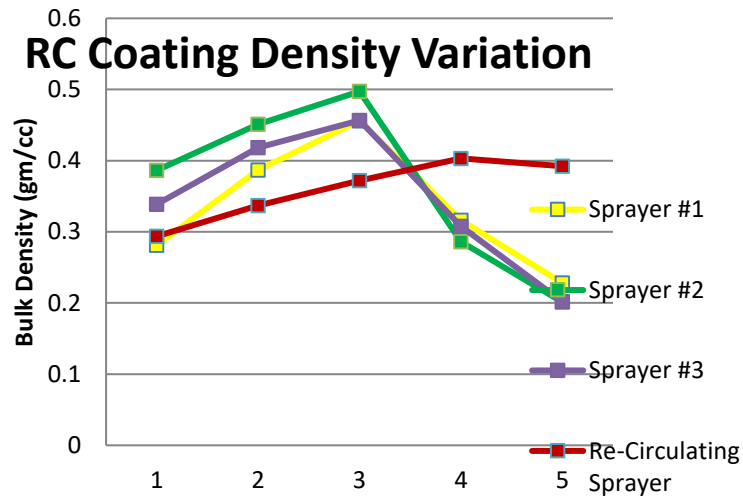


Figure 23.2. Density variation of the RC coating using regular spray guns vs a spray gun that utilizes a re-circulating pump.

However, in 2019 it was found that by sifting microspheres into smaller sizes and filtering out the broken/cracked ones, density measurements have become much more consistent. For this reason, a simpler spray system that only incorporates a mixer rather than a re-circulating pump had been developed and is currently being used. As shown in Figure 23.3, early results were promising. Even without the use of a recirculating sprayer, the density results have so far proven to be more consistent.

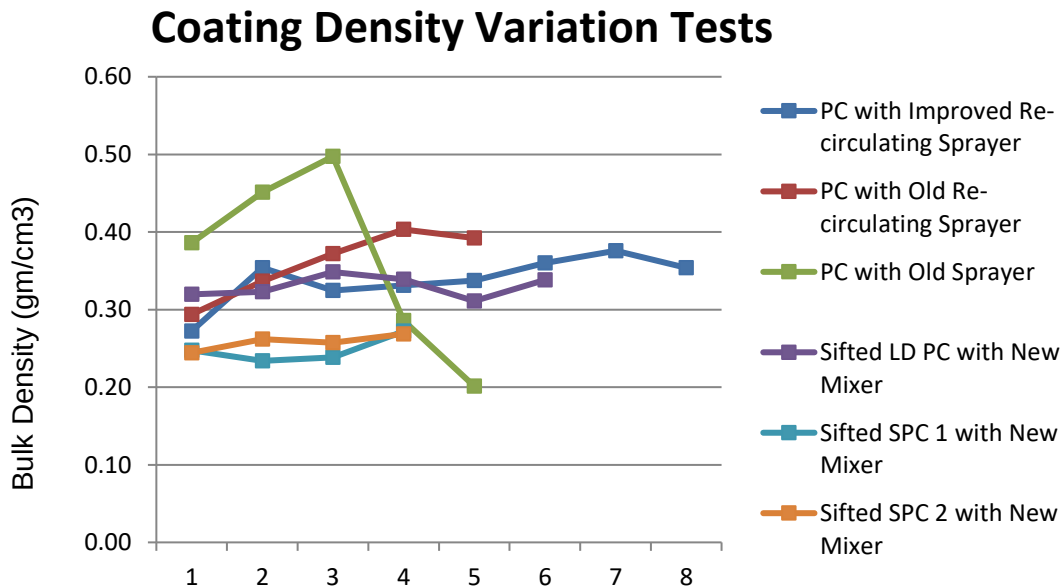


Figure 23.3. Coating density variation of coatings through the use of different sprayer systems.

In order to meet Milestone 23: Demonstration of a Quality Control Plan for Manufacturing Engine Components with TBC Applied at the Component Factory, a preliminary PFMEA was created in Q3 of 2018. After several revisions, the PFMEA was reviewed and finalized in Q2 of 2019 for its deliverable. Shown below is the finalized version of the PFMEA.

PFMEA Form														
Process/Product Name: <u>PC and RC Coatings</u>					Prepared By: <u>Alex Kamo</u>			PFMEA Date (Orig.): <u>9/7/2018</u>						
Process Step/Input	Potential Failure Mode	Potential Failure Effects	SEVERITY (1 - 10)	Potential Causes	OCCURRENCE (1 - 10)	Current Controls	DETECTION (1 - 10)	RPN	Action Recommended	Resp.	SEVERITY (1 - 10)	OCCURRENCE (1 - 10)	DETECTION (1 - 10)	RPN
What is the process step, change or feature under investigation?	In what ways could the step, change or feature go wrong?	What is the impact on the customer if this failure is not prevented or corrected?		What causes the step, change or feature to go wrong? (how could it occur?)		What controls exist that either prevent or detect the failure?			What are the recommended actions for reducing the occurrence of the cause or improving detection?	Who is responsible for making sure the actions are completed?				
Purchasing of Materials (resin, microspheres, additives, sealer)	Materials might not be the same between batches	Coating may flake off or not perform to expectations	3	Materials might not possess the same properties causing variation in bond strength, density, and thermal conductivity	8	None are being utilized currently	9	216	Test each batch of materials using density and bond strength before utilizing coatings	Alex and Lloyd Kamo	3	8	2	48
Preparation of Substrate:	Surface not prepared well enough impacting bond strength of coating	Coating may flake off	3	NACE-1/SSPC-SP-5 surface preparation standard (White Metal Blast Surface) isn't met	5	Visual inspection in order to ensure substrate has been prepared properly	3	45	Thickness probes may be able to measure grit-blasted areas (will have to be tested first)	Alex and Lloyd Kamo	3	4	2	24
Removal of Oil	Oil may not be removed completely	Coating may flake off	3	Part wasn't soaked long enough in cleaning solution or cleaning solution didn't completely remove all oil	6	Physical and visual inspection of parts to ensure oil has been completely removed	3	54	Steel and cast iron parts will be heated in furnace to burn off oil (650F) and aluminum parts will be soaked in Greasemaster cleaning solution rather than 409 (it is more effective)	Alex and Lloyd Kamo	3	4	3	24
Masking Part for Grit-Blasting	Part is not masked well enough which causes unwanted areas to be grit-blasted	Unwanted grit-blasting in certain areas could cause a leak in the gasket	6	Masking doesn't completely mask off all non-coating areas or it comes off during grit-blasting	3	Masking is inspected before and during grit-blasting	2	36	Mask will be molded so that tape will no longer be necessary	Alex and Lloyd Kamo	6	2	2	24
Grit-Blasting	Part is not grit-blasted up to NACE-1/SSPC-SP-5 surface preparation standard (White Metal Blast Surface) standards	Coating may flake off	3	Grit becomes dull after too much use or not enough time is spent grit-blasting	3	Visual inspection in order to ensure that NACE-1/SSPC-SP-5 surface preparation standard (White Metal Blast Surface) standards are met	3	27	Grit will be changed after a certain amount of use (usage will be monitored)	Alex and Lloyd Kamo	3	2	3	18
Removal of Masking	Masking isn't completely removed before cleaning	Masking residue may be left on the substrate, weakening coating bond strength causing coating to flake off	3	Masking isn't completely removed from the part	3	Visual inspection in order to ensure all masking is removed before grit-blasted area is cleaned	2	18	Mask will be molded beforehand so that it fits perfectly with each part and ensures a complete masking that is easily removable	Alex and Lloyd Kamo	3	2	2	12

Cleaning of Grit-Blasted Area	Part is not cleaned up to standards or grit isn't completely removed from part	Leftover grit may become embedded in the coating, causing a change in performance or bond strength	3	Grit-blasted area isn't cleaned well enough by a solvent or blasted with air	3	Visual inspection to ensure that the entire area has been cleaned	2	18	Part will be dipped into solvent tank and pressurized air will be applied	Alex and Lloyd Kamo	3	2	2	12
Re-Masking for Coating	Part is not masked well enough which causes unwanted areas to be coated	Masking residue may be left on the substrate, weakening coating bond strength causing coating to flake off	3	Masking isn't correctly placed onto or completely removed from the part	3	Visual inspection to ensure that masking completely covers non-coated areas	2	18	A mask will be molded and will be inspected to ensure that it is correctly placed on each part	Alex and Lloyd Kamo	3	2	2	12
Coating Application Process:	Coating isn't mixed well enough before or during application, spray gun gets plugged, overspray. Masking isn't removed before curing	Coating may flake off or not perform to expectations	3	Spray gun wasn't cleaned well enough beforehand, too much air pressure is used, coating isn't mixed well enough, etc.	7	Spraying guidelines have been determined and must be followed during each application	5	105	Re-circulating pump will be used in order to ensure that coating remains well mixed throughout coating process	Alex and Lloyd Kamo	3	4	4	48
Sifting of Microspheres	Microspheres aren't sifted to the correct size, resulting in a more or less dense coating and a rougher coating surface	Coating may impact combustion due to it affecting combustion air flow and swirl leading to an increase in hydrocarbons	7	Damaged sieve allowing microspheres to pass through that shouldn't	3	Sieves are inspected on a regular basis in order to ensure that they haven't become damaged	2	42	Sieves should be replaced on a to-be-determined schedule depending on the recommended lifetime of a sieve	Alex and Lloyd Kamo	7	2	2	28
Formulation of Coating	Coating isn't formulated properly (too much or too little of its contents)	Coating may flake off or not perform to expectations	3	Too much or too little resin or additives are put into coating	3	High precision scale is used to weigh resin and additives before they are mixed together	1	9	None. As long as current controls are exercised correctly, there should rarely be an occurrence	Alex and Lloyd Kamo	3	3	1	9
Mixing of Coating	Coating isn't properly mixed before or during application causing a problem in the resulting coating	Coating may flake off or not perform to expectations	3	Coating isn't mixed well enough during application	10	Coating is manually mixed after each pass	5	150	Re-circulating pump will be used in order to ensure that coating remains well mixed throughout coating process	Alex and Lloyd Kamo	3	5	2	30
Spraying of Coating	Coating isn't sprayed following the guidelines set forth: As soon as possible after formulation, 20 PSI, 6-10 inches away from part, continuously mixed, 2 thin layers	Coating may flake off or not perform to expectations	3	Spraying guidelines have not been followed	7	Spraying guidelines have been followed	2	42	Test piece will be sprayed alongside part in order to ensure that density and bond strength are consistent with expectations	Alex and Lloyd Kamo	3	5	2	30
Cleaning of Spray Gun	Spray gun isn't cleaned properly between batches which causes it to become plugged during next application	Coating will not be sprayed properly and will result in the coating flaking off or not performing to expectations	3	Cleaning guidelines are not met in-between batches of coating	6	Cleaning guidelines are met: between batches, gun is taken apart and washed out and then solvent is sprayed through the gun and inspected to ensure that its passageways are unblocked	2	36	New spray gun will be used whenever spray gun isn't performing as expected	Alex and Lloyd Kamo	3	4	2	24

Removal of Masking	Masking isn't completely removed before curing	Masking residue may be left on the substrate, weakening coating bond strength and causing coating to flake off	3	Masking isn't completely removed from part	3	Visual inspection to ensure that masking has been completely removed after part has been coated	2	12	Mask will be molded beforehand so that it fits perfectly with each part and ensures a complete masking that is easily removable after coating	Alex and Lloyd Kamo	3	2	2	12
Curing Process for Coating	Coating isn't cured to a high enough temperature or for a long enough duration	Coating may flake off due to low bond strength	3	Furnace controller isn't set up correctly or malfunctions	4	Coating is cured to 390F for at least 3 hours for aluminum and at 850F for cast iron and steel before part is removed from furnace	9	108	Test piece will be cured alongside part in order to ensure bond strength is consistent with expectations	Alex and Lloyd Kamo	3	4	2	24
Sealing of Coating	Too much or too little sealer is used, resulting in a coating that is too dense or one that has too much porosity	Liquid fuel may be absorbed into the porosity and not burn during combustion, increasing hydrocarbons	7	Too few or too many layers of sealer are applied to the coating	5	Quality control sheets are filled out which keep track of the amount of sealer layers applied to coating	1	35	Test piece will be cured alongside part in order to ensure density and porosity is consistent with expectations	Alex and Lloyd Kamo	7	4	1	28
Curing Process for Sealer	Sealer isn't cured to a high enough temperature or for a long enough duration	Lack of sealer cure may cause coating to flake off due to low bond strength	3	Furnace controller isn't set up correctly or malfunctions	4	Sealer is cured to 390F for at least 3 hours for aluminum and at 850F for cast iron and steel before part is removed	9	108	Test piece will be cured alongside part in order to ensure bond strength is consistent with expectations	Alex and Lloyd Kamo	3	4	2	24
Thickness Verification of Coating	Coating may be too thin or too thick	Coating may flake off if too thick or not perform well if too thin	3	Coating was applied too thin or too thick	10	Thickness probes are used to measure thicknesses of the coating	3	90	Part is heated before spraying which will cause coating to set-up and not run nearly as much	Alex and Lloyd Kamo	3	9	3	71
Inspection and Removal of Unwanted Coating before Shipping	Coating may be applied to unwanted areas due to overspray or running of the coating	Coating may cause problems with engine installation	8	Unwanted coating wasn't completely removed before shipping	9	Close visual inspection is used in order to identify any possible issues	2	144	QC sheet must be signed by employee that breaks down exactly which areas of each part must be inspected before shipping	Alex and Lloyd Kamo	8	9	1	72
Shipping/Handling	Coating may be damaged during packaging or transporting	Coating may be damaged upon acceptance and flake off during testing	3	Parts are not packed well enough (i.e. not enough packing materials were used)	2	Shipping manager must sign off on each order to ensure that packing guidelines have been met	1	6	None. As long as current packing guidelines are met, there should rarely be an occurrence	Alex and Lloyd Kamo	3	2	1	6

Figure 23.4. Finalized PFMEA for applying both the RC and PC Coatings.

Conclusions

Milestone: 23.1: Demonstration of a Quality Control Plan for Manufacturing Engine Components with TBC Applied at the Component Factory was initially expected to be completed at the end of Q4 of 2018, but was completed at the end of Q2 of 2019 after a project extension was granted. This milestone was completed after a PFMEA, which addresses possible causes for coating failures as well as the optimized application procedure, was created and finalized for delivery. Currently, the TBCs are applied at Adiabatics' facility, but preliminary discussions were had with Permawick Corporation if large quantities of coating would have to be applied in the future.

Task 24 - Techno-Economic Analysis

Summary

The objective of the Techno-Economic Analysis is to identify the key attributes related to cost and performance that will increase the likelihood of market acceptance relative to alternatives. These cost and performance attributes will be used to inform technical project direction. Technology costs will be evaluated on a total cost of ownership (TCO) basis. Information, methods, and models developed as part of the Techno-Economic Analysis will be used to strengthen the value proposition to potential customers and strategic stakeholders.

Results

Over the course of the three year project there have been significant changes to the commercialization related activities for the GENSETS program. Techno-Economic Analysis (TEA) focus shifted from the mCHP application, the focus at program initiation, to the prime-mover technology powering a mCHP appliance. Along the way, it was demonstrated that the GENSETS performance and cost targets for the prime-mover may not be appropriate for large scale adoption of the technologies. This has led to a reconsideration of the deliverables, focusing on project developed technologies that may be appropriate for current production Briggs & Stratton engines. Task 20 summarizes the technical evaluation of Thermal Barrier Coatings (TBC's). If the hydrocarbon emissions increase can be mitigated and the temperature and knock tendency further improved, there would be considerable interest in TBC application to certain Briggs & Stratton engines.

Discussions with Adiabatics staff indicated the TBC application would be in the \$0.50 - \$1.00 per cylinder range. This could be of value with increases in engine power, fuel efficiency, and durability, improvements thought possible directly with TBC application or enabled with the use of TBC's (such as allowing higher compression ratio or operating with lean fuel mixtures). Volume production methods of applying and curing of TBC's on pistons, cylinder heads, and valves would also require investigation to ensure a robust and consistent application.

Conclusions

Further development of TBC's must mitigate the increase in HC emissions seen with initial testing. If further development proves significant benefits to engine power, fuel efficiency, and durability at the projected cost adder, it is likely that Briggs & Stratton would implement TBC's on some engines. Task 24 was successfully completed.

Task 25 - Technology-to-Market (T2M)

Summary

The objective of the Technology-to-Market activities is to identify and execute the necessary steps associated with commercialization of the subject technology. Technology-to-market activities include the protection of intellectual property, market assessments, manufacturing reviews, talent and knowledge management, and securing of continued funding beyond the initial ARPA-E program.

Results

Intellectual Property Protection

Briggs & Stratton filed US Provisional Application No. 62/641952 titled "Engine with Compact Valvetrain" on March 12, 2018. The GENSETS engine is described in detail, with focus on the unique Type II valvetrain arrangement. The application was not converted (required by March 12, 2019 to retain the original priority date), after discussion with ARPA-e staff due to the narrow scope of the IP with limited usefulness to Briggs & Stratton. Briggs & Stratton is interested in further development of TBC technology as protected by Adiabatics filed IP.

Design and Development Continuation

One area of effort on this task was the development of an engineering plan to advance the GENSETS Phase II engine towards commercialization. The initial GENSETS concept engine proved to have exceptional efficiency with early indications of good durability. Still, there are several design and development areas that require additional investigation for improved performance, durability, and integration into the mCHP application

Design

For the initial concept, the engine lubrication was provided by a dry sump system including electrically driven supply and scavenge pumps. The system worked well, with some difficulty maintaining an adequate oil temperature. Heat loss from the oil tubing, control valves, and reservoir was higher than anticipated. The reservoir provided the added flexibility to size oil volume to ensure acceptable oil quality and quantity over the desired maintenance interval. Alternatives to the dry sump reservoir system should be considered such as a large, active oil sump or a small active sump with a make-up system for consumption and oil quality maintenance. In any case, the inclusion of an engine driven supply and/or scavenge pump should be explored. Different types of pumps, such as a gear pump or gerotor pump, should be considered as well as different methods to drive the pump (gear drive, direct drive, etc).

Inspection of both intake and exhaust cam follower hydraulic lifter (HLA) sockets has shown a contact misalignment as evidenced by off-center polishing of the spherical geometry (see endurance summary under Task 15). This condition causes improper valve dynamics and high wear rates. A geometry review and tolerance stack-up is needed to review the design. Additionally, a full inspection of components exhibiting this misalignment should be accomplished to ensure design intent.

In addition to investigating the contact misalignment mentioned above, an alternative cylinder head and valve train design should be considered. The large included angle between intake and exhaust valves of the current design requires an expensive cylinder head casting as the port runners have a large radius of curvature and are very long. This creates a tall and wide head casting. Flow efficiency and in-cylinder motion are excellent but trade-offs should be investigated. Also, the Type II valve train is highly efficient but complex and costly to execute. The twin lobe exhaust cam requires precise machining and a complex cam follower. Again, trade-offs should be evaluated.

Another design study of interest would integrate the engine crankcase and cylinder into one casting. This configuration would likely be a component part and assembly cost reduction as well as a potential

improvement in engine reliability (fewer joints to seal) and improved powertrain stiffness. The wet sleeve configuration could be maintained or eliminated with an integrated water jacket core.

Exhaust gas recirculation (EGR) was an important technology needed to achieve high overall engine efficiency. A laboratory level system was used during performance development. Future design and development efforts should focus on a valving and control system integrated into the GENSETS engine design.

Potential design study should also include the inclusion of an integrated alternator system, properly sized for engine output and CHP needs. The relative low required output makes an under-flywheel system attractive, although a bolt on or belt drive system should also be considered. A more complete engine/alternator system would make endurance testing easier to accomplish and would allow the initial development of a CHP system concept.

Development

During performance testing of the GENSETS Phase II engine at UW-Madison ERC, a very significant efficiency difference was seen between 2 cam covers that were thought to be identical. At the time, a difference in valve train oiling was thought to be causing this difference but additional comparison tests did not uncover the issue. Valve train oil flow volume and locational direction effects need to be studied to optimize lubrication and engine efficiency.

Component changes to reduce engine friction are recommended for further evaluation. A cylinder sleeve with a low friction Iron-Titanium coating (available from Adiabatics, Inc) should be evaluated. A 20% reduction in oil ring tangential tension did not show a detrimental increase in oil consumption during UW-Madison ERC testing. Further tension reduction should be evaluated for friction reduction. Passive spark plug pre-chamber evaluations at UW-Madison ERC showed a significant improvement in combustion speed, but without the expected increase in engine efficiency. Higher heat transfer was thought to be causing the efficiency loss. Further design study and development tests are recommended to optimize the pre-chamber configuration to reduce heat transfer losses.

A limited duration (500+ hours) endurance test was accomplished on a Phase II GENSETS engine but still showing several areas requiring improvement (see the endurance summary report in Task 15 above). It is recommended for development continuation that these deficiencies be corrected and evaluated with a longer term endurance test of 2,500 to 5,000 hours on 3 engines. Pre-test measurements and periodic inspections during the test period are required to establish component wear rates and to extrapolate an engine life estimate.

Conclusions

Continued design and development tasks to advance the GENSETS Phase II engine towards commercialization have been provided. Exploring alternatives for cost and durability optimization as well as continued efforts to improve engine efficiency are discussed. The Thermal Barrier Coating (TBC's) technology, as developed with Adiabatics, is the project IP of most interest for current production Briggs & Stratton engines. Task 25 was successfully completed.

Task 26 Spark Ignition TBC's

Summary

The coatings applied to the small-bore natural gas engine showed increases in brake thermal efficiency. Testing performed at UW-Madison using natural gas suggested minimal increases in HC emissions; however, complimentary testing at Briggs and Stratton using a different engine fueled by gasoline showed increases in HC emissions. The purpose of this task is to provide additional information to evaluate changes in HC emissions when the coatings are applied. Results show increases in HC emissions when using the SPC piston for both gasoline and methane cases. Additional efforts would be required to minimize HC emissions.

Details

Experiments were conducted using a medium-duty test cell. Key parameters of the test cell are given in Table 26.1.

Table 26.1. Key parameters of medium-duty engine test cell used for conducting experiments.

Parameter	Value(s)
Engine	2013 Cummins 6.7 L ISB
Engine Type	Medium Duty 6 Cylinder Diesel Engine
Configuration	Inline six-cylinder converted to single-cylinder operation
Displacement [L/cyl]	1.17
Bore [mm] X Stroke[mm]	106.9 x 124
Compression Ratio [-]	17.3:1
Fuel Pump	Bosch CP3
Particulate Measurement	AVL 415SE Smokemeter

The stock pistons are made of cast aluminum and feature a stepped lip design; a schematic of the piston's profile is shown in Figure 26.1 and this profile is used in all pistons studied in this report. For the TBC pistons, the thickness of the coating was compensated for by first removing a layer of aluminum from the substrate. The volumes of the bowls were compared to ensure that the coating process did not affect the displacement and/or the compression ratio.

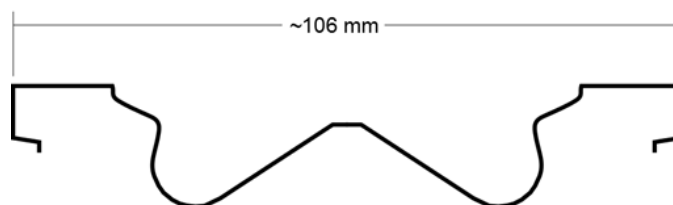


Figure 26.1. Piston profile of stepped lip piston used in ISB6.7 engine.

Two types of TBCs are chosen for study: a conventional YSZ coating and an advanced version of the organic phosphate coating developed in the current program referred to as the SPC. Properties of the coatings are shown in Table 26.2.

Table 26.2. Properties of TBCs used in this study.

Coating	YSZ	SPC
Thickness (Top/Bond) [mm]	0.10 /0.10	0.15 /none
R _a Surface Roughness [μ m]	5.5	7
Density (Top/Bond) [gm/cc]	3.34 /4.10	0.35 /none
Porosity (Top/Bond) [%]	8.5 /8.5	25 /none
300°C Vol. Spec. Heat (Top/Bond) [kJ/m ³]	2200 /2300	380 /none
Thermal Conductivity (Top/Bond) [w/m ³ -K]	1.4 /1.9	0.31 /none

An NGK R5671A spark plug was placed in the custom-made cylinder head and three port-fuel injectors were used to introduce both gasoline and methane into the intake manifold. Due to the high compression ratio of the ISB engine, a considerable vacuum was needed at the intake manifold to simultaneously achieve a stoichiometric equivalence ratio and avoid autoignition and modifications were made to the intake manifold to accommodate this.

Due to the constraints posed by the equivalence ratio and knocking/autoignition conditions, a single operating point was chosen for study. The intake pressure was fixed at the lowest vacuum that could be achieved, 35 kPa, which uniquely fixed the airflow. Fueling was then adjusted to achieve a stoichiometric equivalence ratio. Spark timing was swept between stability and knock limits to understand the role of combustion phasing in the emissions trends. The study was subsequently repeated with the SPC piston in place to fulfill the requirements for M26.

Figure 26.2 shows the trends in CO, HC and NO emissions as spark timing is varied. The results show increases in HC emissions when using the SPC piston for both gasoline and methane cases. In most cases, the increase in HC for gasoline operation is greater than that for methane operation. NO_x emissions decreased in many cases for the coated pistons. This is thought to be due to lower compression stroke temperatures as observed in previous testing.

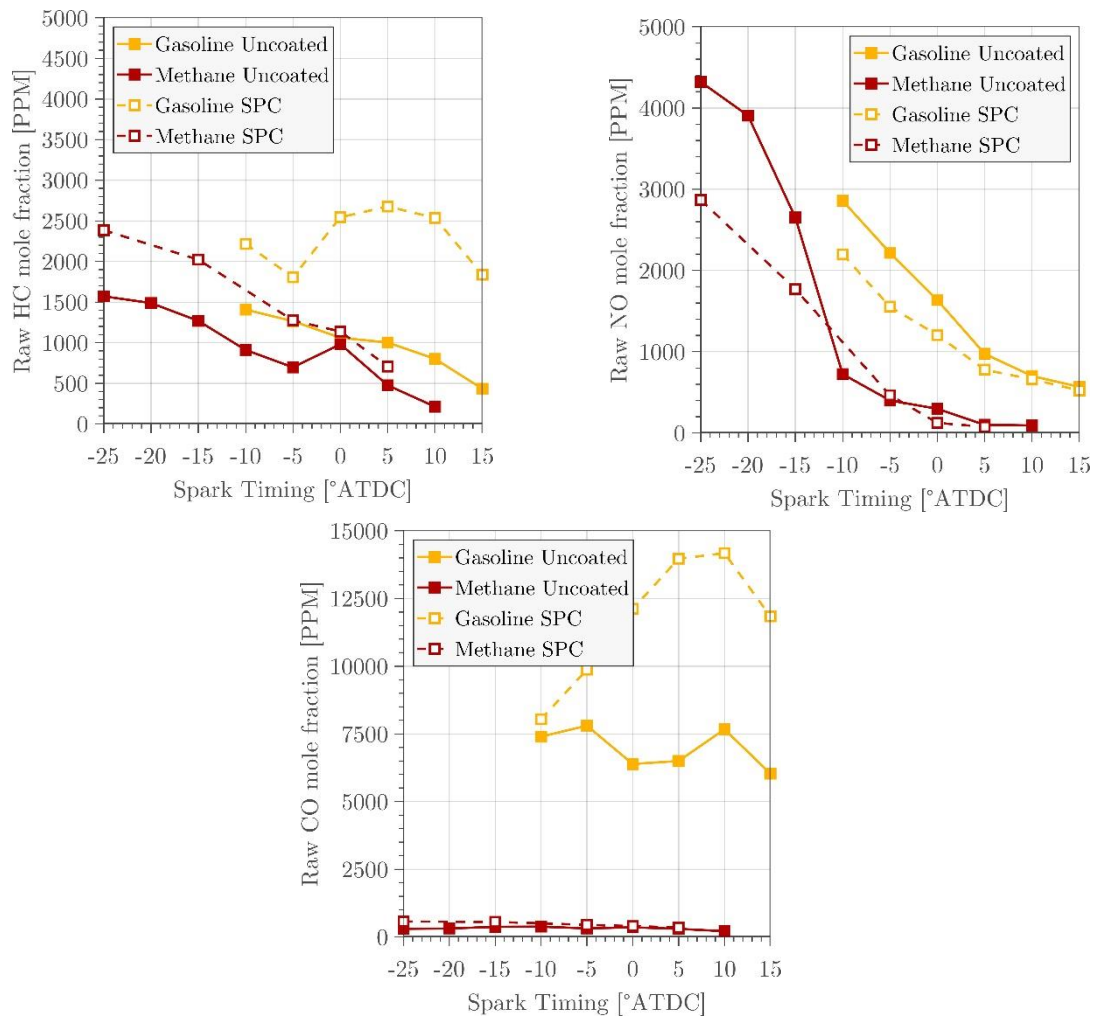


Figure 26.2. Comparison of HC emissions (top left), NO emissions (top right) and CO emissions (bottom) when using gasoline and methane with the uncoated and SPC pistons.

Task 27 Evaluation of Adiabatics Coatings in a Medium Duty Diesel Engine

Summary

The TBCs developed in the GENSETS program showed promising results on a small bore, natural gas-fueled, spark-ignited engine. Substantial market opportunity and impact is possible in the medium- and heavy-duty markets; however, the performance of current coatings has not been verified in a diesel engine environment. To provide pre-competitive engine data, engine experiments with coated pistons were completed. Overall, an improvement in efficiency was observed at the higher load conditions. Minimal impact in emissions was observed. Coated pistons enabled increased exhaust temperature.

Details

The same diesel engine used in Task 26 was used to evaluate the coatings under diesel engine operation. The conditions are named according to the conventions set forth in 40 CFR § 1065.610. All the conditions are chosen to be at the A speed (1500 RPM). Three load points (A2, A6, and A13) are chosen – one each at low, mid, and high loads. These points are shown on the load-speed map of the ISB engine in Figure 27.1; the key details of these points are given in Table 27.1.

Table 27.1. Details of test points chosen for study.

Parameter	Value		
Name	A2	A6	A13
GIMEP [bar]	3	8	15
Intake Pressure [kPa]	130	150	190
Exhaust Pressure [kPa]	175	200	220
Intake Temperature [°C]	50		
Coolant/Oil Temperature [°C]	90		
Engine Speed [RPM]	1500		
Pressure Difference across Engine [kPa]	50		
Fuel Flow Rate [mg/cycle]	23	44	83
Injection Pressure [bar]	950	1300	1500
EGR Rate [%]	20	18	15
Main Injection SOI [dATDC]	Swept		
Duration of Main Injection [ms]	Adjusted to hold fuel flow constant		

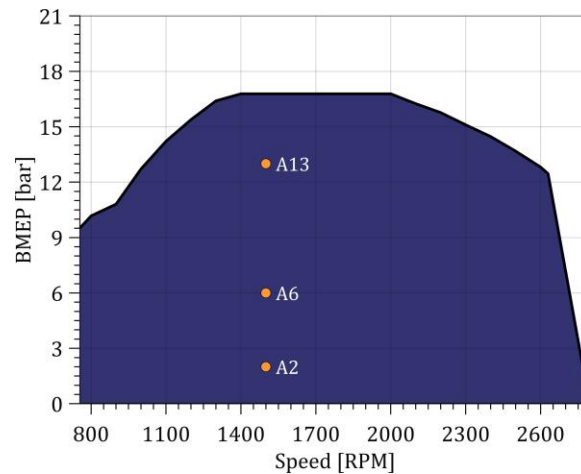


Figure 27.1. Load Points for the TBC Study shown on Cummins ISB operating map.

Injection pressure, exhaust gas recirculation (EGR) rate, air fuel ratio (AFR) and other boundary conditions were chosen to be representative of the values that the stock engine is calibrated to use at the respective load points. At each load point, a Start of Injection (SOI) timing sweep was carried out and the injection schedule was chosen to only have a single injection, to make the results as easy to interpret as possible. The objective of the selection of conditions is to select the widest variety of SOI timings at each load point that makes sure the operation stays within reasonable safety and mechanical limits for the engine and test cell. When advancing SOI timing, the SOI timing at which either the peak cylinder pressure (PCP) reaches 180 bar or fuel specific NO_x emissions (fsNO_x) reaches 60 g/kgfuel was chosen to be the most advanced SOI timing of the sweep. When retarding the SOI timing, either the earliest SOI timing at which the end of injection (EOI) occurs at 30 °ATDC or the earliest SOI timing at which the Coefficient of Variation (COV) of the Gross Indicated Mean Effective Pressure (GIMEP) was greater than 10%, whichever occurs first, was chosen as the most retarded SOI timing. In addition, a soot limit of 3 FSN and a Peak Pressure Rise Rate (PPRR) limit of 30 bar/CAD were also imposed i.e., any point that falls within the advancement and retardation limits but violates the soot or PPRR limits was rejected.

Figure 27.1 shows the comparison of GIE between stock, SPC and YSZ-coated pistons at the A2, A6 and A13 conditions. It is seen that there is no improvement in GIE from the use of the YSZ-coated piston and a 1% improvement in the GIE from the use of the SPC piston at SOI timings prior to 5 °ATDC at the A2 and A6 conditions. At the earliest SOI timing at the A13 condition, the YSZ piston is seen to offer superior efficiency to the stock piston, but this effect is seen to diminish at later SOI timings; the SPC piston offers a consistently higher GIE at the A13 condition also.

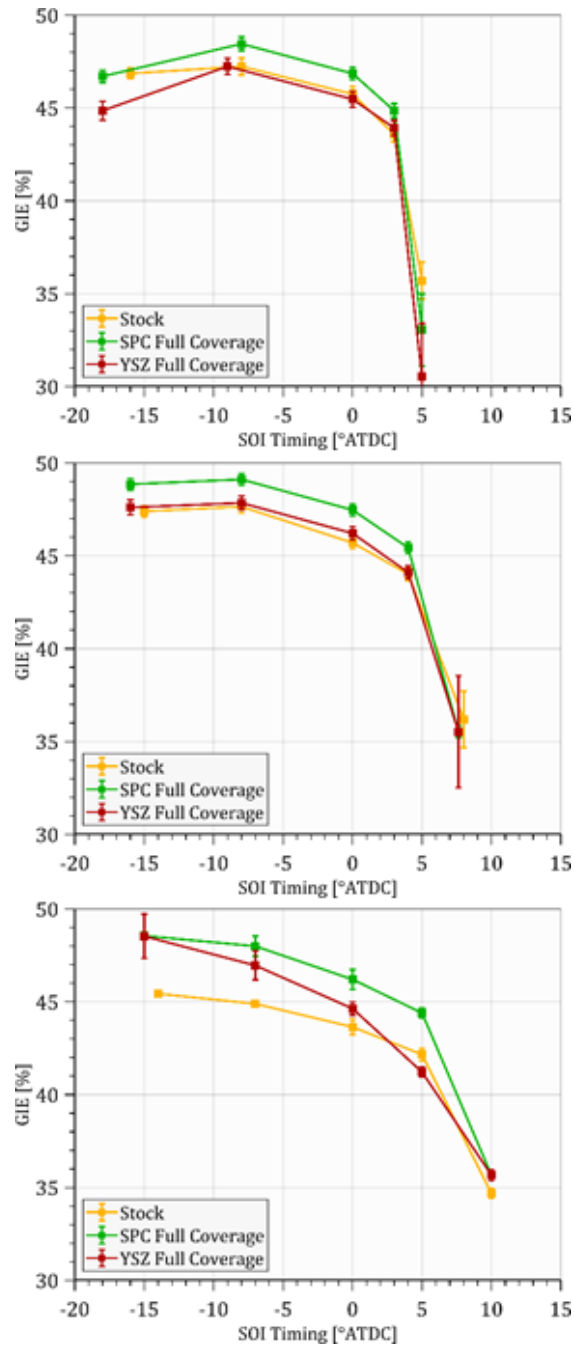


Figure 27.2. Comparison of GIE between stock, SPC and YSZ-coated pistons at the A2 (top), A6 (middle) and A13 (bottom) conditions.

Figure 27.3 shows the comparison of NO_x emissions between stock, SPC and YSZ-coated pistons at the A2, A6 and A13 conditions. The NO_x emissions of the YSZ-coated piston are seen to be the lowest, followed by the SPC and stock pistons; the differences in NO_x are minimal at the A2 and A6 conditions but become more significant at the A13 condition.

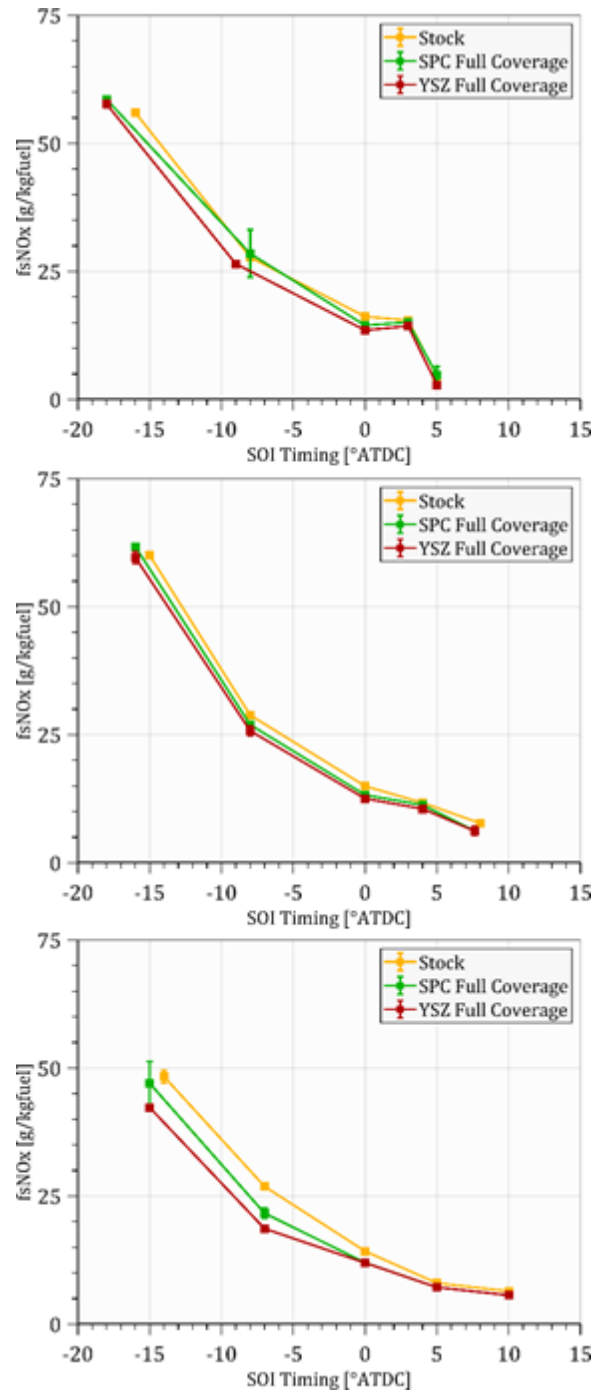


Figure 27.3. Comparison of fsNOx emissions between stock, SPC and YSZ-coated pistons at the A2 (top), A6 (middle) and A13 (bottom) conditions.

Figure 27.4 shows the comparison of exhaust gas temperature (EGT) between stock, SPC and YSZ-coated pistons at the A2, A6 and A13 conditions. The EGT of the stock piston is seen to be lowest, followed by the YSZ and SPC pistons; the differences in EGT at the A2 condition of 150 °C are of particular significance, as this would aid aftertreatment warmup strategies. That is, to enable use of urea-based selective catalytic reduction (SCR) systems, which are employed on nearly every modern diesel engine, exhaust temperatures must be above ~220°C. For the stock piston at the A2 condition, the SOI timing

must at top dead center or later. The coated pistons achieve this temperature at all SOI timings.

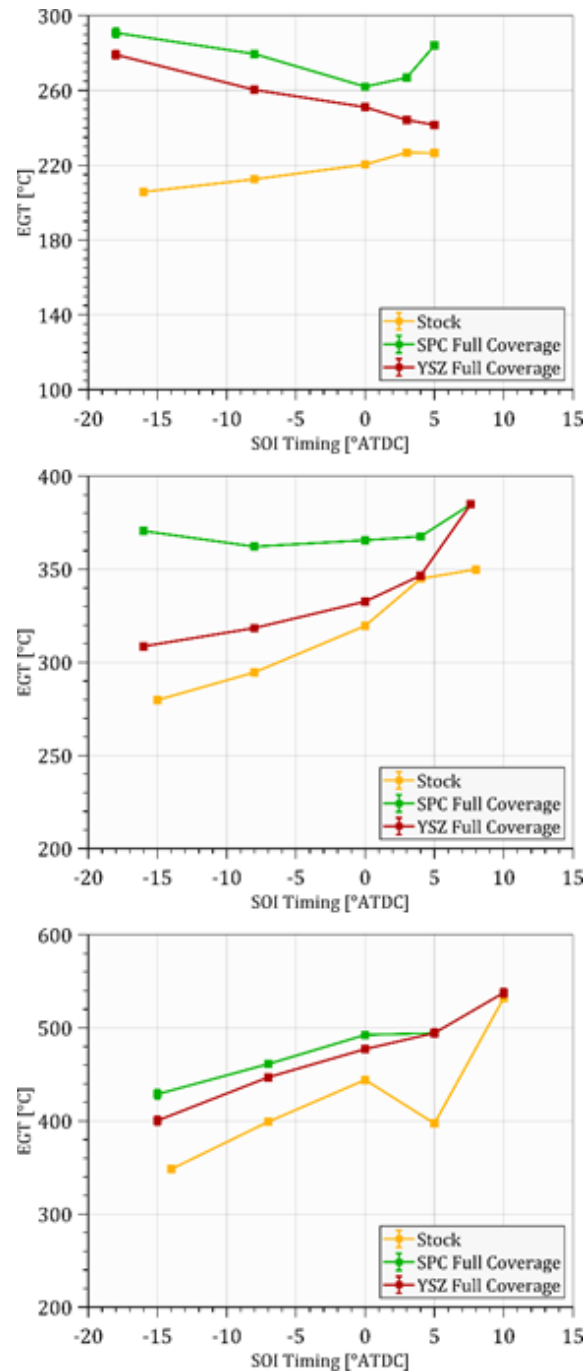


Figure 27.4. Comparison of EGT between stock, SPC and YSZ-coated pistons at the A2 (top), A6 (middle) and A13 (bottom) conditions.

Figure 27.5 shows the comparison of soot emissions between stock, SPC and YSZ-coated pistons at the A2, A6 and A13 conditions. Soot emissions are not seen to be significantly different between the pistons at most conditions studied here.

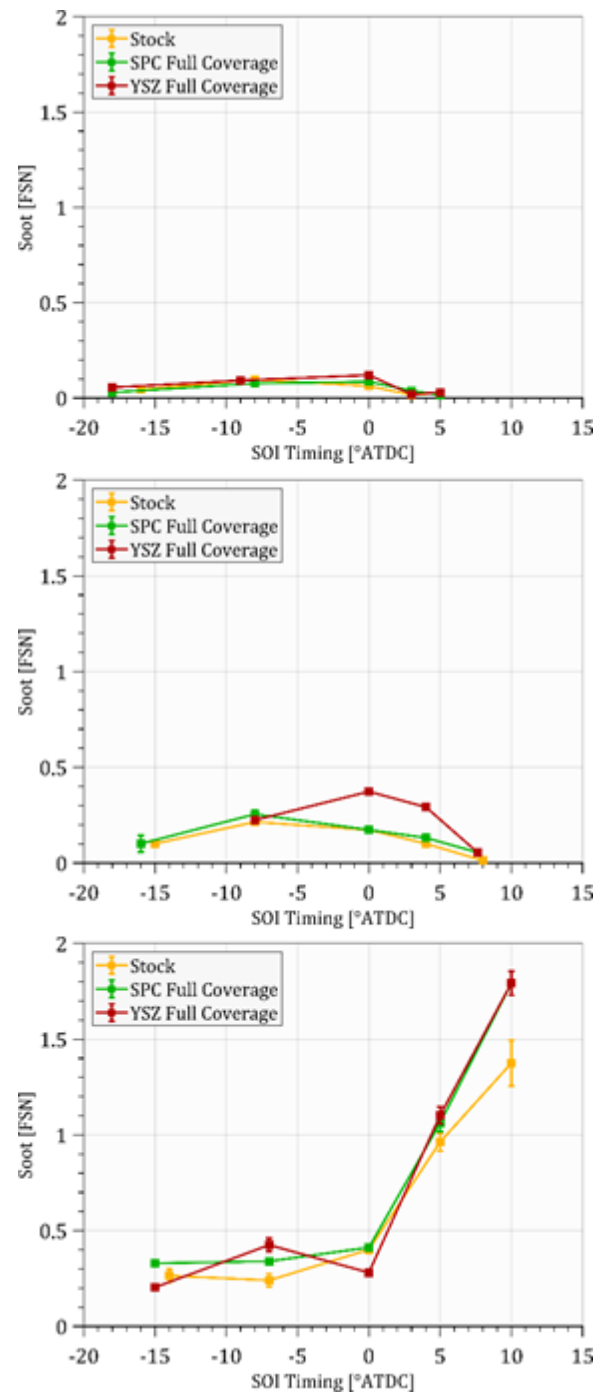


Figure 27.5. Comparison of soot emissions between stock, SPC and YSZ-coated pistons at the A2 (top), A6 (middle) and A13 (bottom) conditions.

Task 28 TBCs for Gasoline Spark-Ignited Engines

Task 28 focused on developing a new TBC for use in a gasoline spark-ignited engine, providing TBC parts for Briggs & Stratton test engine, and testing the new TBC for durability on a spark-ignited gasoline engine.

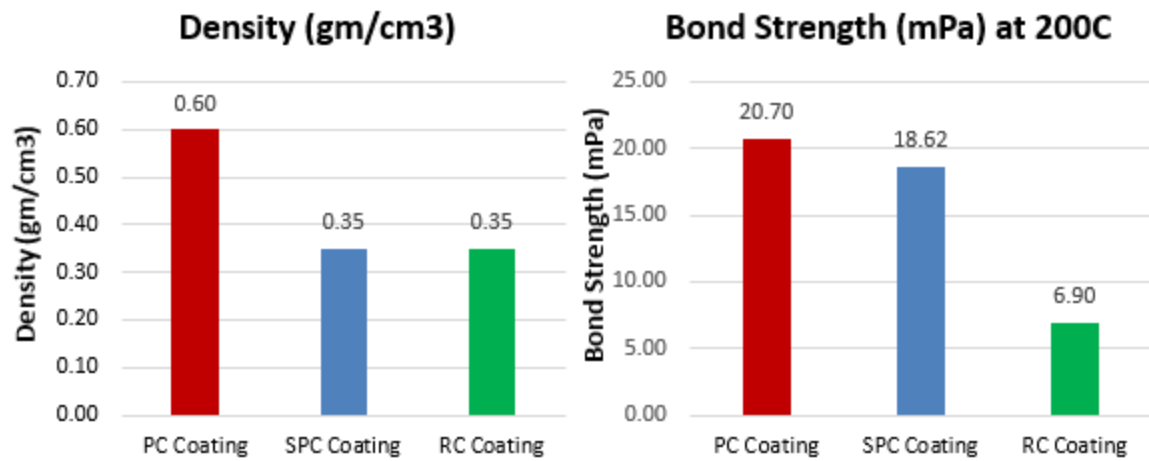
Figure 28.1 shows the timeline for Task 28 which was done in conjunction with Task 26: TBCs for Gasoline Spark-Ignited Engines. Milestone M28.1: Provide TBC Coated Parts for Briggs & Stratton Test Engine(s) was completed at the end of Q1 2020 and Milestone M28.2: Run Durability Test on New TBC on Spark-Ignited Gasoline Engine was completed at the end of Q4 2020. Task 28 was completed at the end of Q4 2020.



Figure 28.1. Timeline of Task 28: TBCs for Gasoline Spark-Ignited Engines.

Results

During Q4 2019, a set of pistons, cylinder heads, and valves were coated with TBCs developed during the Phase I effort in order to test for HC emissions. The SPC TBC was chosen for the cylinder heads and pistons because it possessed the best combination of thermal and durability properties of any of the TBCs that had been developed. It also possessed the smoothest surface profile as its microspheres are sieved into smaller sizes. The RC Coating was chosen for the valves because it possessed good thermal properties and because it had proven to bond more strongly than the other TBCs to substrates with a high nickel/chromium composition and when it can be heated and fully cured at 850 °F. A comparison of TBCs developed during the Phase I effort is shown in Figures 28.2 and 28.3. Once these two sets were coated, their thicknesses were documented (shown in Figure 28.4) and sent back to Briggs & Stratton to undergo HC emissions testing.



Figures 28.2 and 28.3. Comparison of TBC densities and bond strengths.

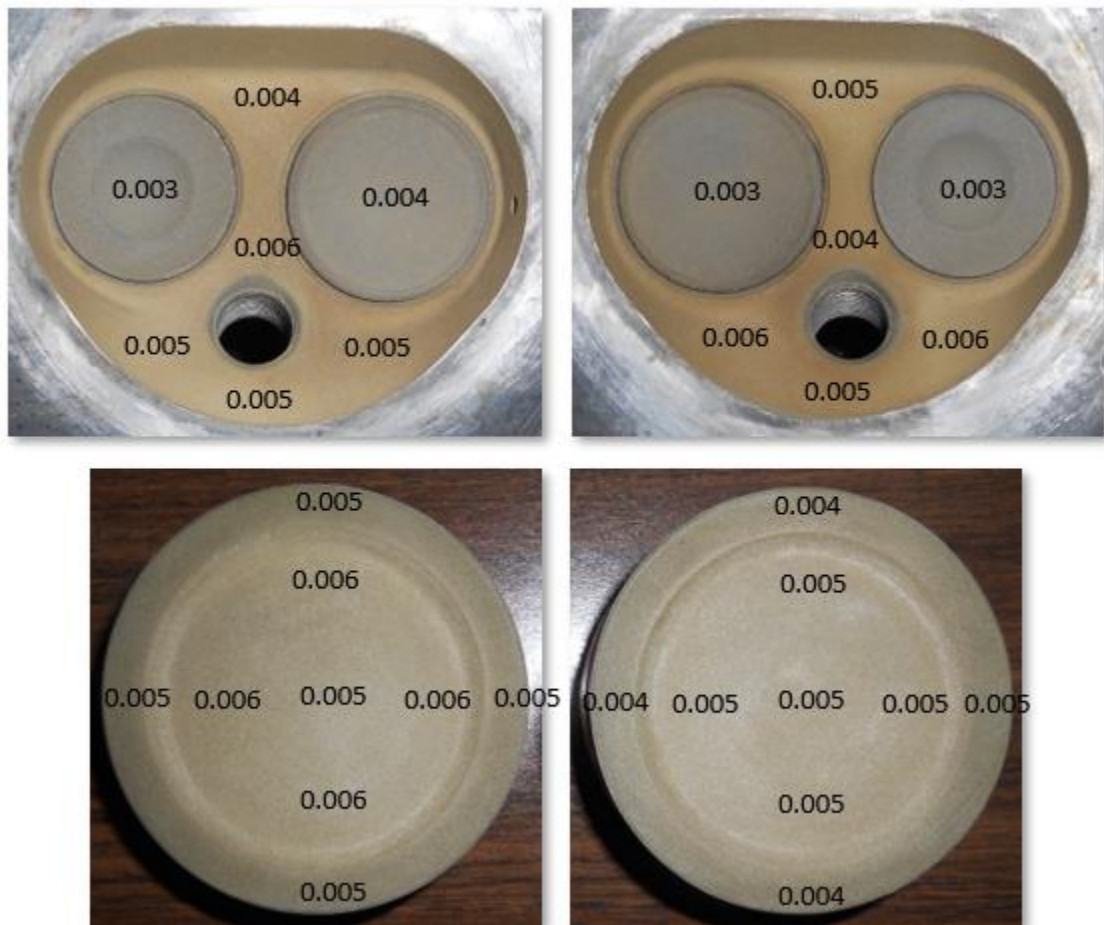


Figure 28.4. Set 1 (Left) and Set 2 (Right) coated with SPC Coating on cylinder heads and pistons, and RC Coating on valves (thicknesses in inches).

HC emissions testing for the parts, along with analysis, was completed by Briggs & Stratton in Q4 2019. While it was theorized that the better density and smoother surface profile of the SPC TBC would produce better HC emissions data, this unfortunately proved to be untrue as the SPC TBC Iteration actually showed slightly higher HC emissions than the PC TBC Iteration for every mode tested except for Mode 7 (10% load). Results are shown in Figure 28.5.

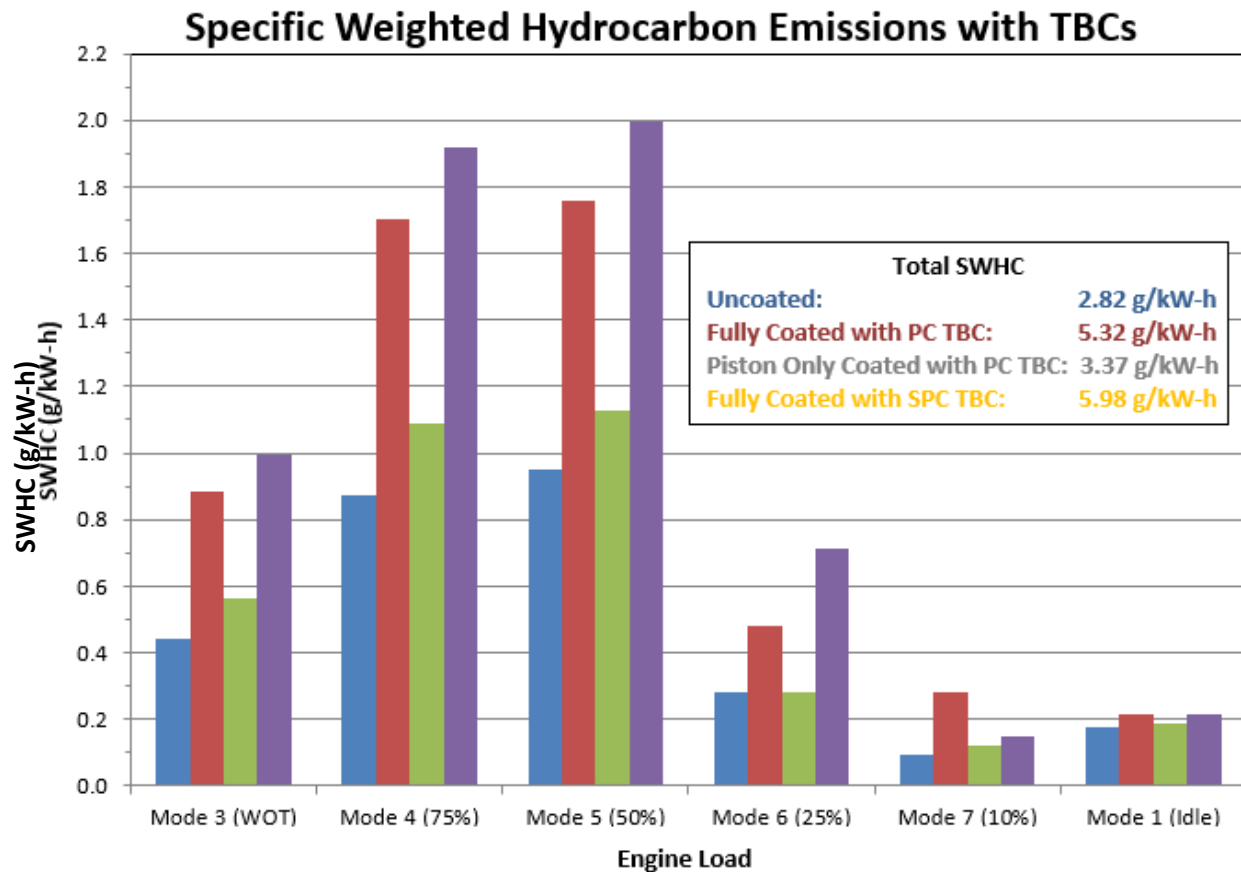


Figure 28.5. HC emissions test analysis conducted by Briggs & Stratton.

Due to the results from Briggs & Stratton's engine testing, multiple variables were identified (thickness, sealing/porosity, surface roughness, and oxidation of HCs in coating pores) as possible causes for the increase in HCs. These variables were investigated through engine testing using a Snap-On 5-gas analyzer to measure HCs as detailed in Task 26. Engine testing these four variables showed that the main contributor to HCs was porosity, so the SPC Coating + Sealer was selected for 100-hour durability testing in which the HCs were measured at both the beginning and the ending of the durability testing.

Before the SPC Coating + Sealer was tested in the gasoline genset, however, thermal shock fatigue testing was performed on the coating in order to help determine whether it would survive in the genset engine. For this reason, a thermal shock rig sample was coated with the SPC Coating + Sealer at 0.004" thickness and run for the 50 cycle/14 second standard test on the oxy-acetylene thermal shock rig. After the test was completed, the SPC Coating + Sealer sample remained in good condition without any

cracking or delamination. Shown in Figure 28.6 is the result of oxy-acetylene thermal shock rig testing for the SPC Coating + Sealer.

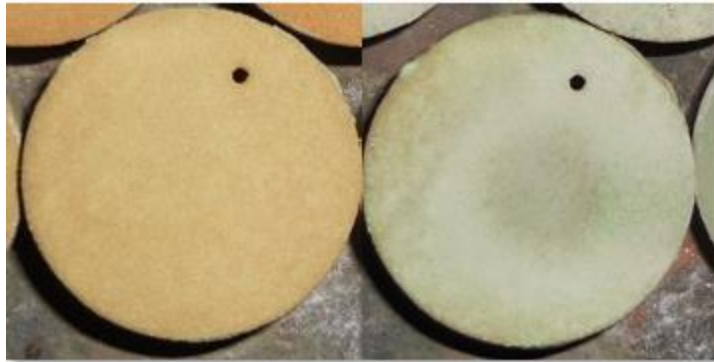


Figure 28.6. Before (Left) and After (Right) 50 cycles of 14 second oxy-acetylene thermal shock rig testing of the SPC Coating + Sealer.

Since the SPC Coating + Sealer Iteration showed the best results during TBC variable testing, it was selected for the 100-hour durability test. For this test, both the pistons and cylinder head faces were coated with the SPC Coating + Sealer at a targeted thickness of 0.004-0.005". Shown in Figures 28.7 and 28.8 are the recorded measurements for the pistons and cylinder heads before the 100-hour durability test.



Figure 28.7. Piston 1 (Left) and Piston 2 (Right) coated with the SPC Coating + Sealer before 100-hour durability testing (measurements in inches).

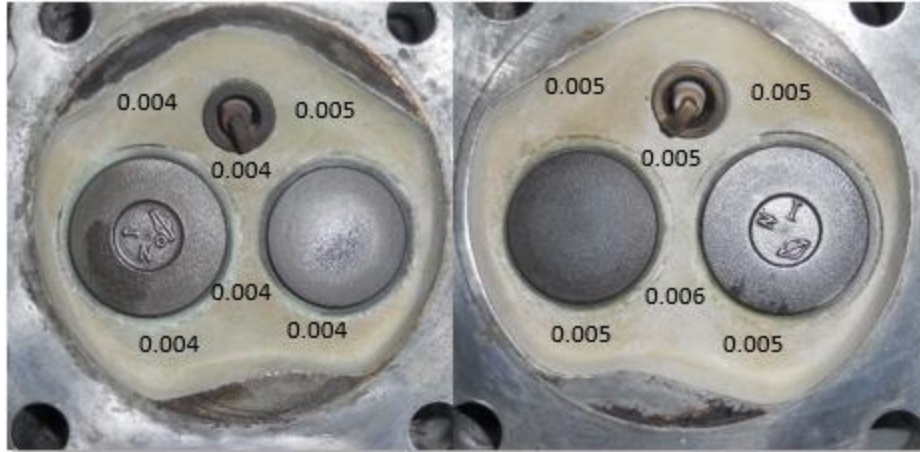


Figure 28.8. Cylinder Head 1 Face (Left) and Cylinder Head 2 Face (Right) coated with the SPC Coating + Sealer before 100-hour durability testing (measurements in inches).

In order to ensure that there were no coating failures during testing, the coated parts were photographed using a borescope at 28 hours and 84 hours. Photographs showed that there wasn't any significant change to the coating at 28 hours as well as at 84 hours. Shown in Figures 28.9 and 28.10 are the borescope pictures taken at these intervals.

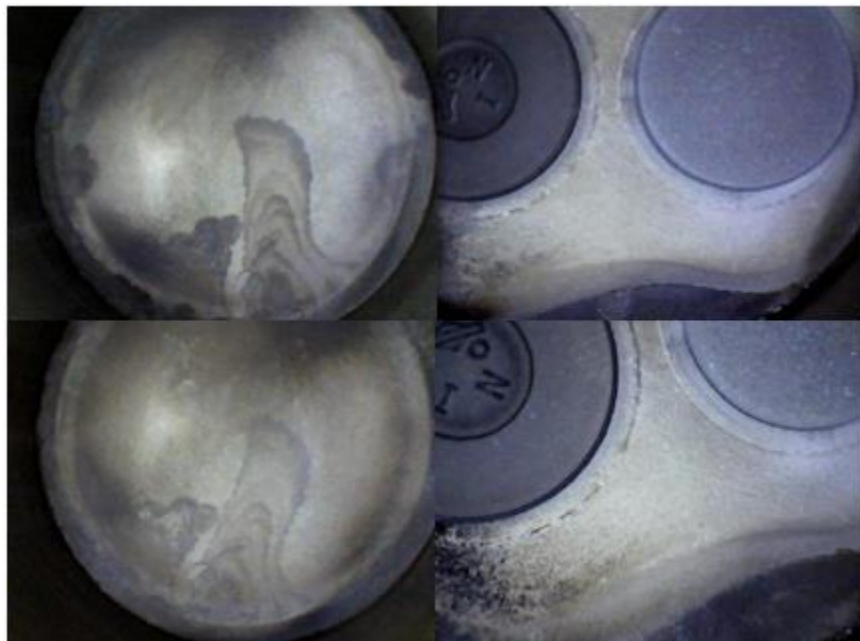


Figure 28.9. Set 1 borescope pictures after 28 hours (top) and after 83 hours (bottom). Coating remains in good condition with no signs of delamination.



Figure 28.10. Set 2 borescope pictures after 28 hours (top) and after 83 hours (bottom). Coating remains in good condition with no signs of delamination.

After the 100 hours of durability testing was completed, the pistons and cylinder heads were removed, inspected, and the thicknesses were recorded. TBCs on both sets of cylinder heads and pistons remained in good condition after the 100 hours were completed. Shown below in Figures 28.11 and 28.12 are the recorded measurements after the 100-hour durability test.



Figure 28.11. Set 1 (SPC Coating + Sealer) after 100 hours of durability testing (measurements in inches).

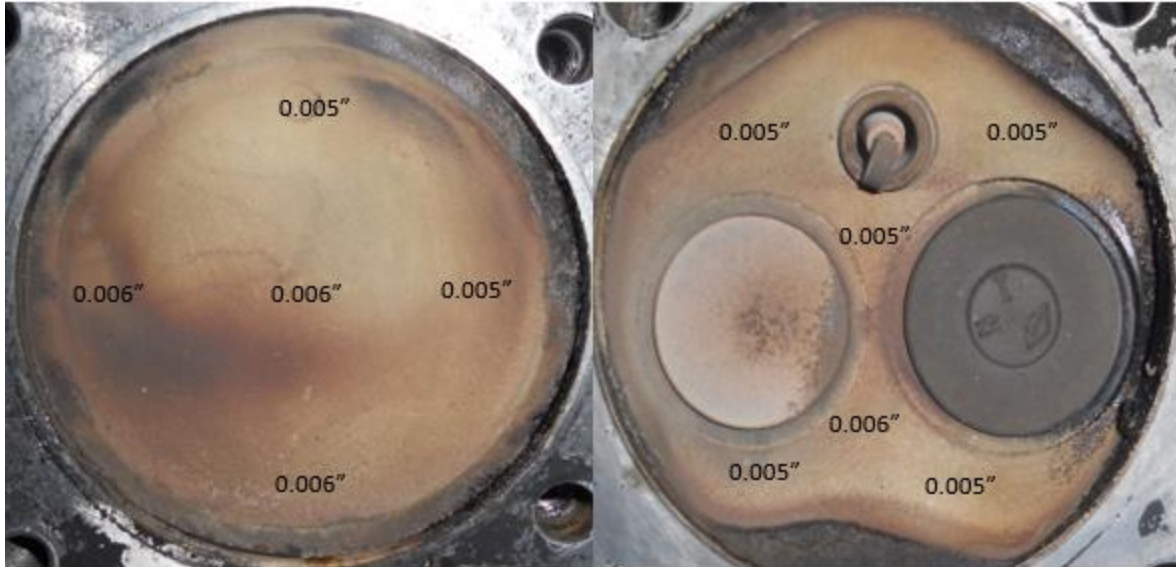


Figure 28.12. Set 2 (SPC Coating + Sealer) after 100 hours of durability testing (measurements in inches).

After the 100 hours of durability testing was completed, emissions data for the SPC Coating + Sealer on both the heads and pistons was collected. Afterwards, the coating was removed from the heads and pistons and run again in order to establish a comparative baseline. Shown in Figure 28.13 are the results from the emissions testing. Similar to previous data, the emissions from the SPC + Sealer Iteration in comparison to the Baseline Iteration showed a gap that continually decreased from the normal (rich) setting to the mid setting to the lean setting.

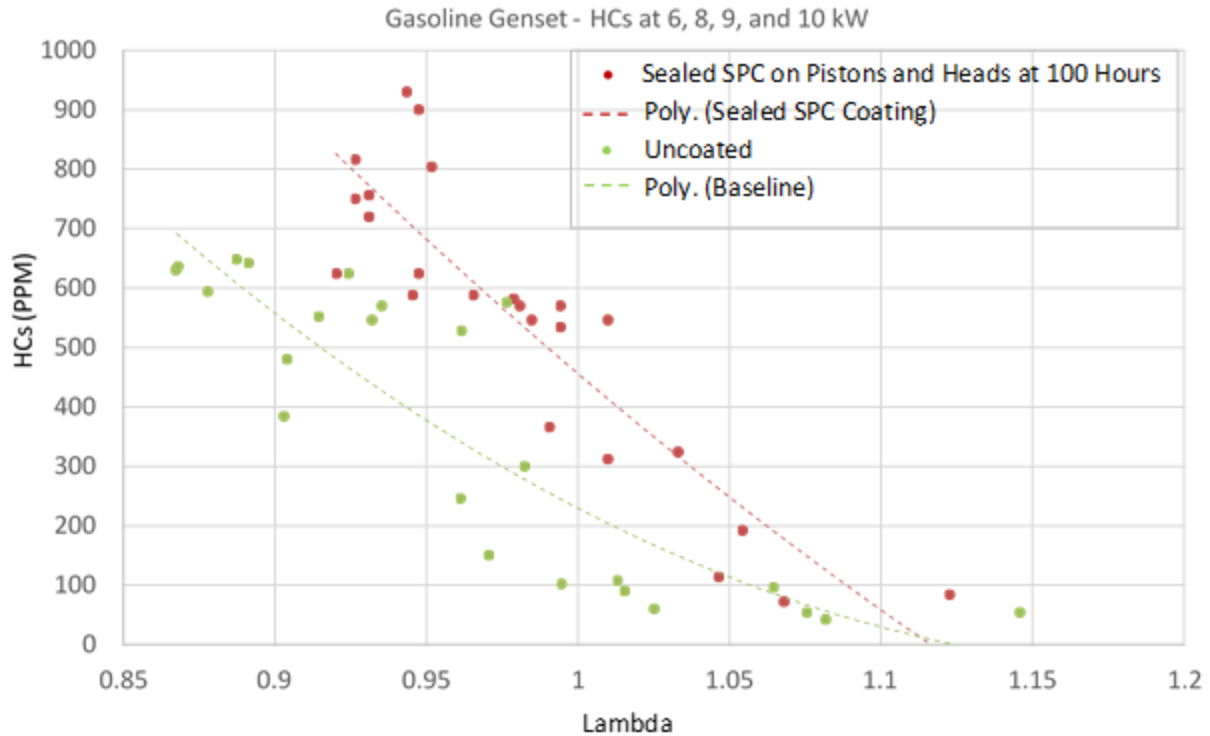


Figure 28.13. Emissions data for the SPC Coating + Sealer on both the pistons and heads in comparison to an uncoated baseline.

Conclusions

Milestone M28.1: Provide TBC-Coated Parts for Briggs & Stratton Test Engine(s) was completed at the end of Q1 2020. Results from testing showed that the SPC Coating Iteration exhibited slightly higher HC emissions than the previous PC Coating Iteration that was tested by Briggs & Stratton. For this reason, four variables (thickness, sealer/porosity, surface roughness, and oxidation of HC in pores) were identified and then investigated in Task 26. Results from Task 26 engine testing showed that porosity of the coating was the main contributor to the increase in HCs, so the SPC Coating + Sealer, which produced improved HC results over the SPC Coating in Task 26, was chosen for 100-hour durability engine testing.

Milestone M28.2: Run Durability Test on New TBC on Spark-Ignited Gasoline Engine was completed at the end of Q4 2020. Before the SPC Coating + Sealer Iteration underwent 100-hour durability engine testing in the 10kW Briggs & Stratton genset, the coating was laboratory-tested for durability using the thermal shock fatigue rig. During this test, the coating underwent the standard 50 cycles at 14 seconds/cycle through an oxy-acetylene torch flame for which it passed. Therefore, the SPC Coating + Sealer was then installed into the genset and subjected to 100-hours of durability testing. After the 100 hours were finished, a HC lambda sweep using a Snap-On 5-gas analyzer was completed which showed similar results to previous data. During the testing, the emissions from the SPC + Sealer Iteration showed a gap that continually decreased from the normal (rich) setting to the mid setting to the lean setting in comparison to the Baseline Iteration. After the 100-hour durability testing and HC testing were completed, the engine was disassembled and the coated parts were examined, which showed that the

TBCs on both sets of cylinder heads and pistons remained in good condition after the 100 hours was completed.

Task 29 TBCs for Diesel Engines

Summary

Task 29 focused on furnishing plasma-sprayed partially stabilized zirconia (PSZ) and new TBCs to the Univ. of Wisconsin for testing as well as assisting in the analysis of the engine tests.

Figure 29.1 shows the timeline for Task 29: TBCs for Diesel Engines. The only milestone for this task was Milestone M29: Furnish Plasma-Sprayed PSZ and New TBC to Univ. of Wisconsin for Testing, so all efforts were directed toward this task. Task 29 was completed at the end of Q4 2020.

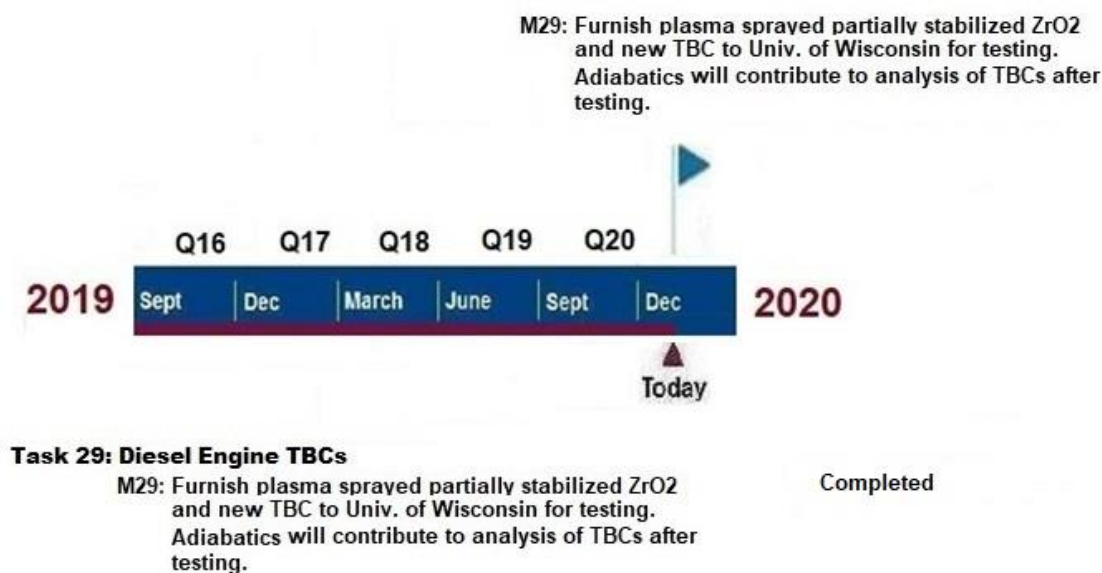


Figure 29.1. Timeline of Task 29: TBCs for Diesel Engines.

Results

The current state-of-the-art industry standard coating for diesel engines is PSZ. Therefore, it provides a good reference comparison for Adiabatics' TBCs. During Q4 2019, Adiabatics had two pistons machined down and coated with PSZ, one piston with the entire profile coated at approximately 0.010 inch, and one piston with only the squish area coated at a target 0.010 inch. The reason for coating the different profiles was to examine the effects that surface roughness has on combustion. The TBC and PSZ coatings are relatively smooth, but a machined and polished surface is significantly smoother. Therefore, by coating the different profiles, it would be determined whether smoothness of the surface has a significant effect on combustion, and whether it leads to an increase in HCs. The two PSZ-coated pistons are shown in Figure 29.2.



Figure 29.2. Piston coated with PSZ on entire profile (Left) and piston coated with PSZ on squish area only (Right).

Originally, it was agreed upon that the PSZ coatings would be tested “as-sprayed” as it would give the most accurate representation of how PSZ affects engine performance. However, in Q1 2020, it was decided that the pistons’ sprayed thicknesses were slightly greater than desired (0.013 inch instead of 0.010 inch), so Adiabatics machined the pistons down to 0.010 inch, the desired thickness for testing, documented, and sent back to the Univ. of Wisconsin for engine performance testing. The two PSZ-coated pistons that were machined down to the correct thickness are shown in Figure 29.3.

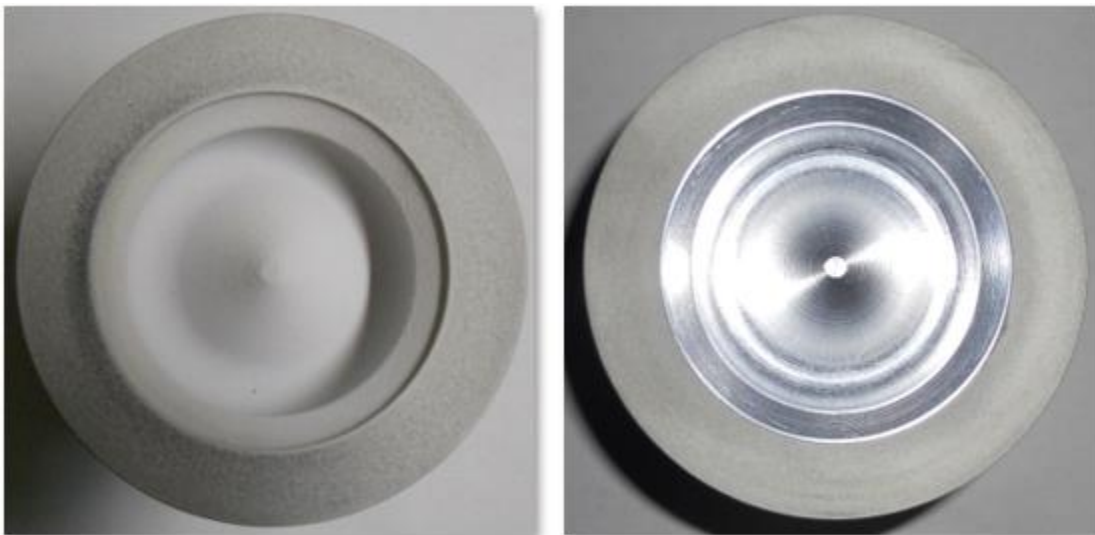


Figure 29.3. Piston coated with PSZ on entire profile (Left) and piston coated with PSZ on squish area only (Right).

In Q3 2020, Adiabatics coated an identical set of pistons with the SPC Coating at a thickness of 0.006". The pistons' measurements were documented and sent to UW for diesel engine testing. Shown in Figure 29.4 are the Adiabatics SPC-coated pistons.

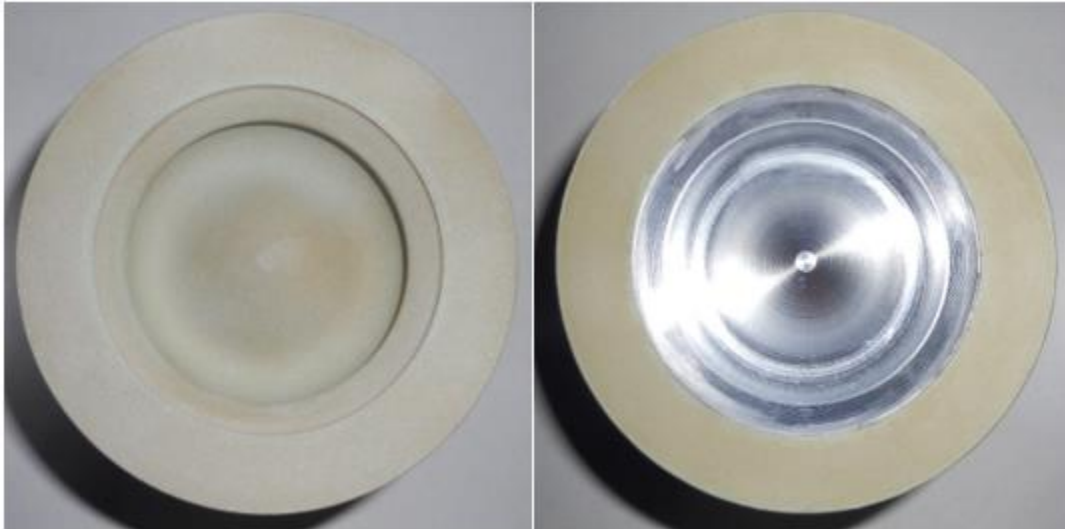


Figure 29.4. Piston coated with SPC Coating on entire profile (Left) and piston coated with SPC Coating on squish area only (Right).

Conclusions

Milestone M29: Furnish Plasma-Sprayed PSZ and New TBC to Univ. of Wisconsin for Testing was the only milestone for this task and was completed in Q4 2020. During this task, Adiabatics coated four pistons: 1) PSZ applied at 0.010 inches on entire profile, 2) PSZ applied at 0.010 inches on squish area only, 3) SPC Coating applied at 0.006 inches on entire profile, and 4) SPC Coating applied at 0.006 inches on squish area only. After the coatings were complete, Adiabatics sent the pistons to UW for diesel engine testing and agreed to assist with any analysis requested.

Project Activities

A transformative small spark ignition (SI) internal combustion (IC) engine fueled with natural gas was developed for combined heat and power (CHP) applications using a combination of cycle simulations, computational fluid dynamics (CFD) modeling, and engine experiments. The resulting 1 kW CHP engine was demonstrated to achieve 36.1% brake thermal efficiency (BTE) while meeting the aggressive exhaust emissions targets using a low-cost three-way catalyst.

Project Outputs

A. Journal Articles

B. Papers

Andrie, M., Kokjohn, S., Paliwal, S., Kamo, L. et al., "Low Heat Capacitance Thermal Barrier Coatings for Internal Combustion Engines," SAE Technical Paper 2019-01-0228, 2019, <https://doi.org/10.4271/2019-01-0228>.

Babu, A., Koutsakis, G., Andrie, M., Kokjohn, S.L., "Experimental and Analytical Study of Temperature Swing Piston Coatings in a Medium-Duty Diesel Engine", Submitted SAE World Congress, 2022.

C. Status Reports

D. Media Reports

E. Invention Disclosures

All disclosures entered into iEdison

"Engine with Compact Valvetrain," David Procknow, Jason Raasch, Jonathon Tyznik, 5/12/2018

"Engine Insulating Coating," Michael Andrie, Lloyd Kamo, Alex Kamo, 7/16/2018

F. Patent Applications/Issued Patents

Utility Patent Application: "THERMAL BARRIER COATINGS CONTAINING ALUMINOSILICATE PARTICLES," Applicant: Wisconsin Alumni Research Foundation, et al., Application Number: 16/036,364, Date: 7/16/2018, Inventors: Michael Andrie, Lloyd Kamo, Alex Kamo

G. Licensed Technologies

H. Networks/Collaborations Fostered

The effort grew the relationship between Wisconsin Engine Research Consultants and project partners, Adiabatics Inc., Briggs and Stratton Corp., and University of Wisconsin - Madison

I. Websites Featuring Project Work Results

J. Other Products (e.g. Databases, Physical Collections, Audio/Video, Software, Models, Educational Aids or Curricula, Equipment or Instruments)

Paliwal, S. "Analysis of a Residential Combined Heat and Power System", MS Thesis in Mechanical Engineering, UW-Madison, 2019.

K. Awards, Prizes, and Recognition

Follow-On Funding

No follow on-funding at the present time.