Final Scientific/Technical Report

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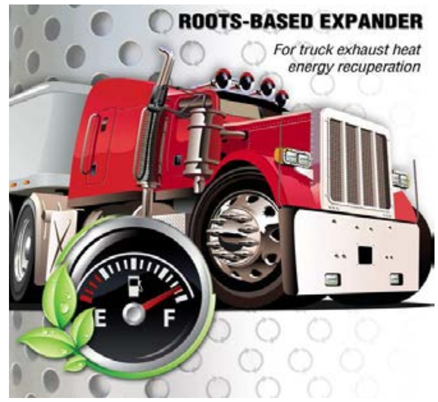

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Recipient Organization: Eaton Corporation
W126N7250 Flint Drive
Menomonee Falls, WI 53051
DUNS: 360921261

Project Principal Investigator and Technical Contact: Swami Subramanian Ph.D.
Office: (248) 226-1754
Mobile: (630) 956-8225
swaminathan@eaton.com

Contractual Contact: Cindy K. Shane, Manager Government Contracts
Eaton Corporation
(414) 449-6607, Lucindakshane@eaton.com

Partners:

AVL Powertrain Engineering, Inc.

Deere & Company

Modine

Electricore, Inc.

Report Submitted For: Energy Efficiency & Renewable Energy (EERE) DOE / National Energy Technology Laboratory (NETL)

DOE Project Officer: Ralph Nine, (304) 285-2017, ralph.nine@netl.doe.gov

DOE Contract Specialist: Bethan Young, (412) 386-4402, bethan.young@netl.doe.gov

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1.0 Executive Summary

Eaton Corporation proposed a comprehensive project to develop and demonstrate advanced component technology that will reduce the cost of implementing Organic Rankine Cycle (ORC) Waste Heat Recovery (WHR) systems to Heavy-Duty Diesel engines, making adaptation of this fuel efficiency improving technology more commercially attractive to end-users in the next 5 to 10 year time period. Accelerated adaptation and implementation of new fuel efficiency technology into service is critical for reduction of fuel used in the commercial vehicle segment.

The Roots expander technology was demonstrated and validated through engine dynamometer testing using an engine in the appropriate displacement, power and technology class representative of many heavy-duty commercial applications matched to appropriately sized heat exchangers and tested over USEPA speed and load conditions.
Specific project objectives included:

Objective

- The HD-REHER project had the primary objective of accelerating the development of enabling technologies for commercial implementation of cost effective Waste Heat Recovery (WHR) expander / work extraction component, with system, sub-system and component level demonstration for the recovery and utilization of energy remaining in the exhaust gas of a heavy-duty diesel engine to achieve at least 5% improvement in fuel economy and reduction in greenhouse gas emissions while maintaining or improving the engine out Mono-Nitrogen Oxides (NOx), Particulate Matter (PM), Carbon Monoxide (CO) and Hydrocarbon (HC) emission levels.

Secondary Objective

- Demonstrate a plan for cost reduction by incorporation of a Roots type expander.

These objectives directly support the DOE's goals and objectives for Subtopic 6B. The successful completion of this proposed project will result in prototype system design and validated prototype hardware for an Organic Rankine Cycle Waste Heat Recovery system incorporating Roots type expander technology.

2.0 Accomplishments

Task 1.0 Project Management, Planning, and Reporting

The project team held weekly project management and planning teleconferences; weekly meeting minutes were distributed to the team members.

On May 22, 2012, the project team attended and presented at the DOE Kickoff Meeting in Washington, DC. May 16, 2013 and June 20, 2014, the project team attended and presented at the DOE Vehicle Technologies Program Annual Merit Review in Washington, DC.

Quarterly reports were submitted on time and presented to DOE key personnel. All the tasks shown in Project Management Plan (PMP) have been successfully completed on schedule.

Task 2.0 Baseline Engine Characterization

The overall objectives of Task 2 were to determine the baseline engine EGR tolerance, sensitivity to exhaust back pressure, and characterize the energy availability in the EGR cooling loop, downstream of the turbocharger, charge air cooler and engine coolant at each of the 13 speed/load combinations of an EPA SET cycle. A MY2012 production John Deere 13.5L Tier4i engine (Figure 1) was instrumented and installed in an AVL performance and emissions test cell at the AVL Plymouth facility. Engine specifications are given in Table 1. One cylinder of this engine (Cylinder 6) was instrumented for cylinder pressure and injector driving current measurements via AVL's INDICOM high speed data measurement system. The test system was configured to acquire the engine speed and torque, flow rates, pressures, temperatures, engine controller, and emission related measurements needed to satisfy Task 2.0. The task was completed per the schedule.

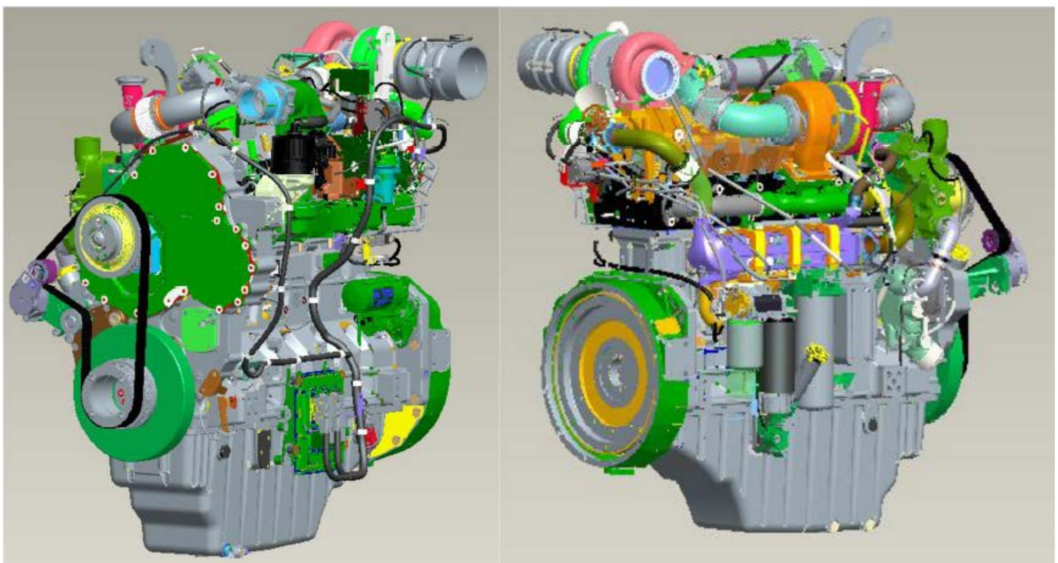


Figure 1: Engineering Renderings of the John Deere 13.5L Engine

Table 1: Engine Specifications

Properties	Value	Unit
Commercial Engine Name	JD 13.5L IT4 6135HFC95	-
Engine Serial Number	RG6135R001878	
Engine Type	4-Stroke	-
Combustion System	Diesel	-
Charging System	2 stage LP fixed, HP VGT	-
Fuel Injection System	Electronic Unit injector	-
Valve Configuration	2 Intake – 2 Exhaust	-
Engine Configuration	I-6	-
Displacement	13.5	L
Bore	132	mm
Stroke	165	mm
Geometric Compression Ratio	16.5:1	-
Connecting Rod Length	265	mm
Piston Pin Offset	0	mm
Rated Power	448	kW
Rated Speed	2100	rpm
Peak Torque	2660	Nm
Peak Torque Speed	1500	rpm
Fuel	ULSD pump fuel	-
Exhaust Backpressure at Rated Power	15	kPa
Intake Restriction at Rated Power	5	kPa

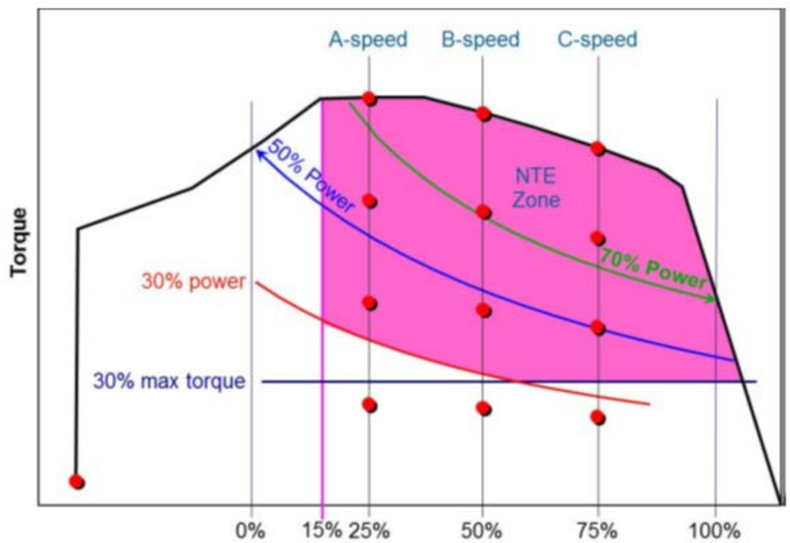


Figure 2: 13 Mode Engine Out SET Points

Table 2: 13 Mode Engine Out SET Points and Corresponding Working Fluids

Mode	Speed	Load	Speed (rpm)	Torque (N-m)	SET WF	In Use WF
1	idle	Idle	802	*	15%	0%
2	A	100%	1307	2595	8%	10%
3	B	50%	1609	1332	10%	10%
4	B	75%	1609	1997	10%	20%
5	A	50%	1307	1297	5%	20%
6	A	75%	1307	1945	5%	20%
7	A	25%	1307	650	5%	0%
8	B	100%	1609	2663	9%	10%
9	B	25%	1609	666	10%	0%
10	C	100%	1910	2339	8%	10%
11	C	25%	1910	588	5%	0%
12	C	75%	1910	1760	5%	0%
13	C	50%	1910	1173	5%	0%

An investigation into the Waste Heat Recovery (WHR) potential from the engine system consisting of the following test configurations was then performed:

- Heat rejection, performance and emission testing over the 13 mode Supplementary Emissions Test (Figure 2 and Table 2) points and full load curve using the baseline engine calibration for exhaust gas recirculation (EGR), injection event and air-fuel ratio (AFR).
- EGR / injection timing / AFR sweeps at each of the 13 mode SET points as feasible within engine system constraints to yield increased EGR heat rejection with little or no penalty to engine efficiency.
- Heat rejection, performance and emission testing over the 13 mode SET cycle and full load curve employing the increased EGR rates.
- Heat rejection, performance and emission testing over the 13 mode points (Figure 3) and full load curve with an increased engine back pressure to infer performance sensitivity to exhaust restrictions from use of a post turbine WHR heat exchanger.

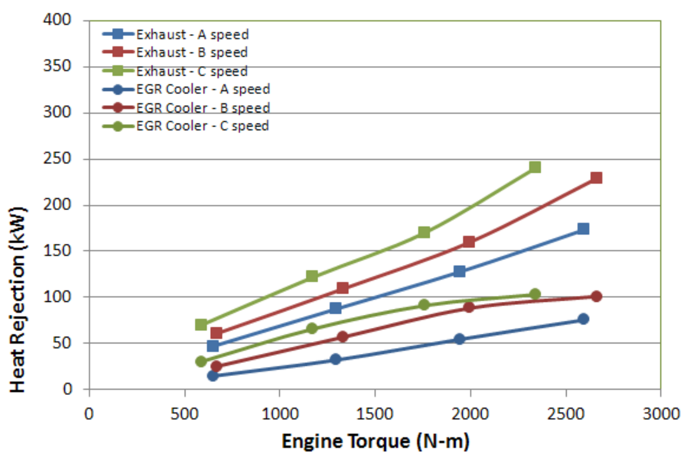


Figure 3: Baseline EGR Heat Rejection and Exhaust Enthalpy Flow

A comparison of the 13 mode engine out SET results using the baseline, elevated EGR rate (‘Optimized’), and elevated back pressure conditions was completed. Averaged values obtained from the optimized run with high EGR rates are shown in Table 3. The duty-cycle weighted heat rejection levels shown in Table 4 were derived by using the estimated ‘line-haul’ duty cycle weighting factors outlined in Table 2. Using the current optimized calibration resulted in attaining close to a 5% improvement for EGR only energy recovery. Data from these tests were used to calibrate a thermodynamic model (AVL BOOST) to perform further system analysis to provide design inputs for the WHR system.

Table 3: Comparison of the 13 Mode Engine Out SET Results

	13 Mode Engine Out SET Values (g/kWh)				Average* Q_{rej}(kW)	
Configuration	NOx	HC	CO	PM	EGR	Exhaust
Baseline w/std restriction	1.428	0.114	0.426	0.181	79.0	173.3
EGR Optimized w/std restriction	1.249	0.117	0.618	0.240	83.1	173.1

*Average based on working estimate drive cycle weighting values, wtd. cycle power = 350.6 kW

Table 4: Duty Cycle Heat Rejection and BSFC Projections

	EGR Cooler		Exhaust
Configuration	Wtd Qrej (kW)	Wtd BSFC (%)	Wtd Qrej (kW)
Baseline w/std restriction	79.0	4.5	173.3
EGR Optimized w/std restriction	83.1	4.8	173.1

Task 3.0 Energy Flow, Trade-Off Analysis and Concept Definition

Energy flow analysis from modeling was completed based on experimental data. Modeling provided the data necessary to size the Rankine Cycle hardware. A 1D thermodynamic model of the engine was completed by AVL. The model was used to maximize the effectiveness of the Rankine cycle waste heat energy recovery. Results are shown in Figure 4.

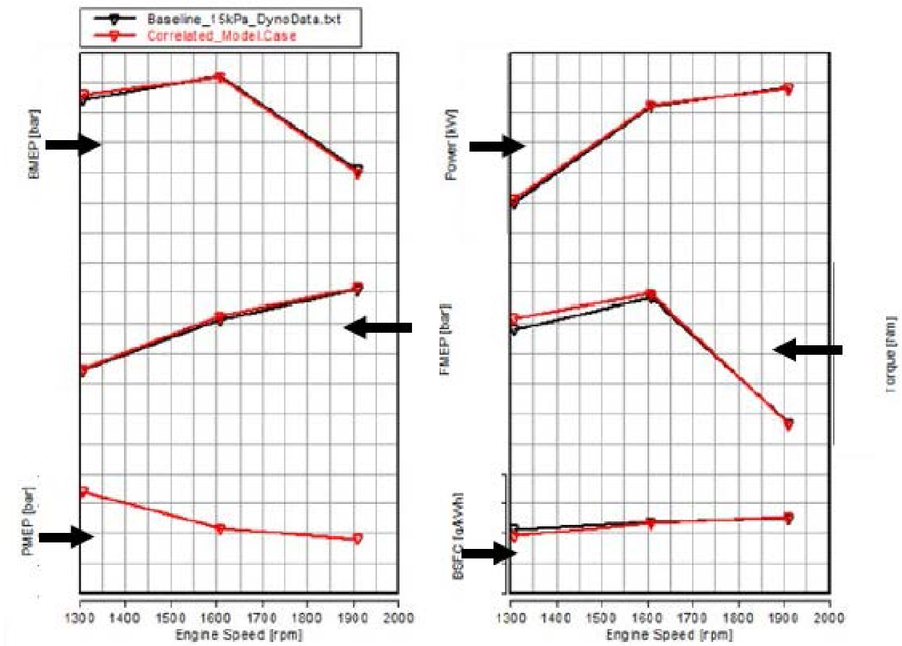


Figure 4: Comparison of Modeled to Measured Engine Performance Parameters

Table 5: Energy and Exergy Summary

Load points	A25	A50	A75	A100	B25	B50	B75	B100	C25	C50	C75	C100
Engine speed [rpm]	1307	1307	1307	1307	1609	1609	1609	1609	1910	1910	1910	1910
Brake power [bkW]	89	178	266	355	112	224	337	449	118	235	352	468
BMEP [bar]	6.0	12.0	18.1	24.1	6.2	12.4	18.5	24.7	5.5	10.9	16.3	21.7
Engine torque [Nm]	650	1297	1947	2595	666	1332	1997	2663	588	1174	1760	2339
Parameters affecting waste heat conditions												
EGR rate [%]	33.5	27.4	26.1	25.1	37.0	33.4	31.5	25.9	39.9	36.7	33.8	28.2
EGR rates were increased from production calibration to help WHR potential												
EGR cooler heat rejection/Fuel Energy [%]	9.8	9.0	9.9	9.9	12.0	12.6	12.8	10.6	13.6	14.1	13.2	11.6
Net exhaust energy/Fuel energy [%]	21.5	21.7	21.3	21.5	21.1	20.7	20.5	22.0	21.4	20.7	20.5	22.1
Exergy (available energy) of EGR before EGR cooler and exhaust after turbine												
Exergy in EGR before EGR cooler [kW]	9	18	31	43	15	33	53	61	20	43	61	71
Exergy in exhaust after turbine [kW]	13	28	42	59	17	34	51	77	18	35	52	81
Total exergy available for WHR (kW)	22	46	73	102	32	67	104	138	38	78	113	152

Table 5 shows the engine energy balance analysis based on the experimental engine test data. It is found that combined Exergy levels in the 12 load points listed vary from 22 kW (A25) to 152 kW (C100). This shows that the waste heat has high exergy levels for most load points. It was identified that the exergy level of EGR is much higher than that of the exhaust after the turbine.

The baseline engine test data was used for correlation of the simulation model. Figure 4 shows the simulation model closely predicts the measured engine performance parameters data within approximately 3%.

Working Fluid Selection

The efficiency of the WHR Rankine cycle is dependent on the thermodynamic properties of the working fluid. R245fa ($C_3H_3F_5$), water and ethanol were selected as three candidates from more than 30 working fluids reported in a literature review. Water was eliminated from the candidate list due to having a low condensation pressure (0.3 bar) leading to sealing concerns and a high freezing point. Ethanol has advantages over R245fa in reduced system pressure, volumetric flow rate through the expander and size of the recuperator. Higher system pressure has significant impact to the cost of the WHR system for mobile applications because the working fluid pump and the heat exchanger's costs increase. Higher volumetric flow rates and comparatively lower pressure are desirable from a system perspective. The above mentioned considerations led to the selection of ethanol as a working fluid.

Organic Rankine Cycle Architecture Study

Three concept Organic Rankine cycle WHR system architectures were defined for evaluation. The first architect is a simple ORC system for which heat is extracted by cooling the recirculated exhaust gas and no recuperation is used. The second system is the same as the first but with recuperation added. Figure 5 shows the first and second architectures.

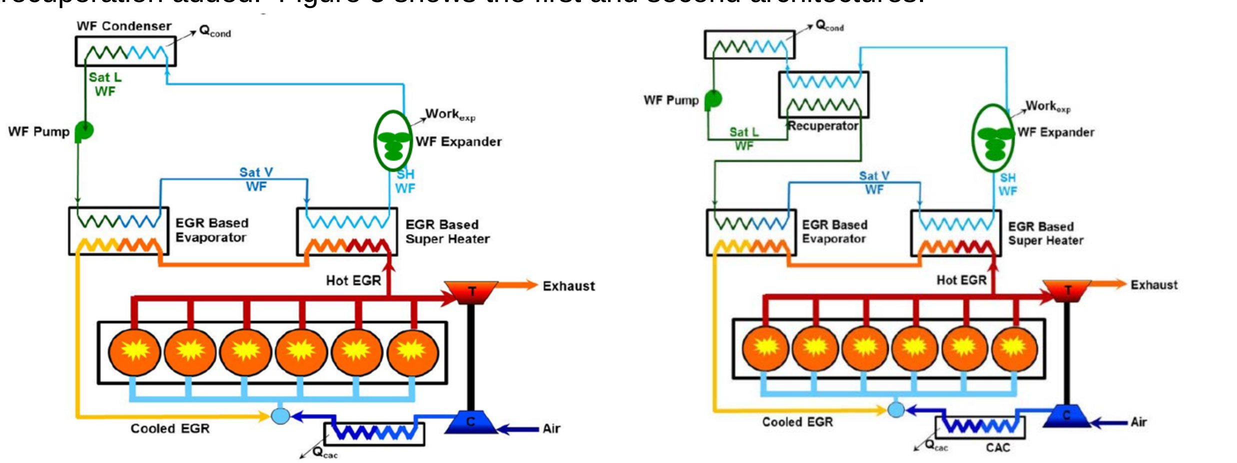


Figure 5: Architecture 1 (Simple EGR WHR) & Architecture 2 (Simple EGR +Recuperation)

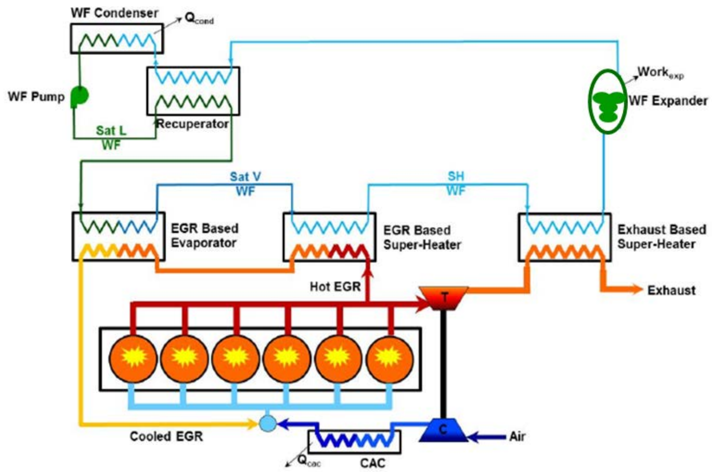


Figure 6: Architecture 3 (EGR + Post Turbine Exhaust Recovery)

The third system architecture adds post turbine exhaust heat energy to augment the EGR energy and recuperation which is shown in Figure 6. The above mentioned three systems were compared using consistent boundary conditions. Each system utilized the same target evaporation temperature (120°C), the same target superheat temperature (350°C), the same target condenser temperature (70°C), and the same expander and working fluid pump efficiencies (60% and 50% respectively). The system architecture was evaluated utilizing 25% post turbocharger turbine exhaust energy recovery. Based on evaluation of the three candidate architectures it was determined that the fuel economy advantages of recuperation merit its inclusion. Architecture 1, 2 and 3 BSFC improvements were 2%, 3% and 5% respectively.

Task 4.0 Concept Refinement, Design and Supporting Analysis

Working fluid pressures over 5 bar and temperatures up to 350°C challenged sealing, bearings, lubrication cooling and material choices for the Roots expander design. Significant design effort and innovation was required to enable for the Roots expander to perform efficiently with no functional issues in an Organic Rankine cycle system. To evaluate the effectiveness of the new Roots expander components, a single stage expander was developed as shown in Figure 7. The single stage expander was tested on an ethanol test stand (Figure 8) to evaluate expander efficiency and functional operation at various pressure ratios and volumetric flow rates.

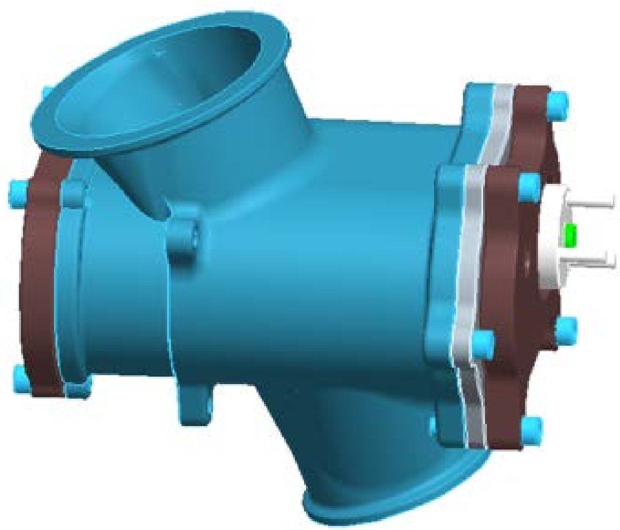


Figure 7: Roots Expander Development

Ethanol Test Stand

An ethanol Rankine cycle test stand (superheater, evaporator and condenser to conceptualize real Rankine cycle) was developed and is shown in Figure 8.

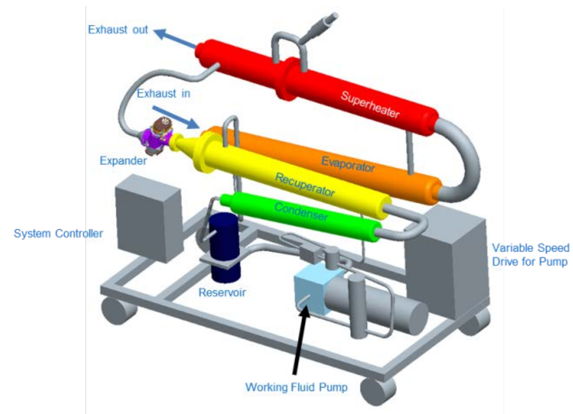


Figure 8: Organic Rankine Cycle Test Stand

ORC System Development

Expander, heat exchangers, working fluid and working fluid pump are major components of an ORC system. Expander design and development were carried out by Eaton Corporation. Initial specification, design and development of heat exchangers, initial specification of working fluid pump and packaging study were completed using a collaborative team effort.

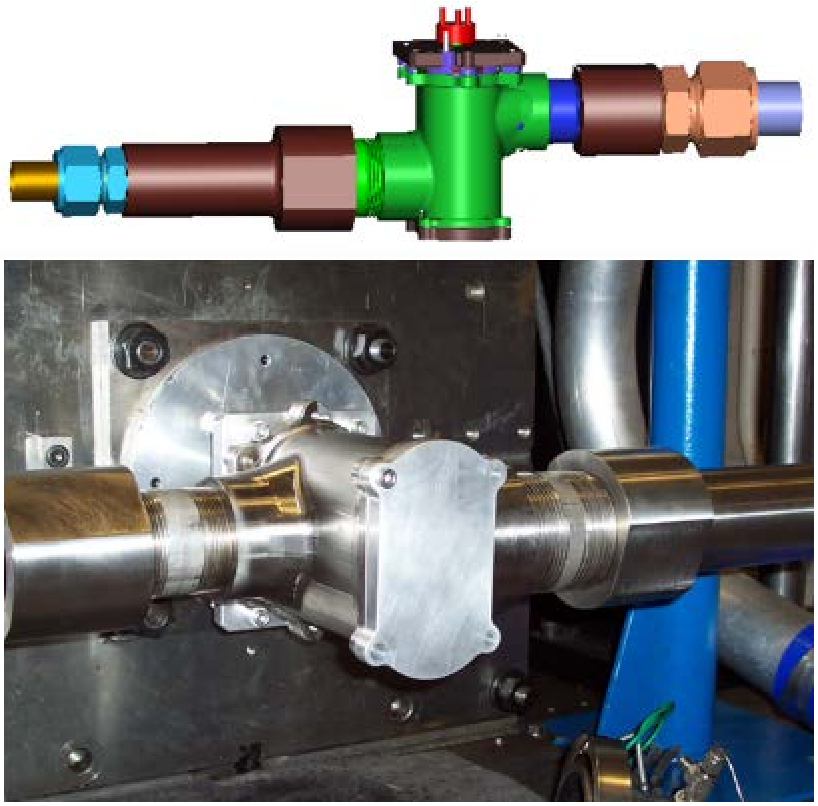


Figure 9: Single Stage Roots Expander

Expander Development

Roots expander development was executed systematically based on working fluid volumetric flow and pressure ratio of the ORC system. Expander size and isentropic efficiency are the two major factors involved in the design and development of an efficient ORC system. Expander size is defined from AVL analytical inputs and isentropic efficiency which is driven by rotor configuration and inlet/outlet porting. Appropriate sealing, bearings, oil cavity cooling, material and rotor coating choices were developed and implemented into the single stage expander design. Significant design effort was focused around the selection of static and dynamic seals based on rotational speeds, pressure and temperature. The single stage expander depicted in Figure 9 has been evaluated through air bench testing and on the ORC test stand (Figure 8) with water as the working fluid.

Expander Sizing

The expander inlet volume flow and expansion ratio characteristics corresponding to the optimized operating conditions shown in Table 6 are:

Table 6: Refined Expander Inlet Volume Flow and Expansion Ratio Specification

Engine		1st Stage		2nd Stage		3rd Stage		Total
Mode (-)	Speed (rpm)	Vol. Flow (m^3/hr)	Exp. Ratio (-)	Vol. Flow (m^3/hr)	Exp. Ratio (-)	Vol. Flow (m^3/hr)	Exp. Ratio (-)	Exp. Ratio (-)
A25	1307	71.4	2.3	158.1	2.3	349.5	1.35	7.13
A50	1307	71.6	2.3	158.5	2.3	350.5	1.79	9.46
A75	1307	71.3	2.3	158.2	2.3	350.1	2.58	13.65
A100	1307	71.2	2.3	158.4	2.3	350.8	3.04	16.07
B25	1609	89.2	2.3	197.5	2.3	436.3	1.53	8.08
B50	1609	89.2	2.3	198.0	2.3	438.0	2.27	12.01
B75	1609	89.8	2.3	199.7	2.3	442.4	2.58	13.66
B100	1609	89.7	2.3	199.5	2.3	442.1	2.58	13.66
C25	1910	106.8	2.3	236.5	2.3	522.4	1.57	8.31
C50	1910	105.5	2.3	234.1	2.3	517.8	2.02	10.69
C75	1910	105.1	2.3	233.7	2.3	517.7	2.20	11.66
C100	1910	105.0	2.3	233.5	2.3	517.1	2.22	11.66

The stage-by-stage inlet volume flows are proportional to engine speed, indicating that a fixed drive ratio between the engine crankshaft and the expander output shaft was feasible; no variable ratio drive system was needed. The high overall expansion ratios are useful in attaining energy recovery from the working fluid.

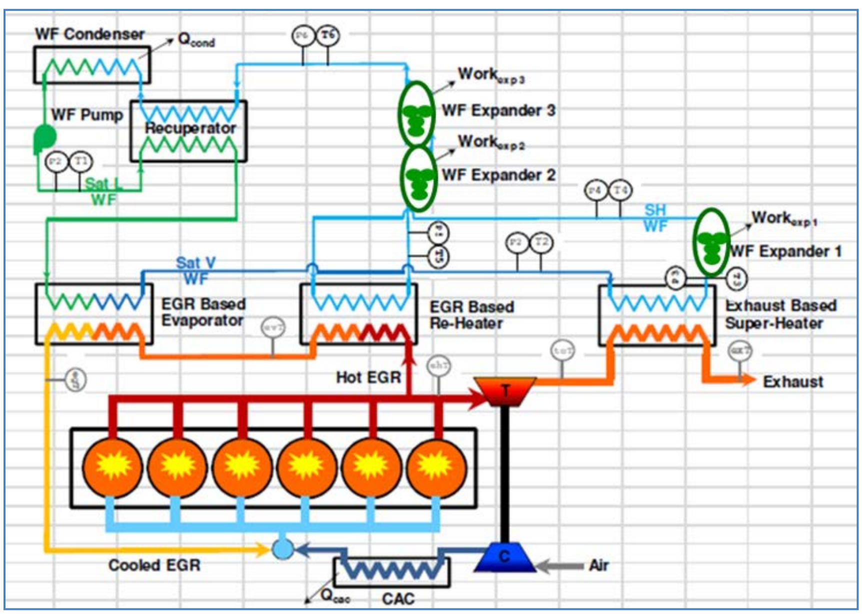


Figure 10: Architecture 4 – Rankine Cycle with Three Stage Expander

Lacking specific map information for the expander, all performance analyses were based on assumed 60% expander efficiency for each of the three (3) stages. The volume flow and expansion ratio specifications shown in Table 6 above are independent of this assumption. The overall fuel economy impacts and condenser heat rejection requirements (which may also affect boiler heat transfer capacities) are dependent on the expander efficiency levels. Higher expander efficiencies will improve fuel economy in a slightly more than linear fashion.

Based on analytical investigations and architecture layout as outlined in Figure 10, utilization of the multistage Roots expander was selected to achieve optimal performance at a wide range of operation in HD diesel engines. Figure 11 provides the fuel economy benefits with fan losses and without. Figure 11 shows that for most of the operating points utilizing the three stage Roots expander, greater than 5% BSFC improvement can be achieved. Different configurations of multistage Roots expanders were compared and investigated. The selected design was made based on performance, flexibility to change internal drive ratios, durability and cost considerations. The inlet and outlet port configurations were optimized using CFD analysis to minimize flow losses and leaks. Figure 12 shows the compact multistage Roots expander and Figure 13 shows the ORC system package.

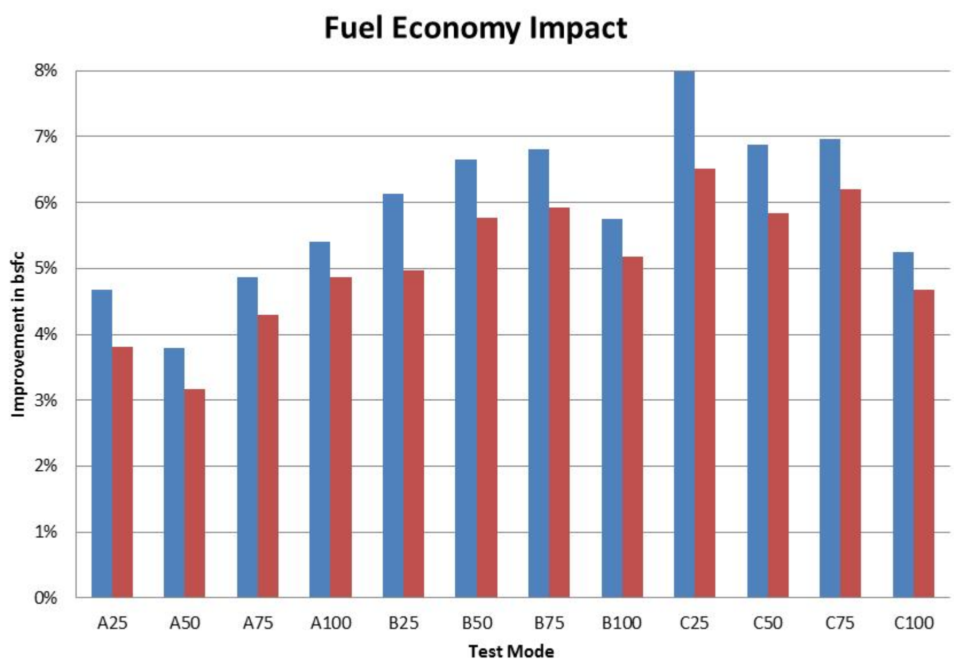


Figure 11: Engine Gross and Net Fuel Consumption Improvements – Three Stage Expander

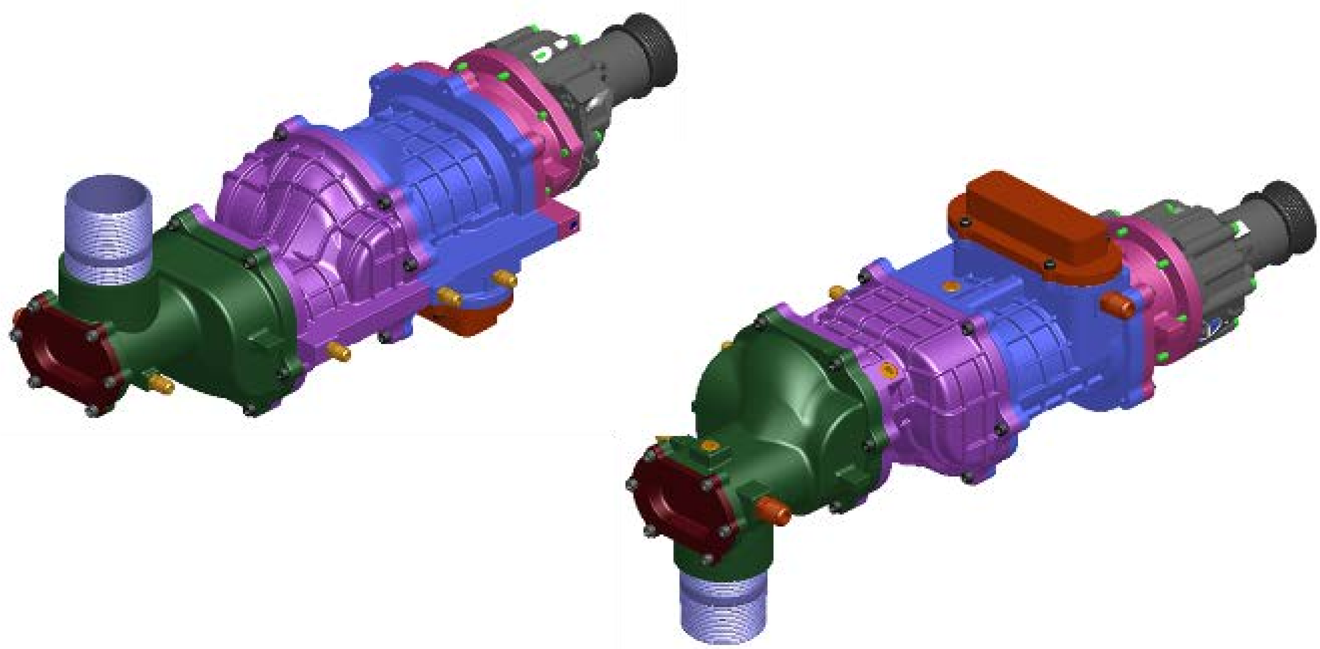


Figure 12: Three Stage Roots Expander

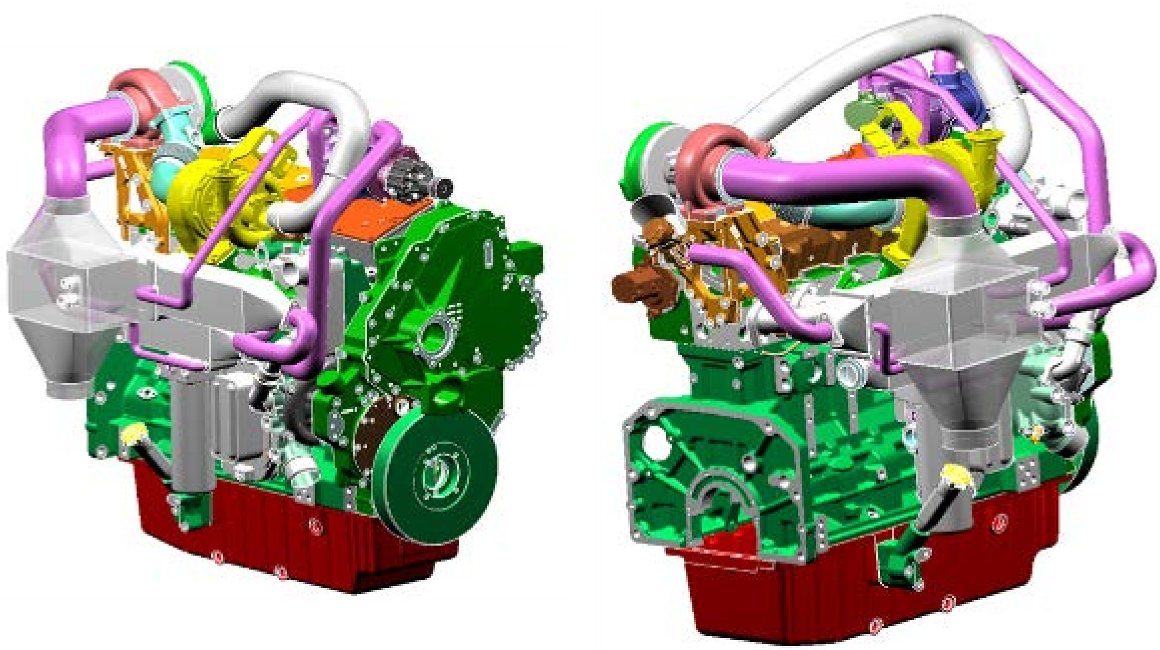


Figure 13: Packaging of Rankine Cycle Roots Expander on John Deere 13.5L Engine

WHR Heat Exchangers

The WHR system that was demonstrated utilized energy recovered from both the Exhaust Gas Recirculation (EGR) path and the post turbine exhaust gas stream. The EGR-based energy recovery path serves to also cool the recirculated exhaust gas. Roots expander ORC WHR heat exchanger system consists of a superheater, evaporator, recuperator and condenser.

A heat exchanger vendor (Modine) was selected for WHR heat exchanger development. Modine, Eaton, John Deere and AVL's expertise are involved in the heat exchanger architecture as well as system specification. These heat exchangers have been analyzed for optimal system performance in parallel and series architecture.

A) Series Architecture - Working fluid passes through EGR source first (evaporator) followed by post turbine exhaust (superheater)

B) Parallel Architecture - Working fluid divides between EGR source and post turbine exhaust. Both will evaporate and superheat the working fluid (Figure 14)

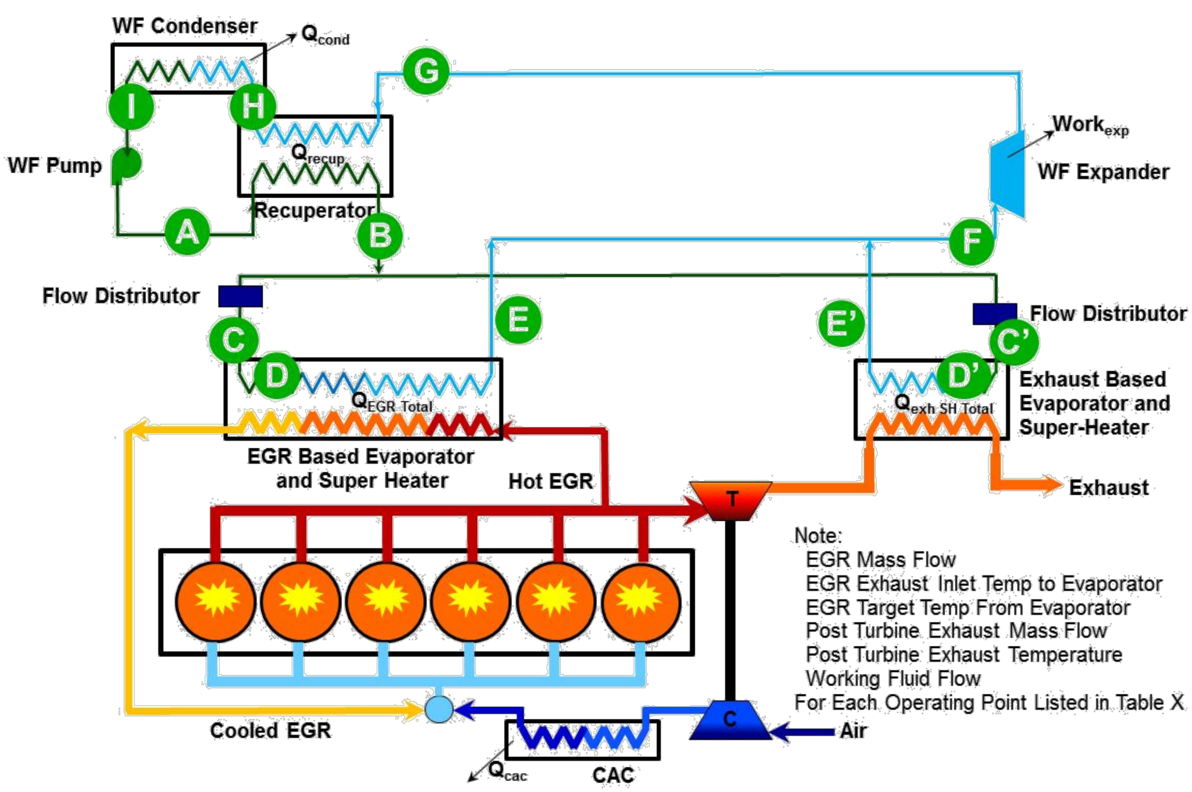


Figure 14: Waste Heat Recovery System – Parallel Architecture

Based on analysis performed by Modine, aluminum was selected as the material for the condenser and recuperator. The test cell demonstration in Phase 3 (2014 -2015) was the first validation of this selection. The EGR boiler and Tailpipe boiler were fabricated with stainless steel because they contact exhaust gas. The EGR boiler replaces an EGR cooler of slightly less cost in most applications.

Working Fluid and Heat Exchanger Architecture Evaluation

The prime path working fluid to be utilized is ethanol, but a mixture of ethanol and water was also evaluated. The ethanol was de-natured with approximately 5% gasoline. Heat exchangers were designed to handle ethanol as well as water. The evaporator and superheater were designed to keep the ethanol working fluid below 275°C on exiting to avoid working fluid degradation.

Working Fluid Pump Requirements

Based on the optimized operating conditions, the working fluid pump specifications are:

- Working Fluid Flow: 0 to 12 kg/min Ethanol (16L/min)
- Working Pressure: 0.1 to 8 bar

Revised Performance Predictions

The initial WHR heat exchanger performance information provided by the heat exchanger supplier was based on the system operating boundary conditions. This information was used as thermodynamic targets to separately specify the capacity of each isolated heat exchanger. During this process, some of the component heat transfer performance was potentially better than initially requested. A systems approach was next utilized to re-optimize the boundary conditions for the WHR system in an attempt to capitalize on the heat exchanger performance and improve overall system performance.

The heat transfer capacities of the heat exchanger components from A75, B75 and C100 conditions were evaluated and included in the AVL system model. The working fluid pressure drop against flow was also incorporated in the AVL analysis, as was an estimate for the diffuser pressure drop required to distribute the slightly super-cooled working fluid across the inlet face of both the EGR and Tailpipe boilers. The parameters were adjusted at each of the 12 engine operating modes, included the target evaporation pressure, total working fluid flow, the proportion of working fluid flow going to the Tailpipe boiler, and the condensation temperature (adjusted subject to the constraints of the condenser behavior). The effectiveness of the superheater portion of the EGR and Tailpipe boilers were adjusted to simulate the hot working fluid bypass and EGR or exhaust gas bypass as needed. This optimization was executed while maintaining a first stage expander volume flow proportional to engine speed, a constant area and wall temperature condenser, and at least 4°C supercooling at the inlet to the boilers.

Task 5.0 - Detailed System Design Implementation

The multistage expander depicted in Figure 12 was designed and developed to achieve optimal performance and efficiency at a wide range of operation in HD diesel engines. The displacement of each expander stage and the drive ratios relative to the pulley speed was defined by outputs from the ORC system thermodynamic model based on operating conditions. The isentropic efficiency of the Roots expander was optimized through CFD to investigate the effect of different inlet and outlet porting configurations, rotor geometries, component material selection to alter the coefficient of thermal expansion and working fluid selection.

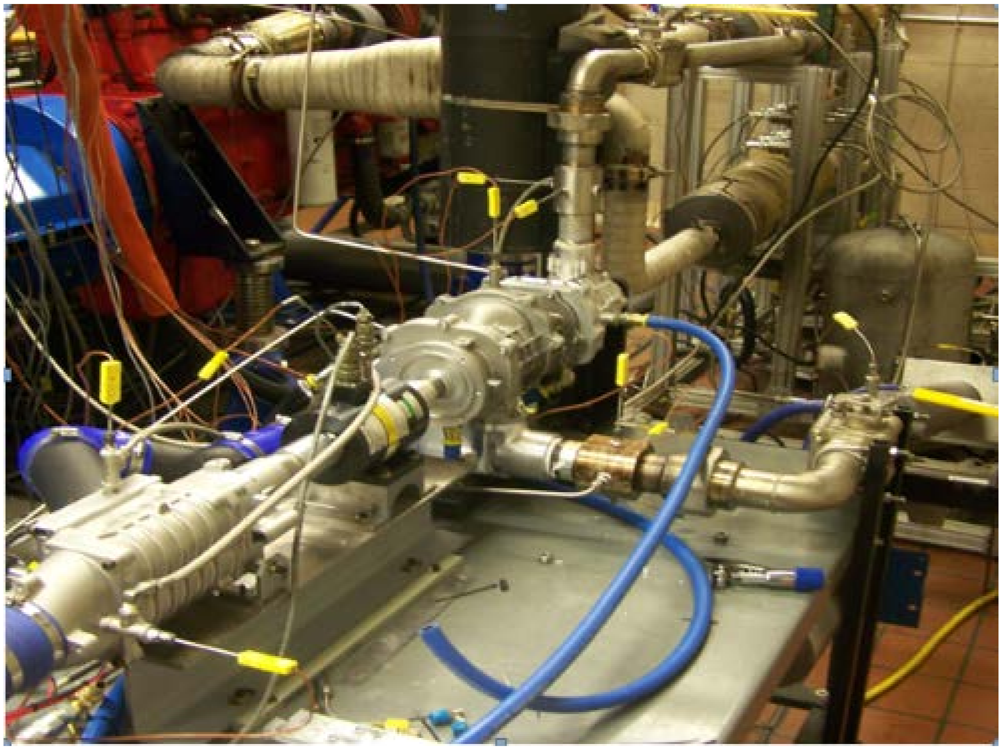


Figure 15: Expander Evaluation Test Setup

The multistage expander was evaluated on the ORC evaluation test setup (Figure 15) with water as the working fluid. The expander mechanically operated as expected with no functional issues experienced during testing and the objective test results correlated to the analytical prediction. For a multistage Roots expander, the drive ratios for each stage relative to pulley speed need to be altered depending on the selected working fluid due to expansion ratio differences of different working fluids. Stage bypassing is an alternative method to compensate for different expansion ratios of working fluids and was implemented into the prototype Roots multistage expander design. Depending on the working fluid and overall operating conditions, varying levels of stage bypassing is possible to maximize output power of the expander.

Task 6.0 - Prototype Procurement

Specifications for all ORC components were developed and system layout design was completed for a 13.5L John Deere engine. Procurement of components and installation onto the engine was successfully completed as outlined in Figure 16.

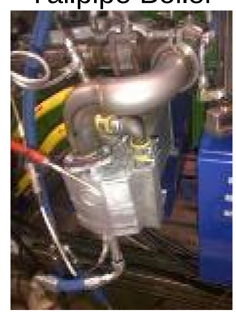
Tailpipe Boiler

Three Stage Roots Expander on Engine

Superheater & Tailpipe Boiler

Reservoir & Working Fluid Pumps

Recuperator & Bypass Valves



Condenser

Figure 16: Roots ORC System Integrated onto Engine

Task 7.0 - WHR System Development Testing AVL

Development of Engine & WHR Controls

The Roots expander has tolerance to two phase working fluids which reduces the system control complexity relative to competing technologies. Matlab models of the three-stage Roots expander WHR system using expander performance equations and efficiency data derived by Eaton and boiler steady-state function were utilized to generate an open loop model to establish operating points for the Simulink control code. The Simulink control code is target compiled for an open ECU controller. Simulations illustrated that the output response of the Roots ORC system is significantly insensitive to mass flow rate variance which simplifies closed loop controllability. An iterative approach was taken for the controls development. As engine data became available, the Matlab model, Simulink components and target code were refined.

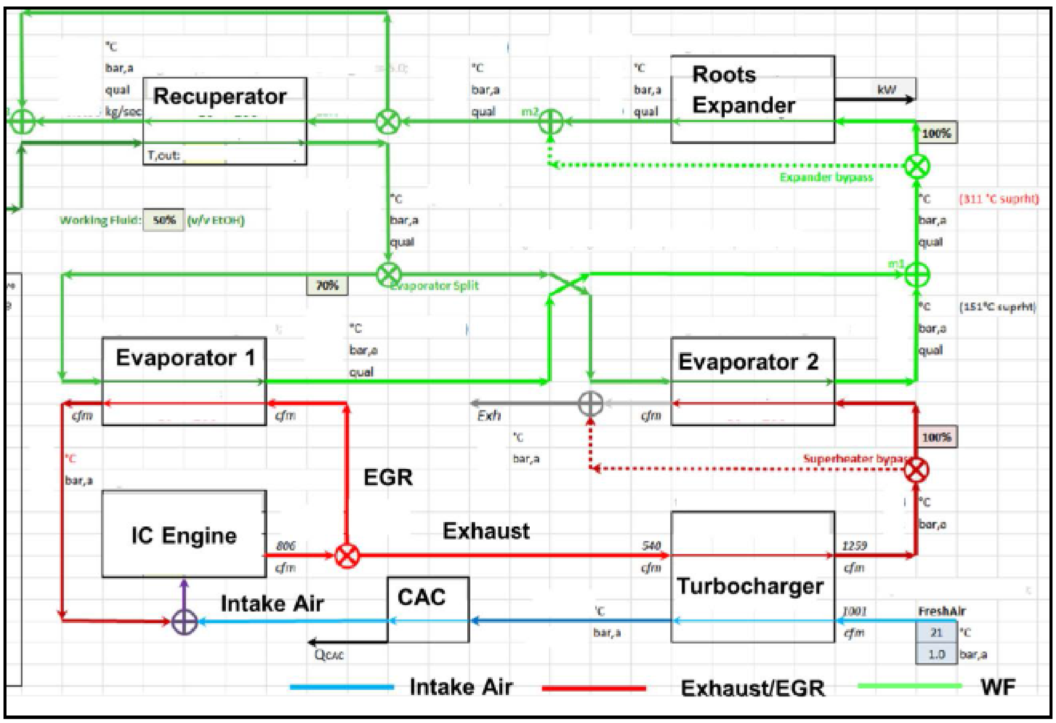


Figure 17: WHR System Control

The control system was initially designed to impart a significant amount of model-based controls, with on-engine testing activities for model refinement and closed-loop tuning as shown in Figure 17. This strategy, aside from expander lubricant pressure control, quickly proved elusive as the many problems encountered forced a new understanding of system behavior and control. The controller software 'user interface' became the 'manual control' panel. Strategies for control were developed by observation of stimulus-response and included into a rule-of-thumb process. The lubricant pressure controller was implemented as a feed-forward set-point with proportional, integral, differential feedback (i.e. F-PID). It was implemented and tuned and proved to work properly.

WHR System Testing

The WHR system with the three stage Roots expander was completely integrated (Figure 18) onto the 13.5L John Deere Engine. The WHR system utilized ethanol as the working fluid. Table 7 provides a description of the data legend for Figures 19 through 26. Due to the system level issues discussed below, it was not possible to obtain data at A25, C25 and C100.

Table 7: Legends Used in the Graphics

Legend	Description
Original Prediction (Vehicle)	Original vehicle bsfc impact prediction done including impact of cooling fan power changes
Original Prediction (Test Cell)	Original prediction done EXCLUDING impact of cooling fan power changes
Originial Prediction + Actual Gearing	Revised prediction done using actual gearing ratios tested
Originial Prediction + Actual Gearing + Actual EGR Gas Conditions (Available Q)	With EGR flow, Q, inlet T and Outlet T from WHR system testing
Originial Prediction + Actual Gearing + Actual EGR Gas Conditions (Available Q) + Add Actual Condenser Conditions	With expander outlet pressure from WHR system testing (by adjusting condenser temperature)
Originial Prediction + Actual Gearing + Actual EGR Gas Conditions (Available Q) + Add Actual Condenser Conditions + Actual Recuperator Cold Out Conditions	With recuperator effectiveness adjusted to match cold side outlet temp from WHR
Originial Prediction + Actual Gearing + Actual EGR Gas Conditions (Available Q) + Add Actual Condenser Conditions + Actual Recuperator Cold Out Conditions + Actual EGR Boiler Out. and Pressure Ratio	With EGR boiler parameters set to match T and Q
Test Results	Results from dyno test cell

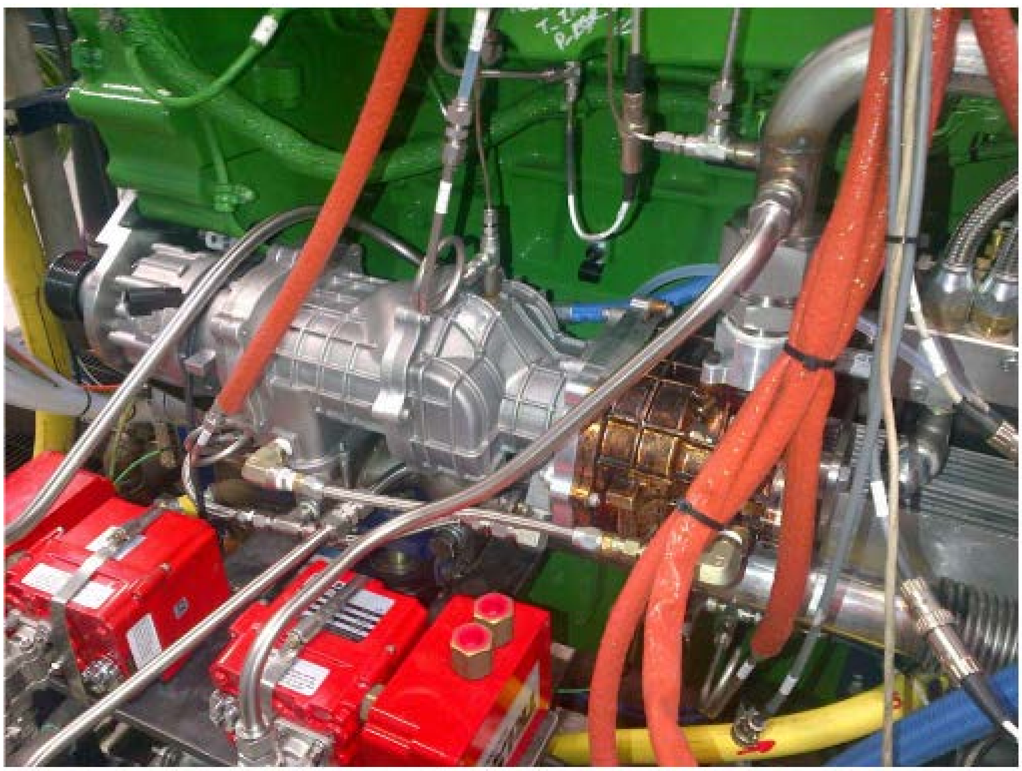


Figure 18: Expander Integrated with John Deere Diesel Engine

WHR System Test Results:

For the demonstration test, the project team used a lubrication circuit that utilized a separation system to separate the working fluid and lubricant. The intention of the system was to deliver working fluid to the heat exchangers and lubrication to the expander gear cases. The implemented separation system delivered very low separation efficiency which subsequently allowed the lubricant to foul all of the heat exchangers. The fouling substantially reduced the available heat energy for the WHR system as shown in Figure 19. Figure 20 shows the fuel economy improvement for predicted, actual with limited enthalpy and extrapolated the experimental results to total available enthalpy systems. The "Test Results" in Figure 20 are calculated as the ratio of the expander power to the engine power. This agrees with the theoretical predictions accounting for lack of post-turbine heat and reduced EGR boiler capability.

The post turbine heat exchanger was completely inoperable due to lubricant fouling and the EGR heat exchanger performance was reduced significantly. Figure 21 shows the elevated working fluid flow rates that were required to keep the engine EGR inlet temperatures at required levels. This increase in mass flow rates resulted in non-superheated expander inlet conditions at lower expander inlet pressures (Figure 22). This result is significant because the ability of the Roots-based expander to accept non-superheated inlet conditions is fundamental to the system robustness assumptions. There was no mechanical impact found due to these conditions.

Pressure ratio is an important parameter to achieve high Rankine cycle efficiency and generate power with the Roots expander. The higher mass flow rates of the waste heat recovery working fluid prevented the system from operating at the target pressure ratios. Heat exchanger fouling increased the condenser backpressure and resulted in higher than predicted pressures at the expander outlet as shown in Figure 23. The combination of reduced inlet pressures and increased outlet pressures significantly restricted the achievable system pressure ratios. The reduced overall pressure ratio resulted in the third stage of the expander generating minimal to no power and in some cases consumed power (Figure 24).

In summary the heat exchanger fouling led to three (3) factors that negatively impacted the overall Rankine cycle efficiency as outlined in Figure 25: (1) reduced available heat enthalpy, (2) total available pressure ratio and (3) minimal pressure ratio for third stage. The expander operated mechanically as expected with no functional issues experienced during testing. Moreover, the expander power was consistent with Eaton's low fidelity expander performance predictions (Figure 26), which supports the Roots-based approach.

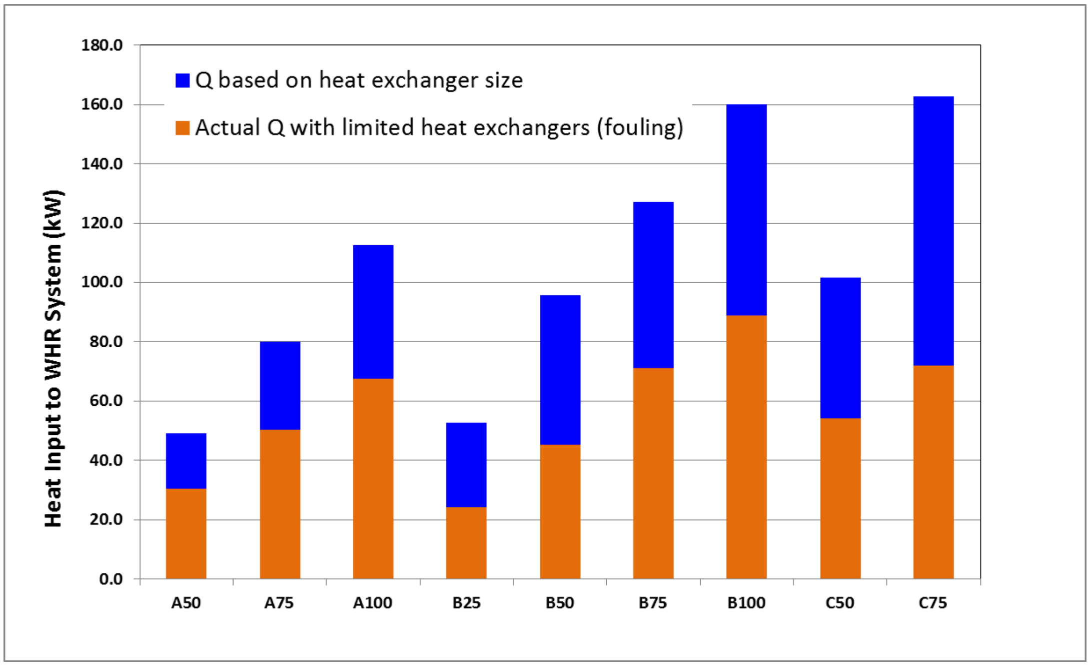


Figure 19: Reduced Heat Input to WHR System

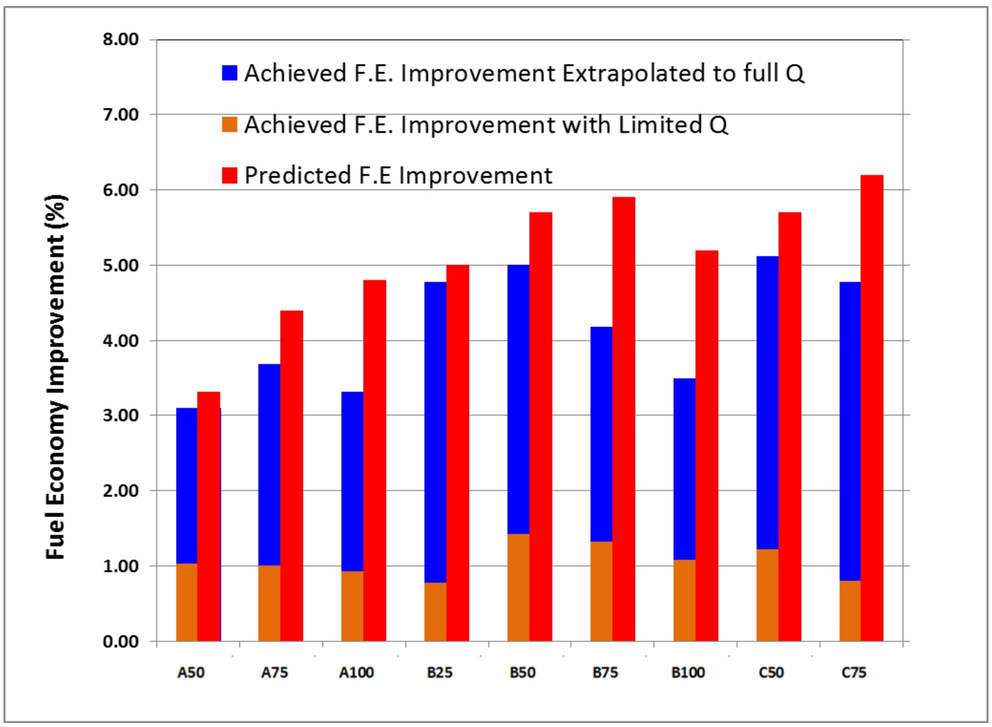


Figure 20: Fuel Economy Impact with Reduced EGR Enthalpy

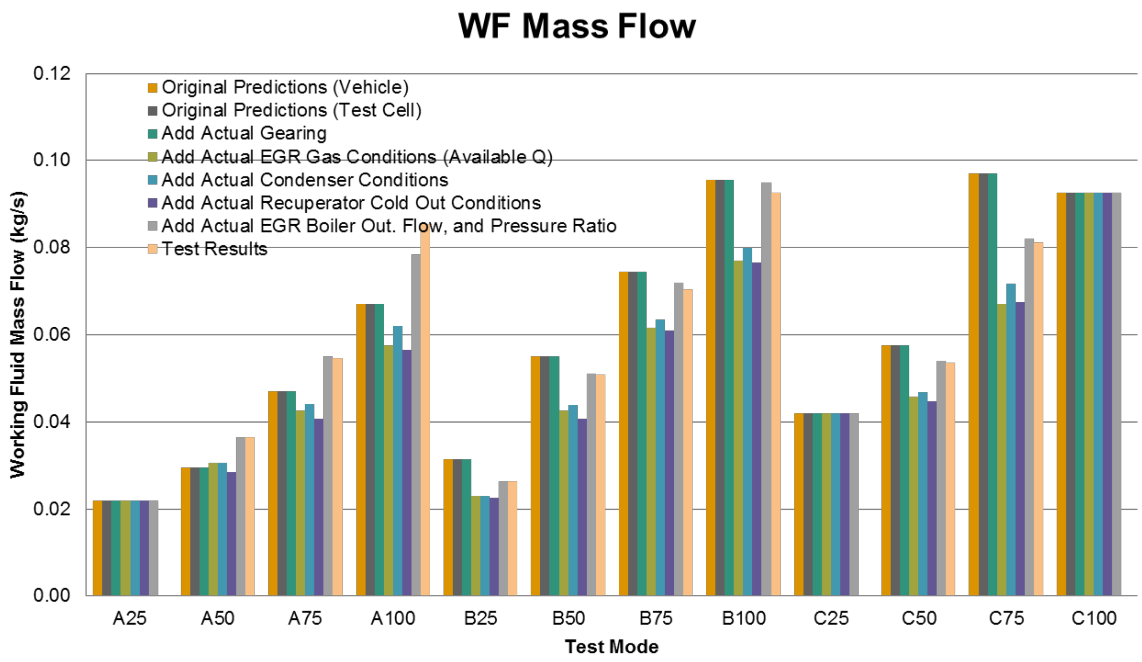


Figure 21: Working Fluid Mass Flow Rate (Higher mass flow rates at lower EGR rates) an impact of reduced performance from EGR heat exchanger

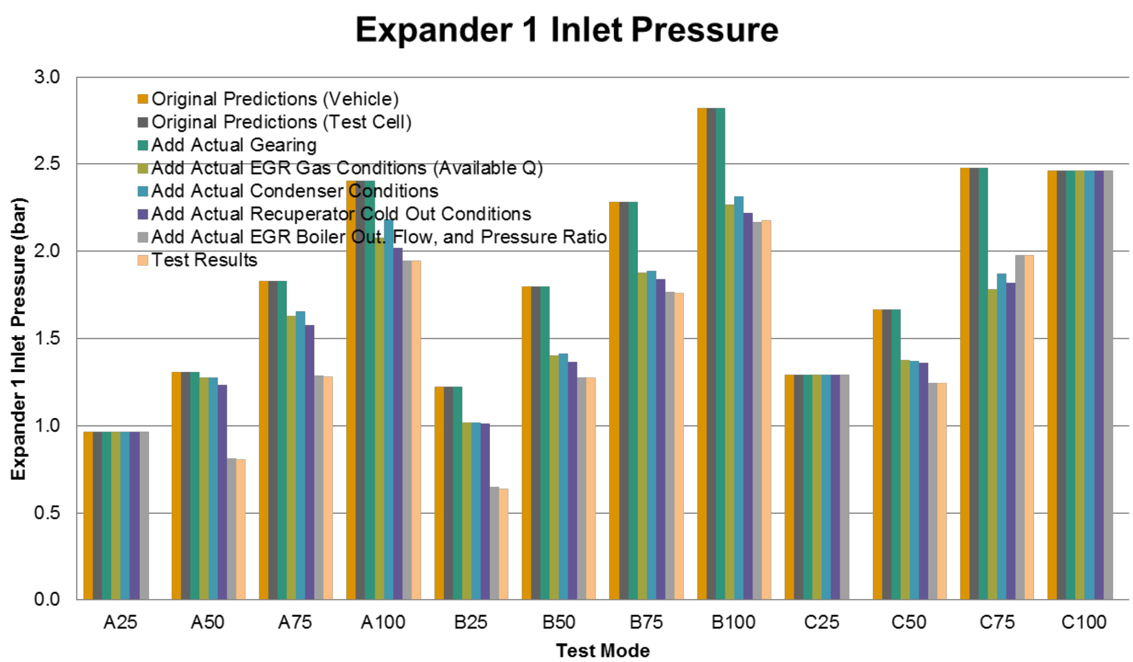


Figure 22: Expander Inlet Pressures Designed Versus Operation

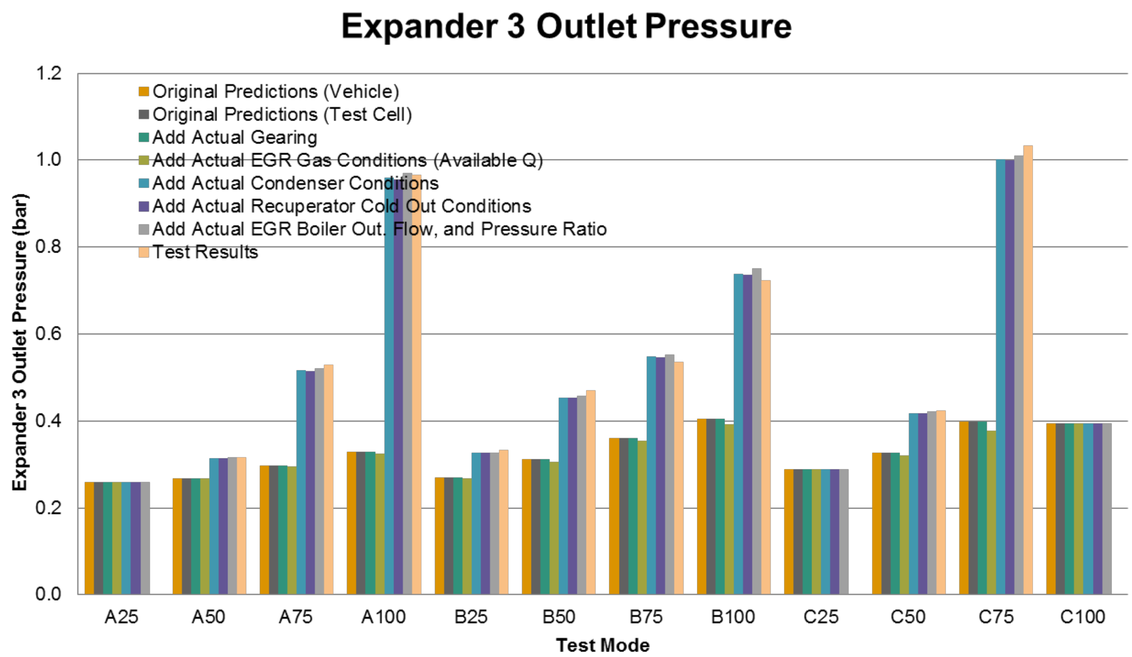


Figure 23: Expander Outlet Pressures (Designed versus Operation)

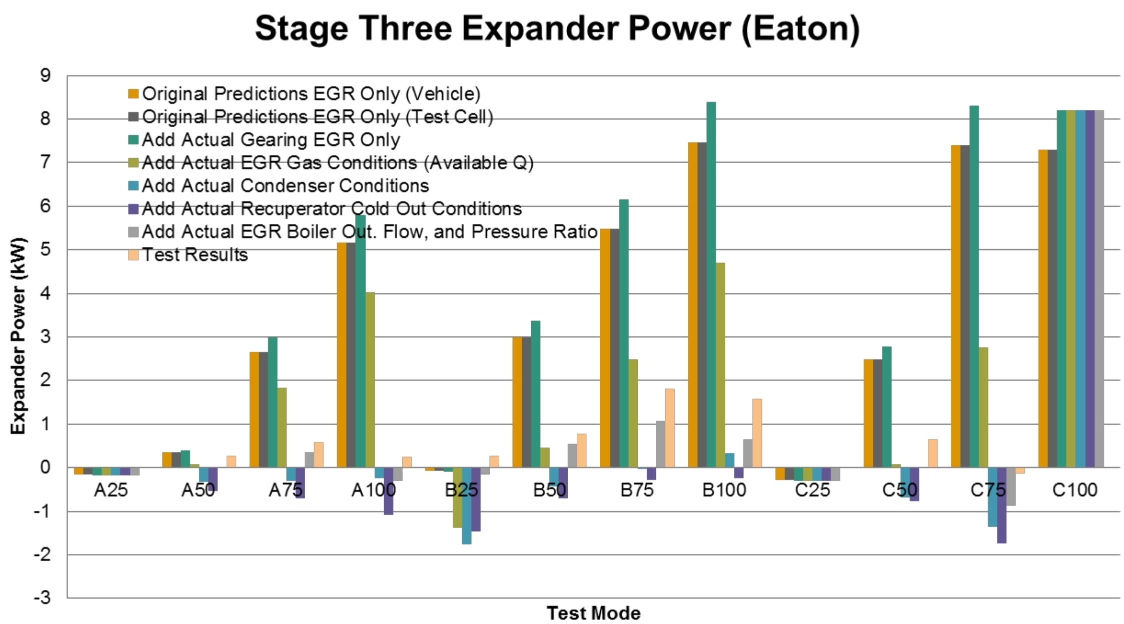


Figure 24: Reduced Third Stage Expander Power Output

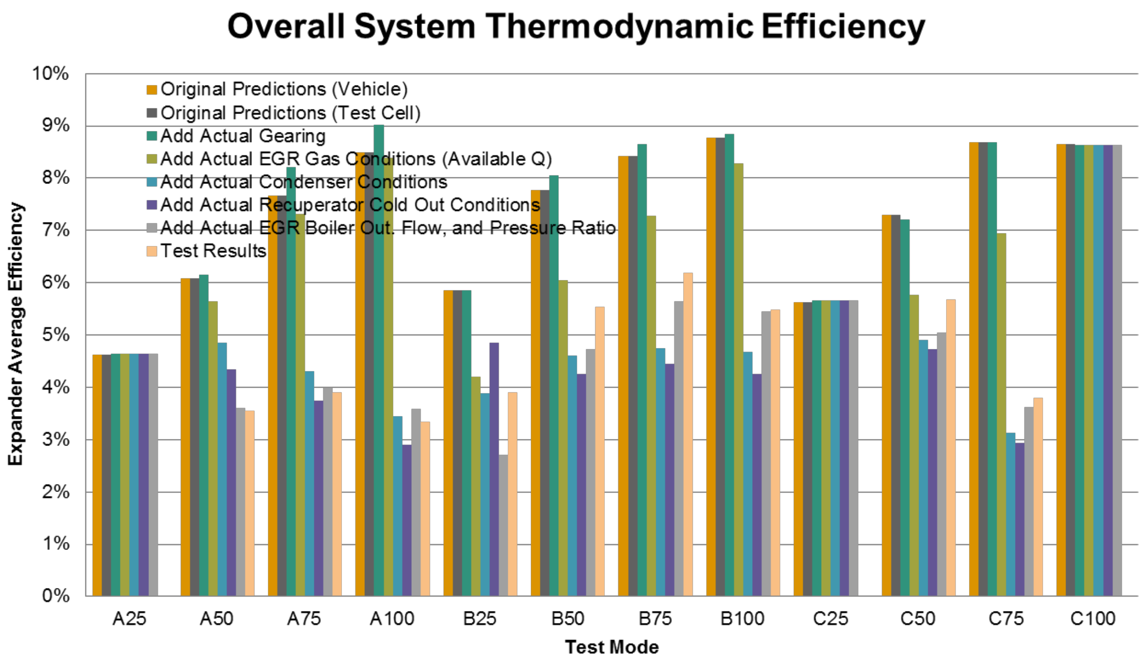


Figure 25: Overall Rankine Efficiency

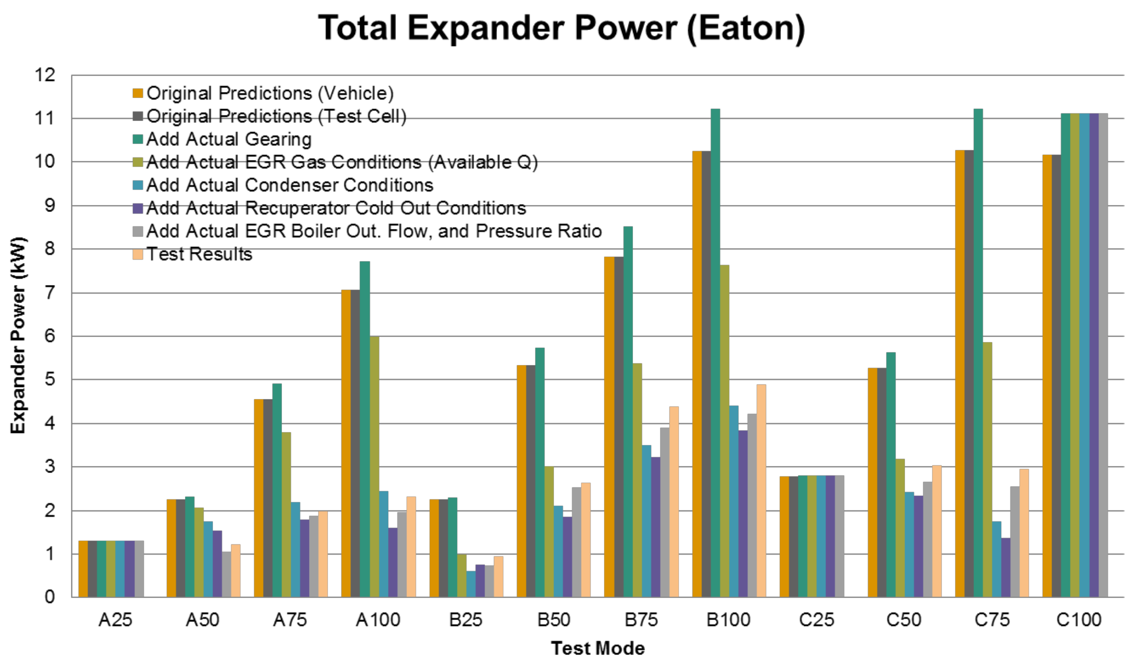


Figure 26: Roots Expander Power from all Three Stages

3.0 Project Activities

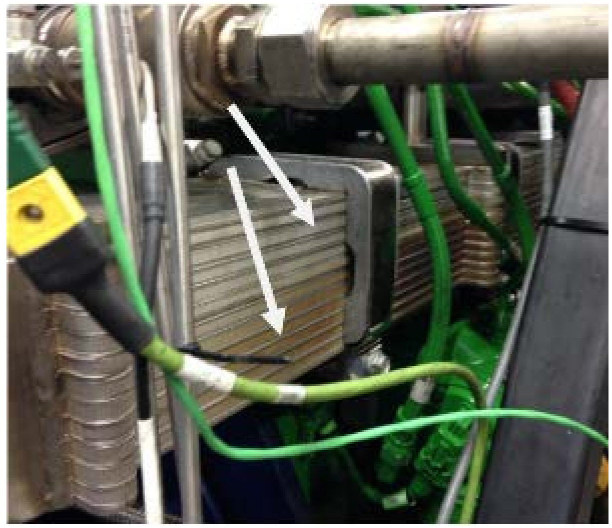
The project was structured to baseline the 13.5 liter HD diesel engine, and to characterize and quantify the potential waste energy sources with a correlated thermodynamic model. The impacts of various WHR heat exchanger layouts on system performance was assessed, leading to specifications of WHR components. The expander development utilized CFD analysis, bench testing, calibration, and validation to maximize efficiency and durability. The developed expander and ORC system was tested on an engine operated over the same speed and load conditions of the baseline engine testing. These results were compared to the baseline engine data at the same NOx emission levels to provide a back to back demonstration of the expander technology and impact on fuel efficiency and engine system performance.

The expander, heat exchangers, working fluid and working fluid pump are the major components within an ORC system. Expander design and development was executed by Eaton. A collaborative team effort between Eaton, AVL, John Deere and ORC component vendors was utilized to develop the ORC system components and all specifications.

The multistage Roots expander WHR system was integrated on the HD Diesel Engine (Figure 18). The WHR system utilized ethanol as the working fluid. The expander operated mechanically as expected with no functional issues and the test results correlate to the analytical predictions. During initial system testing, an external leak of working fluid developed from the EGR boiler 01 (the initial EGR boiler built into the system). The leak was located at the inboard (facing the engine) 'bulge' in the boiler facilitating the working fluid distribution through the internal boiler structure as shown in Figure 27-a. EGR Boiler 01 was replaced with EGR Boiler 02, and EGR Boiler 01 was returned to the heat exchanger supplier to diagnose the failure and attempt to repair the unit by re-doing the vacuum brazing. Both EGR Boilers 01 and 02 were constructed using a 'prototype' brazing process. Testing with EGR Boiler 02 exhibited lower working fluid pressure drop than the original boiler, but also had significantly less heat transfer capacity than the original boiler. It appeared the working fluid flow was not uniformly distributed. The boiler exhibited significant stratified discoloration (Figure 27-b), indicating an imbalance of flow. The system was initially designed with a variable speed working-fluid pump and flow-splitter valve, for partitioning and controlling flow into both the EGR and TP boilers. The valve experienced intermittent and non-linear behavior. The valve failed to operate properly and impacted the boilers. The team replaced the flow-splitter valve with two individual pumps to handle the flow between two heat exchangers.

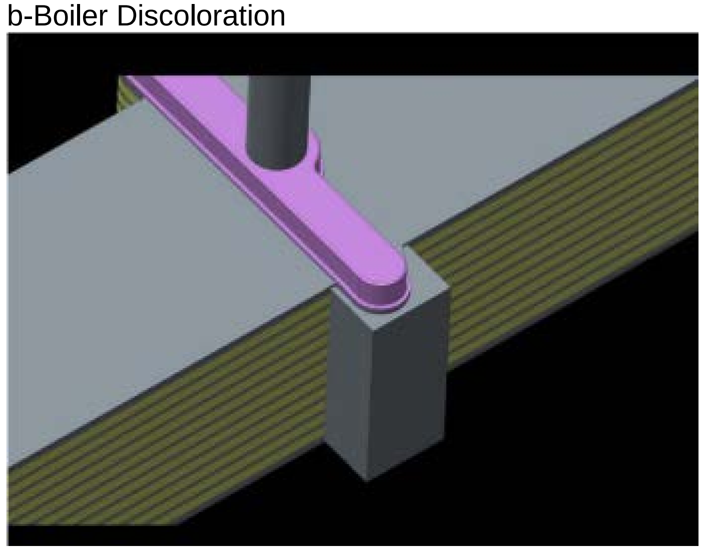
A second parallel-connected pump was added to double flow capacity at higher pressure. The solution failed to deliver the target doubled flow capacity, and the team surmised it was due to pump variability in both command and capacity. The parallel-connected pair operated somewhere between either of the two pumps' performance. The dual-pump solution again ran into the flow-pressure limitation as the higher enthalpy modes required flows inducing even larger pressure drops. The system was reconfigured to a 4-pump solution: two parallel-connected 'lift' pumps, followed by a per-boiler series-connected 'feed' pump. The parallel-pair delivered approximately half of the required boiler pressure, followed by boiler feed pumps supplying the other half pressure. The parallel lift-pump pair was required to deliver flow for both boilers, and the individual feed-pumps only needed to deliver a single-boiler flow.

a- Boiler Leak



b-Boiler Discoloration

c-Boiler Leak



d-Boiler Support Architecture

Figure 27: Boiler Issues

EGR Boiler 03 (new boiler) was installed, and a solenoid-actuated vent on the high temp working fluid side of the boiler duct to the expander was installed to allow for periodic venting of air from the system if required. EGR Boiler 03 had a lower pressure drop than the previous boilers and exhibited better heat exchanger effectiveness before its failure. EGR Boiler 03 developed a leak (Figure 27-c) again at the bulge of the working fluid flow distribution plate, but this time on the outboard side of the boiler. It has been identified that mounting stress (interference fit) and vibration at starting and shutdown process developed this leak.

Discussions as to the cause of the repeated failures with the heat exchanger supplier resulted in investigations of vibration induced failure and AVL performed vibration (acceleration) measurements to provide data for review. The heat exchanger supplier also added gussets to the inboard and outboard regions of the boiler where prior failures had been located as shown in Figure 27-d.

Testing of the parallel boiler (repaired Boiler 03) system ended with degraded working fluid pump performance. The issue was determined to be due to failures of the vanes in the pumps due to cavitation caused by pumping the Krytox lubricant. New working fluid pumps were installed. Again, lack of lubricant caused a bearing failure in the expander. A significant quantity of Krytox was added to the system, attempting to alleviate the symptom. This triggered a redesign of the lubrication system: a larger separator to hold more Krytox, and increase the lubrication flow. Subsequently, corroded gears and bearings stalled expander operation with water. Analysis of gray sediment in the bearing cavities showed rotor coating, believed to be from 'lap-in' wear of the rotors, and fluorine, which is a component of the Krytox lubricant. Eaton believed the abrasive sediment in water working fluid appeared to accelerate bearing failure so a filtering circulation pump system was added to 'clean' the sediment collecting in the separator, so it would not get forced into the bearing cases.

Krytox was likely oxidizing in the high temperature boilers, and creating CO_2(g) and HF(aq) and a heavy 'sludge' residue. This theory fit well with the measured high level of CO_2(g) in the system and the severe corrosion in the steel gears and bearings as HF(aq) is highly corrosive, and sludge residue passivated the boilers.

Table 8 summarizes the system operating data that was obtained with the system configured in the parallel configuration, and with water used as the working fluid. Data was acquired at modes A50 to A100, B25 to B100, and C25 to C75. The expander was unable to operate at the A25 point due to a mismatch between required and available working fluid flow and energy. Engine operation at C100 was not possible due to lack of available EGR cooling function from the degraded EGR boiler function. Due to the reduced EGR boiler cooling function, many of the other high load points were run with lower EGR rates than the baseline engine calibration, resulting in EGR gas temperatures leaving the boiler at or below the baseline values.

Table 8: WHR Summary Data from Parallel Systems Testing with Water

		Mode											
Meas WHR Data		A25	A50	A75	A100	B25	B50	B75	B100	C25	C50	C75	C100
Speed (rpm)		NA	1306	1306	1307	1608	1609	1609	1608	1909	1909	1910	NA
Torque (N-m)		NA	1284	1926	2568	664	1327	1986	2641	583	1172	1757	NA
Power (KW)		NA	175.7	263.5	351.3	111.8	223.5	334.6	444.9	116.6	234.4	351.3	NA
meas bsfc (kg/kW-h)		NA	0.201	0.200	0.202	0.226	0.209	0.204	0.206	0.247	0.220	0.202	NA
EGR Rate (kg/h)		NA	262.0	327.1	399.8	404.3	408.8	540.5	730.8	608.5	640.0	515.9	NA
EGR Inlet Temp (C)		NA	505	623	679	430	538	593	667	379	511	555	NA
EGR Outlet Temp (C)		NA	102	109	139	109	110	99	177	87	134	148	NA
vailable EGR Q_{rej} (kW)		NA	32.5	52.9	68.8	28.0	54.5	83.2	113.8	53.1	74.3	65.4	NA
Exh Rate (kg/h)		NA	752.7	1096.8	1444.0	622.3	1057.8	1529.0	1996.1	823.8	1123.7	1773.8	NA
Exh Temp (C)		NA	361.62	399.99	409.96	347.59	346.12	343.35	401.53	271.87	320.74	342.78	NA
covered Exh Q_{rej} (kW)		NA	49.19	72.99	91.06	38.50	56.86	69.38	86.16	34.59	53.22	74.57	NA
dBSFC %	Eaton dP	NA	1.68%	1.64%	1.43%	1.24%	1.14%	1.52%	1.06%	1.89%	1.14%	0.86%	NA
	m*Cp*dT	NA	1.45%	1.56%	1.54%	0.70%	0.98%	1.59%	1.45%	1.54%	0.98%	0.76%	NA
	dFR	NA	2.11%	2.34%	2.18%	0.85%	0.86%	2.70%	1.54%	1.93%	2.61%	5.07%	NA
Tot Exp Pwr (kW)	Eaton dP	NA	2.91	4.26	4.96	1.37	2.51	5.00	4.67	2.16	2.63	2.98	NA
	dT	NA	2.52	4.04	5.32	0.78	2.17	5.24	6.36	1.77	2.28	2.65	NA

Krytox sludge from the previous testing (water ORC) led to an inoperable tail pipe boiler. Condenser performance was determined to be insufficient for the same reason of lubricant sludge. Further testing of WHR system was carried out with ethanol and 10W-30 oil as the lubricant. The bearing cavities were lubricated by the high-pressure working fluid vapor, as the cavity return lines were left to drain back to the reservoir. This allowed oil-laden vapor to leak past the shaft seals acting as lubricant. This arrangement appeared to work well as inspection of the expander bearings with all subsequent testing showed no wear issues. The demand for higher working fluid pressure into the boilers caused conversion from a vane working fluid pump to a gear pump with a variable speed drive (VFD.) Better flow control was attained due to the new ability to turn the pump at very low speeds for light loads, and the new gear pump easily overcame back-pressure induced by the restrictive EGR boiler.

Table 9 summarizes the most appropriate system operating data acquired with the Ethanol ORC system. The system was configured to use only the EGR boiler due to the severe plugging incurred in the Tail-Pipe boiler, and ethanol was used as the working fluid. Data was acquired at modes A50 to A100, B25 to B100, and C50. The expander was unable to operate at the A25 and C25 points due to a mismatch between required and available working fluid flow and energy. Engine operation at C100 was not possible due to lack of available EGR cooling function from the degraded EGR boiler function. System data was acquired at C75, but no fuel flow or emission measurements were taken at this condition so those data are not reported in this section. Due to the reduced EGR boiler cooling function, many of the other high load points were run with lower EGR rates than the baseline engine calibration to keep EGR gas temperatures leaving the boiler at or below the baseline values.

Table 9: Summary of Data from EGR Only / Ethanol Testing

Meas WHR Data		Mode											
		A25	A50	A75	A100	B25	B50	B75	B100	C25	C50	C75	C100
Speed (rpm)		NA	1304	1305	1304	1606	1606	1608	1608	NA	1910	NA	NA
Torque (N-m)		NA	1295	1946	2595	666	1338	1999	2642	NA	1174	NA	NA
Power (KW)		NA	176.9	265.8	354.5	112.0	225.1	336.4	444.8	NA	234.9	NA	NA
meas bsfc (kg/kW-h)		NA	0.201	0.201	0.201	0.222	0.204	0.204	0.203	NA	0.216	NA	NA
EGR Rate (kg/h)		NA	244.4	315.7	386.1	404.3	376.4	483.9	543.6	NA	493.7	NA	NA
EGR Inlet Temp (C)		NA	501	617	667	409	496	598	644	NA	477	NA	NA
EGR Outlet Temp (C)		NA	96	107	116	91	107	130	125	NA	119	NA	NA
vailable EGR Q_{rej} (kW)		NA	30.4	50.5	67.6	24.3	45.4	71.0	88.9	NA	54.1	NA	NA
Exh Rate (kg/h)		NA	747.1	1090.9	1433.3	590.2	1019.4	1507.3	1922.1	NA	1236.5	NA	NA
Exh Temp (C)		NA	307.49	359.17	386.54	270.89	293.61	348.54	369.57	NA	285.72	NA	NA
covered Exh Q_{rej} (kW)		NA	48.53	84.72	114.27	33.13	62.53	98.11	109.12	NA	85.69	NA	NA
dBSFC %	Eaton dP	NA	1.04%	1.01%	0.94%	0.79%	1.44%	1.34%	1.10%	NA	1.24%	NA	NA
	m*Cp*dT	NA	2.60%	1.38%	1.00%	0.64%	1.95%	2.89%	1.15%	NA	1.61%	NA	NA
	dFR	NA	2.08%	1.90%	2.56%	2.96%	3.31%	2.66%	2.72%	NA	4.26%	NA	NA
Tot Exp Pwr (kW)	Eaton dP	NA	1.82	2.67	3.31	0.88	3.20	4.44	4.82	NA	2.88	NA	NA
	dT	NA	4.48	3.62	3.51	0.72	4.30	9.45	5.07	NA	3.72	NA	NA

The project team learned that the proposed lubrication circuit that utilized a separation system to separate the working fluid and lubricant did not work efficiently as intended. The intention of the system was to deliver working fluid to the heat exchangers and lubrication to the expander gear cases. The implemented separation system delivered very low separation efficiency which subsequently allowed the lubricant to foul all of the heat exchangers.

The BSFC improvements attributable to the WHR were not sufficient to demonstrate the initial program goal of improving BSFC by 5%. Figure 28 shows the breakdown analysis of the discrepancy between the goal and the measured results. The compromised function of the condenser (from both Krytox fouling and formation of incondensable gasses (CO_2) from Krytox decomposition), reduced boiler capacities and the subsequent need to re-adjust the system operating conditions to compensate for this degradation, and the reduced expansion ratio caused by high condenser inlet pressures at some operating points all contributed to the lower than expected system improvement to BSFC.

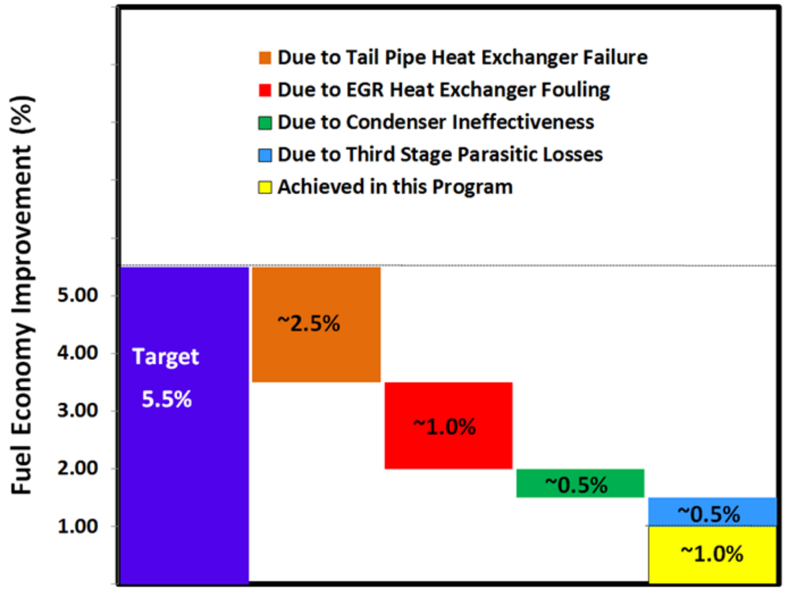


Figure 28: High Level System Analysis

Even with the reduced ability to capture waste energy from the EGR boiler only with ethanol as a working fluid compared to the earlier tests using water as the working fluid and both boilers, many of the modal BSFC improvements demonstrated from the ethanol based tests showed more improvement. This is partially due to improved expander performance as the design evolved and issues were resolved with ethanol compared to water.

The working fluid was removed from the system and the engine and attached WHR system was removed from the AVL test cell and stored at AVL awaiting disposition instructions from DOE. At the time this report was authored, there is a plan in place to continue development of the Roots based expander technology, including lubrication system refinements. This refined expander could be integrated into the engine/WHR system for evaluation.

4.0 Conclusions and Recommendations

The conclusions that can be drawn from the program are:

- The target improvements in BSFC from the Roots based system were not demonstrated due to component degradation from target performance.
- Initial system and expander lubrication issues were sufficiently resolved to allow for eventual system operation and testing, but the damage caused to the heat exchangers during this development process severely limited their performance and subsequently the ability of the WHR system to perform and achieve the efficiency improvement objectives.
- Both water and ethanol use as working fluid were demonstrated.
- The efficiency levels of the expander stages were near or above target after the initial lubrication, sealing and leakage issues encountered were resolved.
- In general, the Roots expander concept was shown to be viable but further development of the hardware is needed.

The efforts to improve expander design should include –

- Improved expander stage sizing and speed (gearing) to better match overall system requirements. Investigate expander stage sizing and gearing that is based on stage by stage volume swallowing capacity.
- Revise the expander shaft sealing and bearing support structure to reduce stage-to-stage working fluid leakage (especially first stage inlet to third stage outlet) and allow for improved clearance control between rotor tips and housing.
- Bench test and map each expander stage over appropriate inlet pressure, temperature (density) and volume flow levels before using the device on an engine-based system.
- Identify and use non-halide based lubricants. Operation with lower superheat / temperature levels may enable use of more conventional lubricants.
- Develop a lubrication concept for the expander that does not rely on separation of slipped lubricant from the working fluid.
- Expander speed measurement should employ speed measurements on both sides of the clutch as well as expander shaft speed for the expander (likely stage one) furthest down the gear train from the clutch
- Further engine based tests should employ an in-line torque meter to directly measure expander output torque.

Recommended efforts to improve the Boilers should include –

- Develop consistent working fluid side pressure drop pressure characteristics of the EGR boiler.
- Refine manufacturing process to minimize twist and/or warp in the EGR boiler.
- Implement an integral exhaust bypass around the Tail-Pipe boiler for cost effective control of the exhaust energy available to the boiler.

Recommended efforts to improve Condenser and Recuperator –

- Continue to investigate the potential of using aluminum construction for these devices when using ethanol as a working fluid. The short duration of component exposure ethanol and the fouling (potentially protective) of the wetted surfaces of these devices by Krytox precluded any conclusion as to the long term viability of aluminum for these applications.
- Further investigate the recuperator hot side bypass approach to control boiler inlet state required.

Recommended efforts to improve working fluid pumps and flow control –

- Develop models of pump behavior to provide modeled working fluid flow values.
- Calculate working fluid flow based on heat exchanger energy balance
- Use submerged (in the working fluid reservoir) working fluid pumps to minimize cavitation at the pump inlets due to line restrictions
- Further investigate the multiple pump flow control attempted during this program to replace splitter valve(s) for flow control.

General recommendations –

- Have at least 1 spare of all boilers (EGR and Tail-Pipe) available, with one more extra spare available for delivery within a 2 week lead time.
- Have at least 1 spare expander available.
- Further refine the working fluid pump selection for robustness against system plugging
- Do not use a lubricant incompatible with the expected surface temperatures possible in the boilers or that results in products of decomposition that cause other working fluid side system components to corrode or fail.
- The system performance trade-off between superheating and higher mass flow should be further investigated. The Roots machines are very tolerant of two phase flow, allowing use of saturated working fluid as an option for balancing EGR cooling function, working fluid volume and mass flow, and system performance that wasn't considered in some of the initial analysis. These parameters may provide more latitude in system set point calibration to optimize performance.

5.0 Products Developed

1. Eaton Engineering Specialist, Matthew Fortini presented at the SAE 2014 Commercial Vehicle Engineering Congress in Rosemont, Illinois on October 9,2014
2. Eaton Engineering Specialist, Matthew Fortini presented at the CTI 2015 Conference in Stuttgart, Germany on May, 2015.
3. Eaton Engineering Specialist, Matthew Fortini to present at 2015 Automotive Organic Rankine Cycle Consortium in Denver, CO on November 20, 2015.