Affordable Rankine Cycle

Program (Phase 1) Final Report

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AFFORDABLE RANKINE CYCLE PROGRAM

Project Background

Nearly 30% of fuel energy is not utilized and wasted in the engine exhaust. Organic Rankine Cycle (ORC) based waste heat recovery (WHR) systems offer a promising approach on waste energy recovery and improving the efficiency of Heavy-Duty diesel engines. Major barriers in the ORC WHR system are the system cost and controversial waste heat recovery working fluids. More than 40% of the system cost is from the additional heat exchangers (recuperator, condenser and tail pipe boiler). The secondary working fluid loop designed in ORC system is either flammable or environmentally sensitive. The Eaton team investigated a novel approach to reduce the cost of implementing ORC based WHR systems to Heavy-Duty (HD) Diesel engines while utilizing safest working fluids.

Affordable Rankine Cycle (ARC)

The ARC concept aimed to define the next generation of waste energy recuperation with a cost optimized WHR system. ARC project used engine coolant as the working fluid. The engine coolant is typically an ethylene glycol (EG) plus water mixture which brings significant value compared to flammable or environmental sensitive working fluids used in conventional WHR systems. This approach reduced the need for a secondary working fluid circuit and subsequent complexity. A portion of the liquid phase engine coolant has been pressurized through a set of working fluid pumps and used to recover waste heat from the exhaust gas recirculation (EGR) and exhaust tail pipe exhaust energy. While absorbing heat, the mixture is partially vaporized but remains a wet binary mixture. The pressurized mixed-phase engine coolant mixture is then expanded through a fixed-volume ratio expander that is compatible with two-phase conditions. Heat rejection is accomplished through the engine radiator, avoiding the need for a separate condenser. The ARC system has been investigated for PACCAR's MX-13 HD diesel engine.

Primary Objectives

The primary objective of this project was to: design, develop, analyze, optimize and demonstrate an Affordable Rankine cycle (ARC) system - a simple, cost-optimized engine waste heat recovery system for HDDE applications.

Fiscal Year (FY) 2016 & 2017 (Phase 1) Objectives

- Analyze baseline engine for exhaust heat energy availability and working fluid feasibility
- Quantify the Affordable Rankine cycle WHR system fuel economy improvement from simulation

Approach

This investigation was structured to baseline the 13 liter HD diesel engine, characterize and quantify the potential waste energy sources for the thermodynamic model development. WHR system components (heat exchangers, expander and pumps) were specified by architecture analysis. WHR components' design and development utilized CFD analysis and thermodynamic models. Those component level models and results were used to predict the ARC system performance.

Phase 1 Summary

On Feb. 18, 2016, the project team completed the Kickoff Meeting. Engine coolant analysis proved the feasibility of using it as WHR working fluid. Engine baseline data helped optimize heat sources and WHR system architecture. Preliminary concept development on heat exchangers (EGR boiler & tailpipe) was completed for ARC operation. The performance characteristics of the proposed Roots expander did not align well with the pressure ratio and flows of the optimum system architecture. The issue was promptly reported to DOE team and alternative expanders (piston, scroll, screw and vane) were evaluated with DOE contract officer approval. The vane expander was selected for ARC demonstration. Early 2017 Eaton team completed detailed ARC system performance analysis using specific OEM defined boundary conditions and constraints. The results identified major challenges with meeting the program target (5% fuel economy improvement). Results were presented to DOE team in a Go/No-Go review meeting. It was decided to stop the program with phase 1 efforts by 31st March 2017.

2 TECHNICAL ACCOMPLISHMENTS

Baseline Engine Characterization & WHR Analsyis

Engine Baseline:

The PACCAR Model Year 2015 MX-13 (Figure 1) baseline test results were used to analyze the available enthalpies for an affordable waste heat recovery system. The team has down selected five steady state operating conditions (Table 1) for the WHR system demonstration which reflects more than 90% of the vehicle operation time. Table 1 shows the potential heat sources available in 2015 MX-13 diesel engine platform.

Properties	Value
Charging System	Variable geometry turbocharger
Fuel Injection System	2500 bar Common rail fuel injection system
Bore	130 mm
Stroke	162 mm
Compression Ratio	17.4 : 1
Connecting Rod Length	262 mm
Valve Configuration	4 / cyl

PACCAR MX13 (I-6) 360kW @ 1700 rpm 24.3bar peak BMEP with #2 diesel

Figure 1: Engine Specifications

WHR Analysis:

Eaton, PACCAR and AVL evaluated different WHR system architectures based on heat availability and potential work recovery. Figure 2 shows the initial work to identify potential heat sources and architectures

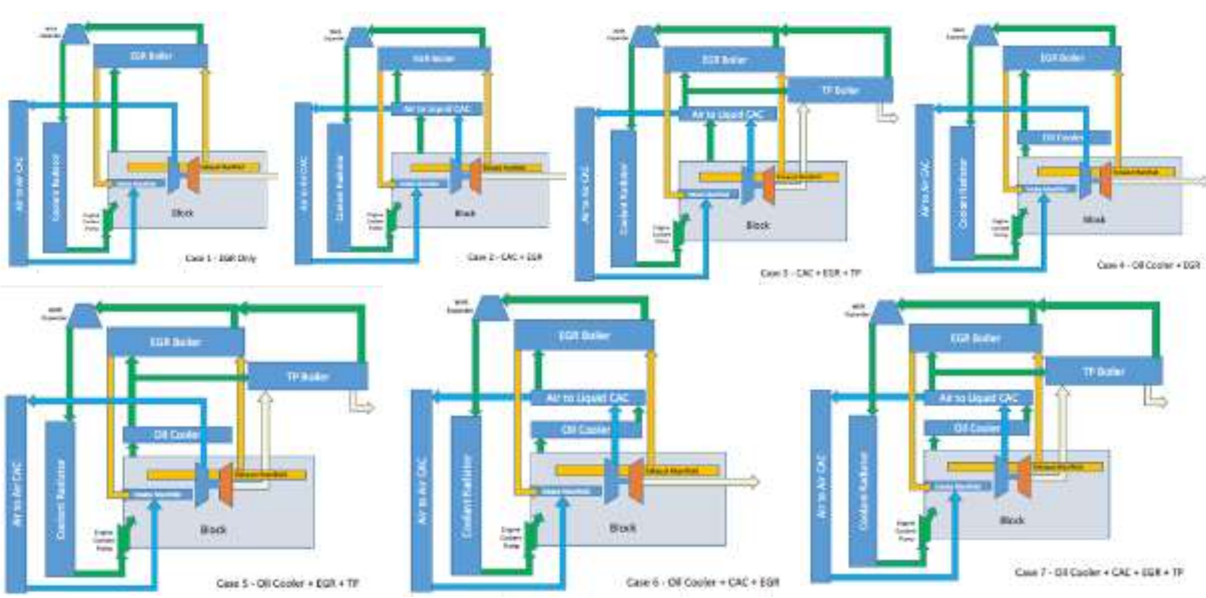


Figure 2: Preliminary WHR Architecture for ARC Study (TP in this figure represents Tail Pipe).

Table 1: Potential Heat Sources and Heat Rejections at Selected Steady State Points.

Heat Sources	
Exhaust	EGR Cooler
	Engine Exhaust Energy
Coolant	Engine Coolant (w/o EGR)
	Engine Oil Cooler
	Charge Air Cooler (estimated)

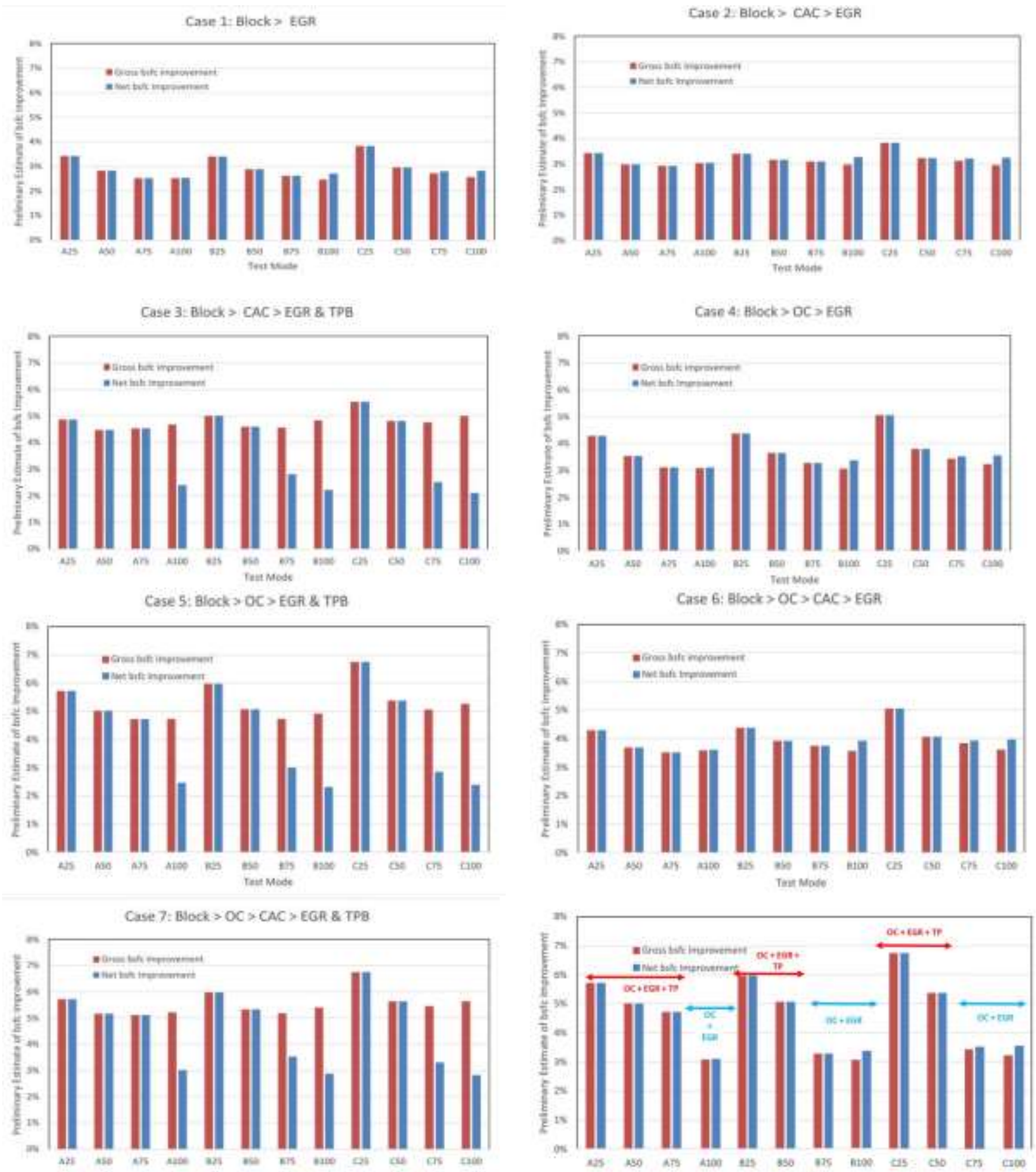


Figure 3: Architecture Analysis (TPB in this figure represents Tail Pipe Boiler).

Figure 3 shows the potential fuel economy improvements from various architectures with an assumption of 6% overall WHR system efficiency. Net brake specific fuel consumption (bsfc) shown in the graphs in Figure 3 included pump power and fan losses. Considering the above performance analysis and system cost, the team finalized the WHR architecture. Other than tail pipe exhaust boiler, all other heat exchangers (EGR cooler, oil cooler, charge air cooler) are currently installed on the vehicle (assuming conversion of the EGR cooler to a boiler). However, the addition of the tail pipe boiler, exhaust splitter valve and corresponding working fluid control system adds significant system cost.

Direct comparison of case 2 and 4 with respect to case 1 from Figure 3 shows that the oil cooler and charge air cooler (CAC) are nearly equivalent for a majority of the operating points. The oil cooler is a consistent heat source and also shows better results at lower loads than the CAC. A tail pipe heat exchanger adds value at lower loads but fan losses penalize the net fuel economy improvements at higher loads. The team decided to arrange the tail pipe boiler in parallel with the EGR boiler so that it can be bypassed at higher load operating conditions to minimize fan losses.

Coolant Feasibility Analysis
Engine coolant feasibility analysis was carried out in three steps. Simple theoretical analysis, preliminary laboratory scale experimental analysis and detailed experimental analysis of subjecting the coolant in to real environment (high temperature diesel exhaust through a heat exchanger).

Theoretical Analysis:
A theoretical study of a two component engine coolant boiling behavior characteristics was completed as the first step of feasibility analysis. Figure 4 shows the engine coolant (ethylene glycol plus water) behavior from saturation liquid to saturation vapor. This illustrates that engine coolant is a zeotropic mixture. We can possibly run the WHR system without vaporizing glycol component. Theoretical results were validated with preliminary experimental analysis (shown in Figure 5) derived from laboratory-scale experiments at Shell. Figure 5 shows the enthalpy change of the engine coolant with respect to temperatures. . Direct comparison of Figure 4 and 5 gives us the information of ethylene-glycol plus water vaporization behavior at different pressures. For example evaluation at 2 bar pressure (red line in Figure 4 and black line in Figure 5) from both figures reveal that vaporization starts at 128.7 °C and vaporization is continued till 180°C. A temperature glide of ~51° C is noted from saturated liquid to saturated vapor. Around 150°C, nearly 80% of vapor is contributed by water at the quality of 0.5 (50% of the total mixture is in vapor condition).

Simple Laboratory Scale Experimental Analysis:
Figure 6 shows the engine coolant degradation experimental setup. A Standard Rotating Pressure Vessel Oxidation Test (RPVOT) was performed. A small quantity (50 ml) of the working fluid was subjected to high temperature (165°C) for 30 minutes, 180 minutes and 24 hours residence time at three different pressures (16 bar, 12 bar and 10 bar). Samples were kept in the high temperature oil bath and maintained at 165°C throughout the experiments. These samples were analyzed through standard coolant testing protocols prescribed by coolant manufacturers. The results are shown in Figure 7. Table 2 shows the laboratory scale test results at 3.5 bar and 165°C conditions and corresponds to a working fluid quality of 0.4. Although trace levels of ethylene-glycol decomposition (glycolate and formate) were detected, corrosion inhibitors (2-EH and sebacic acid) appeared stable (within measurement error). Hence, based on these laboratory scale test results the team concluded that the coolant can be utilized as the WHR working fluid.

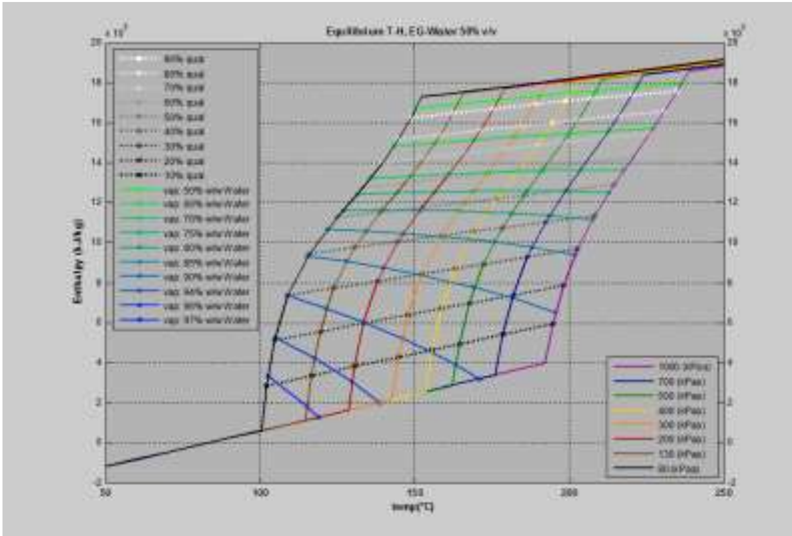


Figure 4. Enthalpy-Temperature diagram for water-glycol mixture

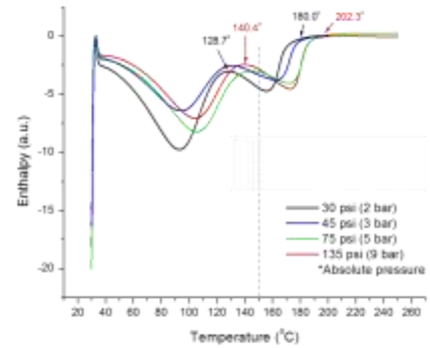


Figure 5. Enthalpy diagram

Figure 6: Coolant Degradation Test Setup (Rotating Pressure Vessel Oxidation Test)

Table 2: Thermal Stability Test at 3.5 bar.

			Degraded EG			Corrosion Inhibitors	
Temp	Pressure	Duration	Glycolate	Formate	pH	2-EH	Sebacic
165 °C	RP/3.5 bar	3 h	48 ppm	28 ppm	8.2	1.98%	0.23%
165 °C	RP/3.5 bar	24 h	260 ppm	89 ppm	7.6	1.93%	0.17%

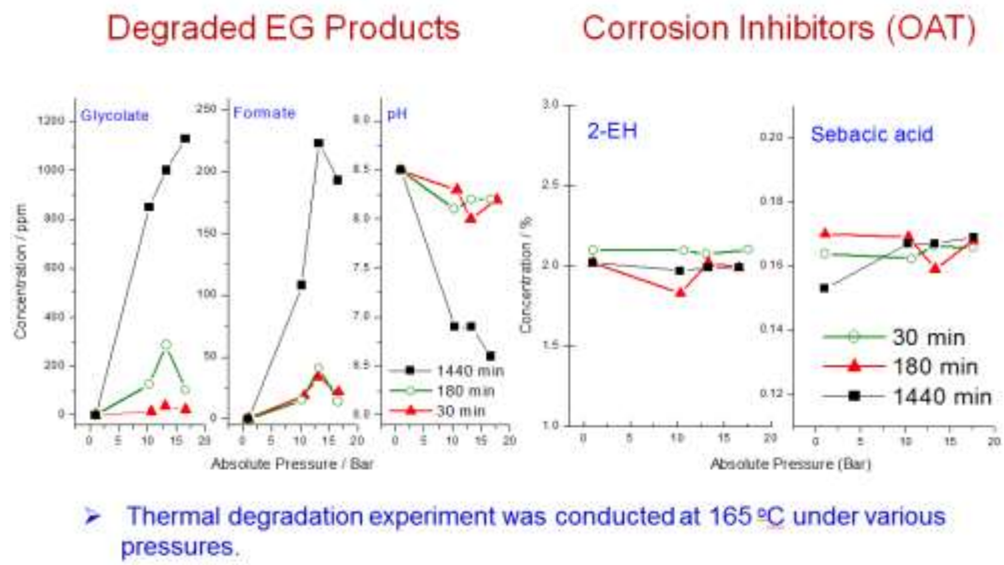


Figure 7: Coolant Degradation Analysis.

Real Environment Experiments:
The light duty diesel engine (1.9L) WHR test setup was developed to quantify the engine coolant degradation in a real operating environment and to validate the two-phase performance in the expander. This engine testing helped us understand the ethylene-glycol and corrosion inhibitor decomposition rates when the engine coolant system is incorporated into a waste heat recovery loop. It also helped identify the risks in developing the control system. Figure 8 shows the 1.9L WHR test setup. Test results were used to validate the previously discussed simple laboratory scale experiments.

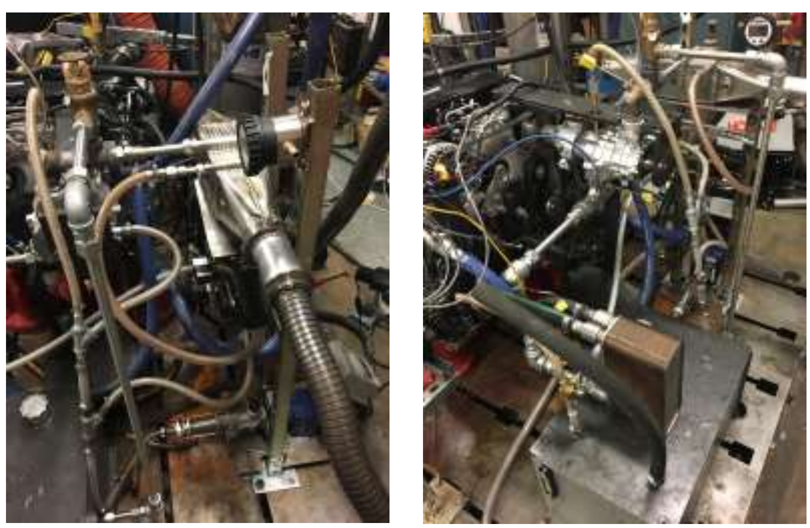


Figure 8: 1.9L Diesel Engine Test Setup at Mississippi State University for Coolant Degradation Study

Shell analyzed the real environment tested engine coolant samples from the 1.9L WHR test setup. Coolant samples were taken at 10 running-hour intervals with the intention of analyzing the coolant degradation from WHR functionality. Figure 9 summarizes the analysis findings of the sample collected from engine test cell. The corrosion inhibitors (sebacic acid and 2-EH) appeared stable and only trace amounts of ethylene-glycol decomposition (glycolate, formate, acetate, and oxalate) were detected. These results support the feasibility of using the engine coolant in the WHR loop.

Visual Testing							
#	Foam	Color	Oil	Fuel	Magnetic Precipitate	Non-Magnetic Precipitation	Odor
1	None	Cloudy Red / Orange	Moderate Other	None	None	None	Other

Basic Testing									
#	Freeze Point (°F)	Boil Point (°F)	Antifreeze Percent (%)	pH Waters (pH)	Total Hardness (ppm)	Nitrite (ppm)	Specific Conductance (null)	SCA Number	Carboxylic Acid (Pass / Fail)
1	-38	226	52	7.6	7	0 - IC	3750	0.0	

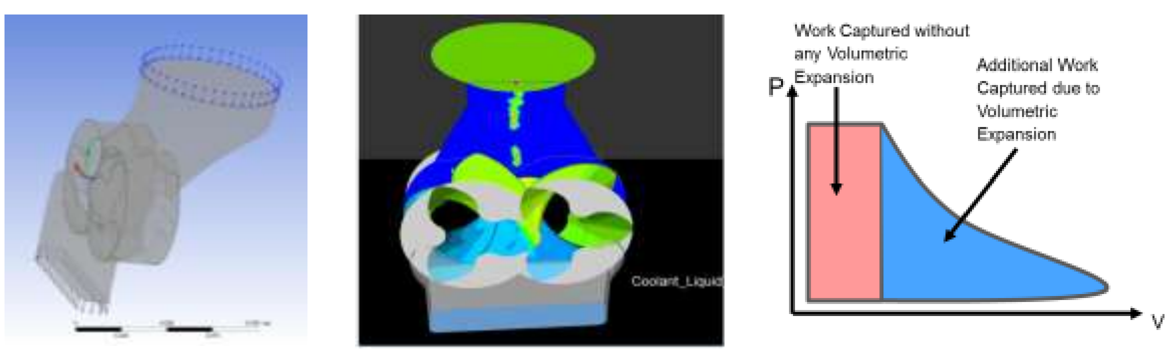
Additional Testing																		
Sample #	Sulfate	Chloride	Nitrate	Glycolate	Formate	Acetate	Oxalate	Adipic Acid	BZT	TTZ	Benzoic Acid	Sebacic Acid	MBT	P-toluic Acid	2-EH	Octanoic Acid	Dodecanoic Acid	Total Dissolved Solids
	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm	ppm
1	4	5	22	35	38	88	119	NA	<10	844	<20	2204	<10	<20	18536	1489	NA	1388

Figure 9. Engine Coolant Degradation Test Results

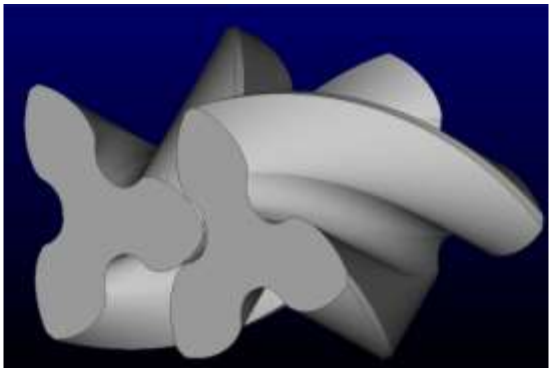
WHR Component Design and Development
Heat exchangers and expander are the major components that need to be designed and developed in this program.

Expander Design and Development:
The Eaton team evaluated a Roots expander to determine its feasibility for use within the ARC system. Expander inlet and outlet boundary conditions in addition to the required expander efficiency were derived from the WHR analysis. CFD analysis using a traditional Roots design was performed and showed that the design was unlikely to meet the program performance objectives due to low efficiency resulting from expansion ratio limitations and high leakage. The team concluded that the Roots expander works is ideally suited for low pressure and high volume flow rate conditions.

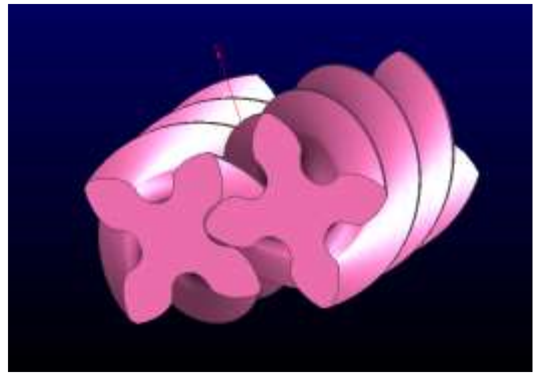
Several approaches were explored to improve the Roots expander's efficiency for the ARC system including liquid injection, altering the design to obtain internal expansion and alternative component materials to minimize clearances due to thermal growth (refer to Figures 10 and 11). All the approaches showed an increase in efficiency but the total improvement did not meet the target efficiency of 60%. Hence the Eaton team proposed implementing an alternative expander for ARC program and the DOE team approved this scope change.



a. V100 Roots Expander **b. Liquid Injection in Roots** **c. Internal Expansion**
Figure 10. Roots Expander and Efficiency Challenge



Existing Roots Profile



Modified Roots Profile

Figure 11: Roots Expander Rotor Profile.

Alternative Expander Selection for ARC Concept:

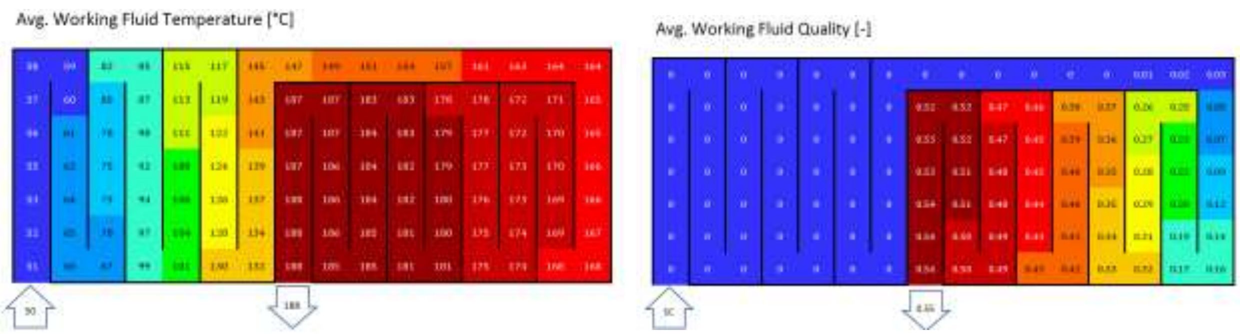
Four volumetric expander types were analyzed and investigated based on ARC requirement as potential fits for the project. The expander tradeoffs were outlined and based on the selection criteria the vane expander was selected as the prime path for the project.

Table 3: Expander Technologies

Technology	Pros	Cons
Piston	• High expansion ratio capability • Eaton's familiarity (hydraulic piston pumps)	• Not a good fit for two-phase flow • More complex with high part count, not a cost effective solution.
Scroll	• High expansion ratio capability	• New to Eaton ilable in WHR market
T		
Va	(Hydraulic pumps and motors) • Differentiator in the WHR market • Very simple, & cost effective solution • Identified a US based small R&D startup company worked in high temperature Vane expanders	with high vane expanders.

Instead of Table 3 consider placing the concept tradeoff matrix we completed in the report.

Heat Exchanger Design and Development:
Baseline results from WHR analysis helped define the heat exchanger specifications for a PACCAR MX-13 13L diesel engine WHR system.



Exhaust Boiler

Figure 12: Heat Exchanger Design and Development.

Table 4: Heat Exchangers Design Specification.

Exhaust	m (g/s)	93.0	93.0	93.0
	λ (-)	1.6	1.6	1.6
	T_in (°C)	430	430	430
	P_in (kPa)	150	150	150
	T_out (°C)	110	110	110
	ΔP_max (kPa)	2.83	2.83	2.83
Working Fluid	m (g/s)	19.37	15.49	13.37
	T_in (°C)	~50	~50	~50
	T_out (°C)	182.1	197.8	208.6
	P_out (kPa)	500	500	500
	x_out (-)	0.5	0.75	1

Exhaust Boiler Specification

Cooling Circuit	m (g/s)	1450	1450	1450
	T_in (°C)	40	40	40
	T_out (°C)	45	45	45
	P_in (kPa)	254	254	254
	ΔP_max (kPa)			
Working Fluid	m (g/s)	19.37	15.49	13.37
	T_in (°C)	109	147.9	157.8
	P_in (kPa)	110	110	110
	x_in (-)	0.51	0.74	0.96
	ΔT_sat_out (°C)	~50	~50	~50

Condenser Specification

Table 4 shows the design specifications for the exhaust heat recovery heat exchangers for 13L diesel engine WHR test rig. The heat exchangers were designed for selected steady state points specified by PACCAR. Working fluid quality and temperature profile for EGR boiler is shown in Figure 12. This program targets a quality no greater than 0.5 at the boiler outlet to avoid ethylene-glycol vaporization and corresponding coolant degradation issues. However, heat exchangers are designed for conditions including varying quality levels between 0.4 to 0.5.

Exhaust Gas Recirculation Boiler (EGRB)
Two different designs of EGRB were analyzed. The first design was a drop-in replacement for the current EGR cooler (~600mm length). This resulted in exhaust gas outlet temperatures 10 to 20°C higher than the target requirements. The higher EGR temperature was not acceptable and hence a second EGR boiler design was evaluated. This second design was ~900mm long (1.5 times longer than the first design) with an overall core cross section of 100 x 100mm. The working fluid and EGR gas temperature for the two different designs at two different operating conditions are shown in Figure 13. The longer second design reduced the exhaust temperature gap to 10°C.

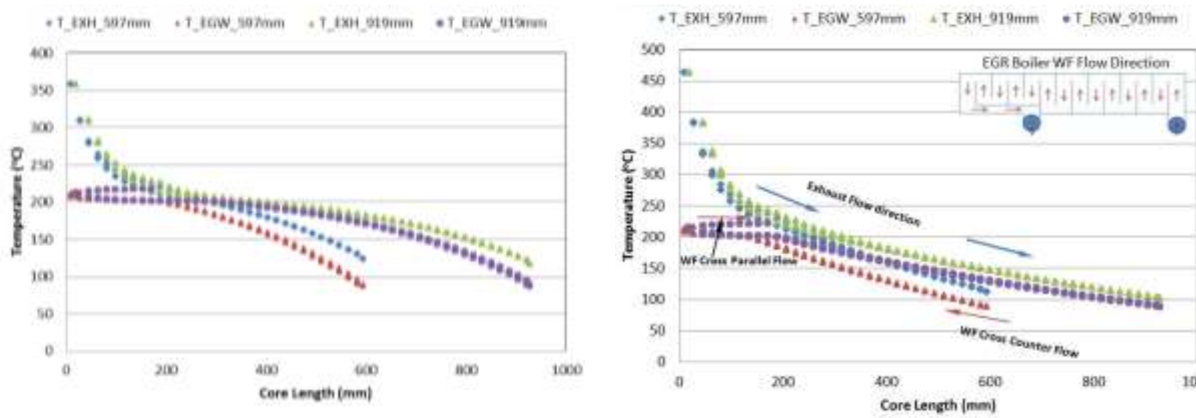


Figure 13. Heat Exchanger Fluid Temperatures Vs Length

Tailpipe Boiler (TPB) or Post Turbine Boiler

Vehicle heat rejection limitation imposed a major constraint on recovering tail pipe exhaust enthalpy. However, preliminary estimation on tailpipe heat recuperation assumption led to a design of tailpipe boiler with 197mm (H) x 250 mm (W) x 148 mm (L). Phase 1 efforts ended before the tail pipe heat exchanger design was completed.

Heat Transfer Correlation Development:

During phase 1 period Argonne National Laboratory (ANL) looked into the details of upgrading the ANL boiling heat transfer test facility to accommodate the fluid parameters of the EGR boiler for the WHR system. The major change to the ANL experimental facility was related to the EGR boiler pressure and glide temperature. The original ANL experimental facility was limited to 195°C by the facility condenser. A replacement condenser was sized and located to accommodate 232°C. Other smaller facility components were also identified for replacement to accommodate the pressure and temperature levels of the EGR boiler. Figure 14 shows the test setup developed by the ANL team for two–phase correlation development and utilization in phase 2 efforts.

Also during the reporting period, ANL looked closely into the fluid parameters for 50/50 ethylene-glycol and water as supplied by the NIST computer code REFPROP. Parameters from the code were compared to published data at low pressure, and parameters were looked at closely near the saturation point. The results generally showed good agreement with the data, and the sensitivity to pressure changes of the order of 15% were small (less than 2.5%) as expected for vapor density.

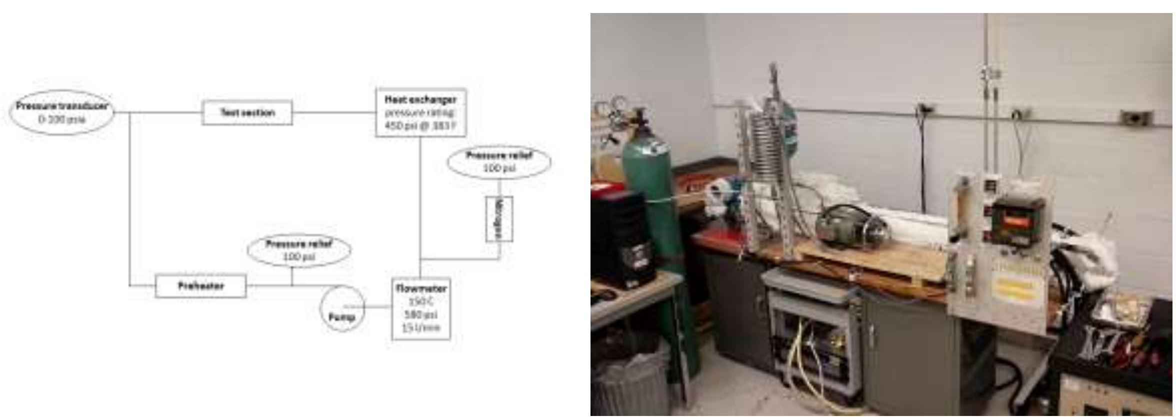


Figure 14. Two-Phase Heat Transfer Correlation Development Test Rig

ARC WHR System Model Development

Expander Model Development:

Positive displacement expanders are characterized by a fixed internal volume ratio (ratio of the chamber volume at discharge beginning to the chamber volume at suction closure). Because of the internal volume ratio, the theoretical internal (or indicated) specific work of an expander (e.g., scroll, screw, vane, etc.) can be computed as the sum of isentropic work and work at constant volume. In order to evaluate the influence of the expander volume ratio on the performance of the ARC running with a 50-50 ethylene-glycol plus water mixture, a parametric study was carried out.

Effect of internal volume ratio

The condensing pressure was set to 110 kPa and the quality of the mixture at the expander inlet was fixed at 0.5. The EGR inlet and outlet temperatures were 430ºC and 98.7ºC, respectively. The tail pipe temperature conditions were 294.4ºC and 110ºC. The internal volume ratio of the expander was varied between 1 (Roots-type) to 7 (Vane-type). Three expander inlet pressures were considered to represent the target application: 1000 kPa (rp = Pressure ratio = 1000/110=9.09), 1500 kPa (rp=1500/110=13.63) and 2000 kPa (rp=2000/110=18.18). REFPROP was used to obtain the thermodynamic properties of the mixture. The results of the calculations are shown in Figure 15. Results show that expander power output tripled with an internal volume ratio of 7 (vane expander) when compared to an internal volume ratio of 1 (Roots Expander).

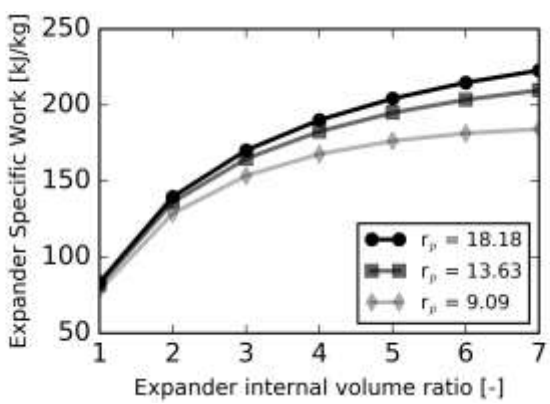


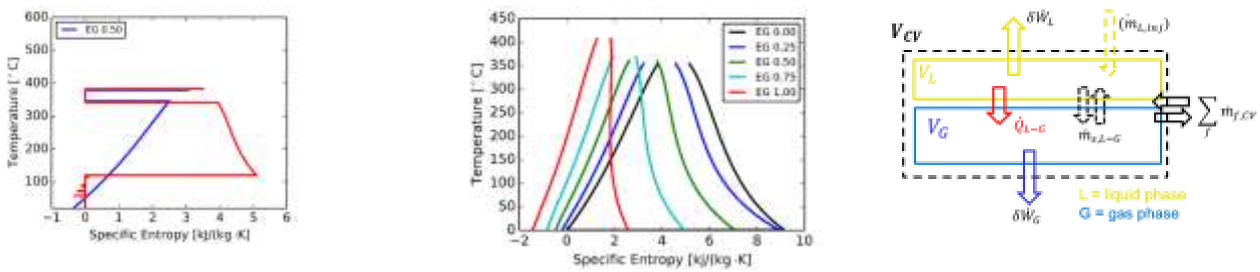
Figure 15. Effect of expander volume ratio on the specific work output

Working Fluid Properties:

The thermo-physical and transport properties of the ethylene-glycol (EG) plus water mixture was limited during early stages of this program. The team worked with NIST and resolved this issue. The original REFPROP file of EG failed above 300°C and below 130°C, as shown in Figure 16.a. With an improved EG file, Figure 16.b shows T-s diagrams obtained at different concentrations of EG plus water. The current working fluid is a binary mixture and in order to obtain the concentrations in both the liquid and vapor phases at any point during the expansion process, evaporation and condensation, vapor-liquid equilibrium (VLE) diagrams are extremely critical.

Control Volume:

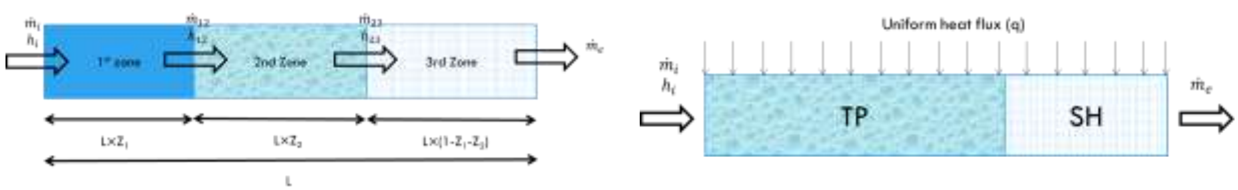
The thermo-physical and transport properties are of key importance to develop a thermodynamic model of the expansion process. The chamber model of a positive displacement machine is based on a set of differential equations representing the conservation of mass and energy applied to a control volume. In the case of a mixture, two formulations can be considered: homogeneous model and heterogeneous model. The homogenous model is based on the assumption that the liquid and vapor are at equilibrium at any moment. The heterogeneous model formulation allows the liquid and vapor phases to be at non-equilibrium conditions and therefore the temperature in each phase can be different and the concentrations can differ from those at equilibrium conditions. The pressure is considered to be the same in both phases. In order to develop such formulation, a generalized control volume (CV) that includes two separate phases (L: liquid and G: vapor) plus heat and mass transfers between the phases are considered and depicted in Figure 16.c. The geometric control volume (GCV) which represents a general working chamber of the expander is divided into two sub-control volumes, VL and VG, one for each phase. Due to this formulation, the thermodynamic state of both phases are evaluated by applying the conservation of mass and energy to each phase. The mathematical formulation also includes models for the heat transfer between the phases and mass transfer. Furthermore, the work rate contribution of each phase is estimated as well as the flows in and out the CV. Liquid injection for lubrication, is also included into the model.



16.a: Initial T-s diagram of Water/EG [50-50] with low accuracy property routines.	16.b: Sweep of T-s diagrams of Water/EG for different concentrations.	16.c: Conceptual schematic of a generalized control volume.

Figure 16. Expander Model Development and Working Fluid Properties

Heat Exchanger Model Development:
Development of an accurate but computationally efficient heat exchanger model is a key step to handle the ORC transient conditions. The finite volume method (FVM) and moving boundary method (MB) are popular approaches for dynamic heat exchanger modeling. While the MB segments heat exchanges depending on thermodynamic phase of refrigerant, i.e. sub-cooled liquid, two-phase and super-heated vapor and moves control volumes as the length of each phase changes (Figure 17.a), FVM divides heat exchangers into a number of fixed control volumes (Figure 17.b).



17.a: A schematic diagram of Moving Boundary Method 17.b: A schematic diagram of Finite Volume Method

Figure 17. Heat Exchanger Model Development

Figure 18 shows the simple comparisons of the MB formulation with the FVM (16 nodes) for an evaporator having two-phase (TP) and super-heated (SH) zones were performed.

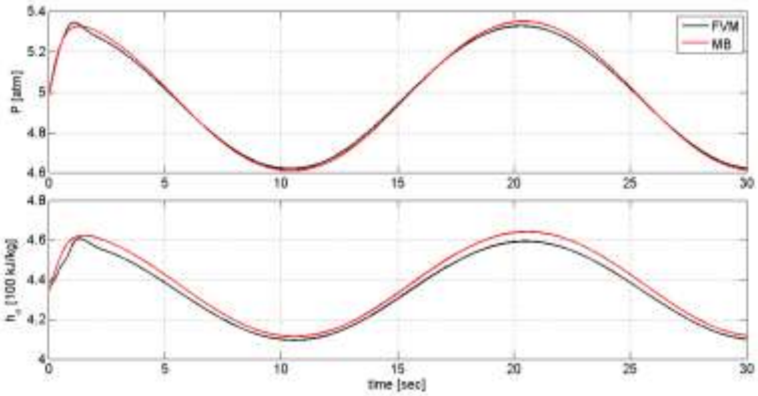
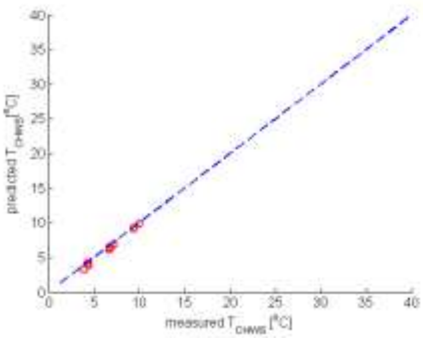
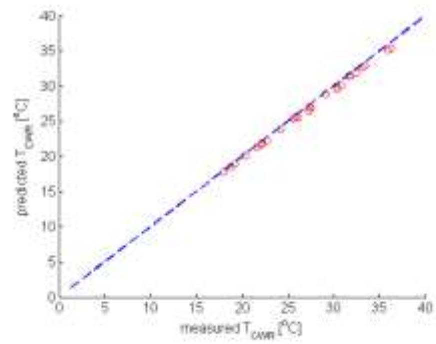


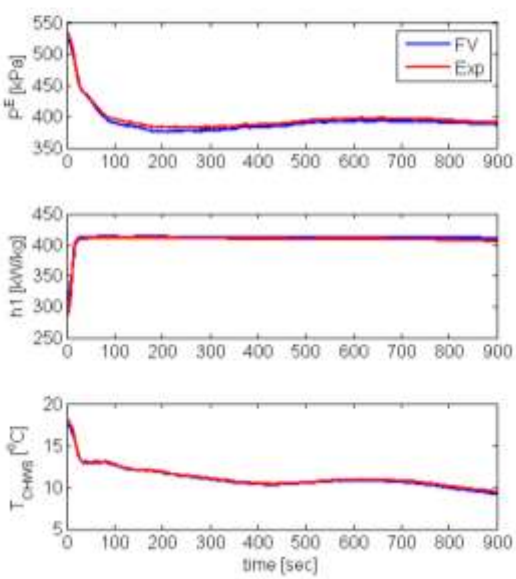
Figure 18: Sample result comparisons between moving boundary and finite volume methods.



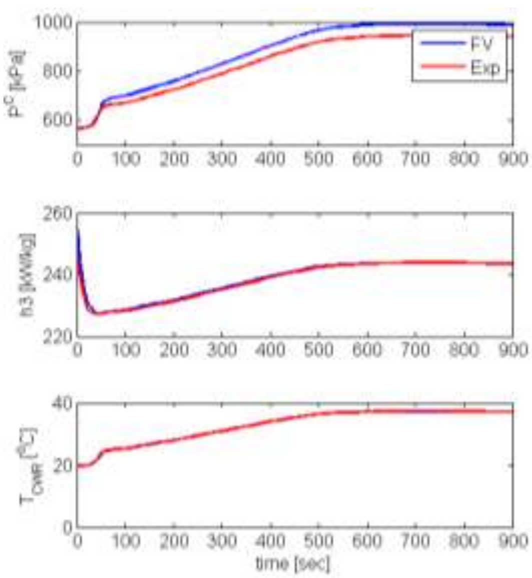
Steady evaporator model



Steady condenser model



Dynamic evaporator model



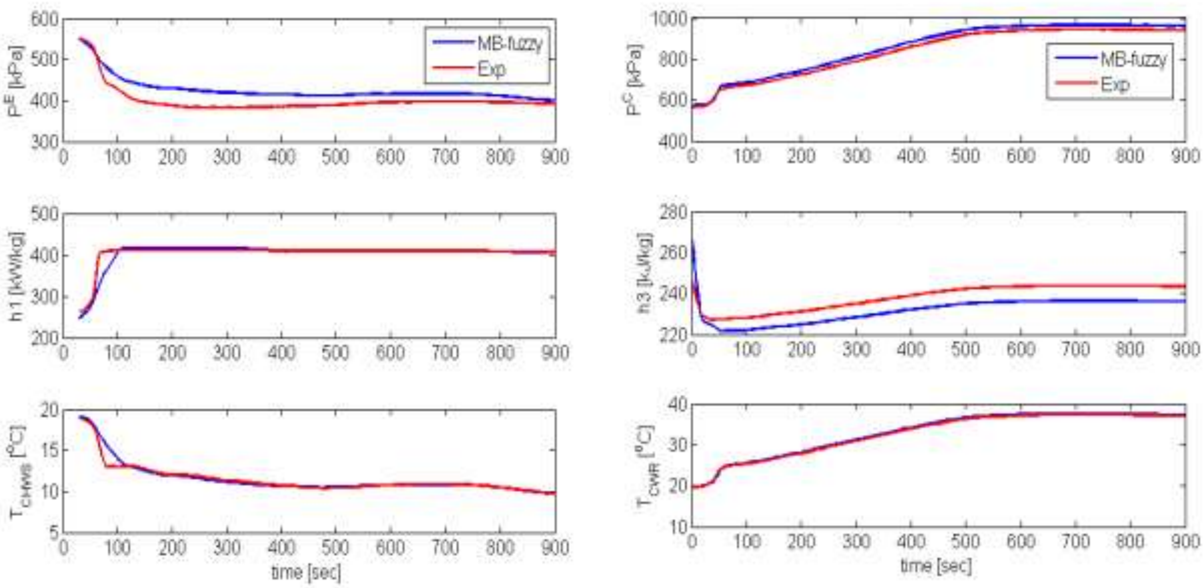
Dynamic condenser model

Figure 19. FVM Heat Exchanger Model Validation with Experimental Data

Heat Exchanger Model Validation:

Heat exchangers influence the transient behavior of ORC systems significantly. The finite volume method (FVM) and moving boundary method (MB) modeling approaches developed in ARC program were validated with experimental data during phase 1 period. Standard (Matlab) equation solvers were used for steady and transient HX models. The steady-state model predictions are very close to the measurements. Figure 19 shows the comparisons of FV HX models to measured values.

A moving boundary modeling (MB) approach for a condenser and evaporator was validated using available measurements (Figure 20). In particular, a novel algorithm that switches moving-boundary was developed. It was designed to eliminate discontinuity introduced by the phase-dependent MB formulations. Shell-and-tube evaporator and condenser models using the MB with the fuzzy switching algorithm were compared with available experimental measurements. They showed very good agreement over a large transient period. This MB mode-switching algorithm will be useful to simulate transient responses for the ORC system under large heat load variations.



MB dynamic evaporator model MB dynamic condenser model

Figure 20. MB Heat Exchanger Model Validation with Experimental Data

Affordable Rankine Cycle Performance Prediction:

ARC system analytical investigation shows the impact on fuel economy at four different static operating conditions. Fan losses are not accounted for in this analysis. The maximum working fluid temperature was capped by preserving coolant life. The coolant thermal decomposition temperature is 240°C. Hence the working fluid out from heat exchanger was set to 220°C, with a margin of safety of 20°C. The quality of the working fluid at the expander inlet was limited to more than 0.5 to minimize the ethylene-glycol vapor formation. These two parameters (temperature and quality) help fix the expander inlet pressure and required working fluid mass flow rate. Expander outlet pressure was dictated by the existing engine coolant circuit. This WHR analysis helps us understand system level performance.

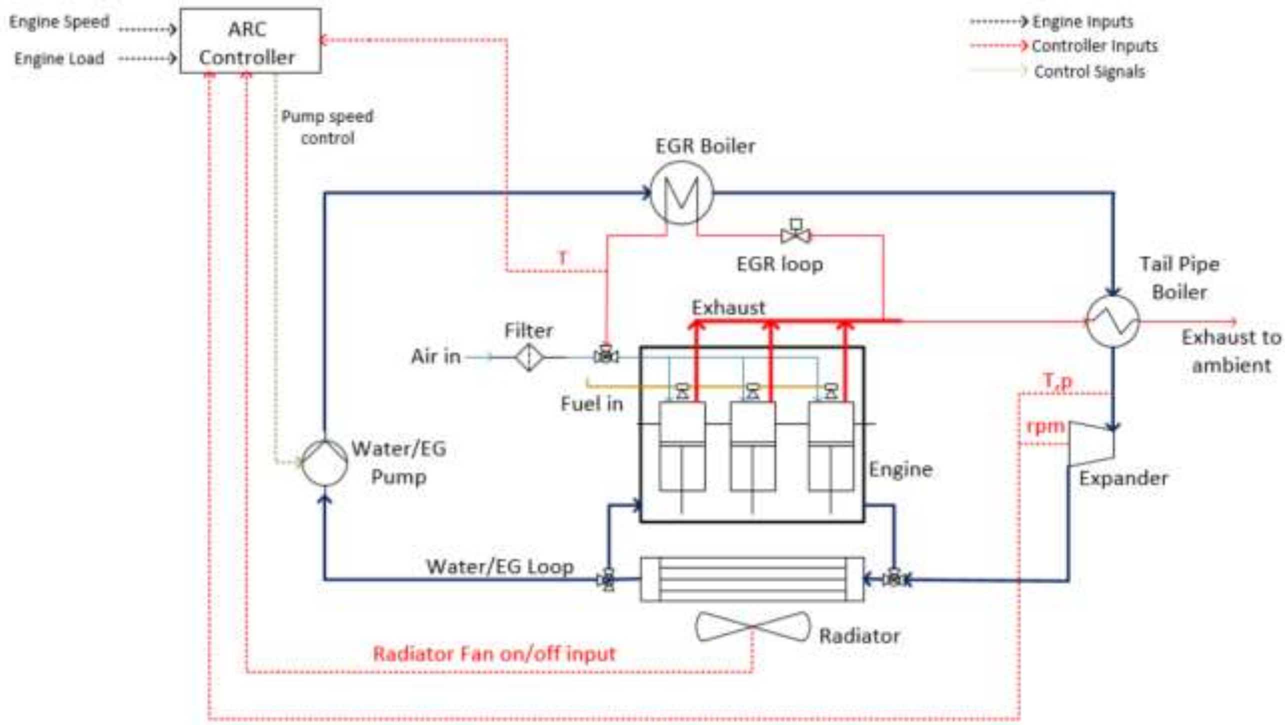


Figure 21. ARC WHR System Architecture

Based on the system architecture shown in Figure 21, a steady-state cycle model was developed to investigate the performance of the ARC system. The heat inputs were determined from the engine operation. Furthermore, constraints on the maximum temperature of the coolant, return gases to the engine, and tail pipe exhaust exit temperature are imposed to ensure safe operation of the engine and control emission. The total heat rate available at the EGR can be quantified as:

$$\dot{Q}_{\mathrm{EGR,in}} = \dot{m}_{\mathrm{EGR}}\left(h_{EGR,in} - h_{EGR,out}\right) \quad (1)$$

where $h_{EGR,in}$ and $h_{EGR,out}$ are the inlet and outlet enthalpies of the EGR which are fixed by the engine operating conditions. A heat exchanger effectiveness is applied to obtain the heat recovered by the coolant. The heat rate available from the exhaust tail pipe is defined analogously to (1):

$$\dot{Q}_{\mathrm{EHX,in}} = \dot{m}_{\mathrm{EHX}}\left(h_{EHX,in} - h_{EHX,out}\right) \quad (2)$$

Both expander and pumps are modeled with a constant isentropic efficiency. The heat rejected by the condenser (i.e. radiator) is calculated as

$$\dot{Q}_{\mathrm{cond}} = \dot{m}_{\mathrm{water/EG}} \Delta h_{radiator} \quad (3)$$

where $\Delta h_{radiator}$ is the specific enthalpy difference across the radiator. Note that the condensing pressure is imposed by the radiator. At each engine operating condition, the spare load available in the radiator is obtained and checked against the needed condensing heat rate.

The cycle performance and the benefits of the ARC system are quantified by defining an ORC thermal efficiency and Break Power (BP) improvement as:

$$\eta_{\mathrm{ORC,net}} = \frac{\dot{W}_{\mathrm{ORC,net}}}{\dot{Q}_{\mathrm{tot,in}}} = \frac{\dot{W}_{\mathrm{exp}} - \dot{W}_{\mathrm{pump,1}} - \dot{W}_{\mathrm{pump,2}}}{\dot{Q}_{\mathrm{EGR,in}} + \dot{Q}_{\mathrm{EHX,in}}} \quad (4)$$

$$BP\ improvement = \frac{\dot{W}_{\mathrm{ORC,net}}}{\dot{W}_{\mathrm{engine}}} \tag{5}$$

The cycle model has been exercised with several engine operating modes to characterize the potential efficiency improvements with the ARC-based WHR. The nominal engine conditions considered are summarized in Table 5. The inlet and outlet expander pressures have been set equal to 1200 kPa and 150 kPa, respectively. The upper limit of the pressure is enforced by the current expander technology employed. Higher pressure ratios and the impact of expander internal volume ratio were showed in Figure 16. At first, pump and expander isentropic efficiency values are set equal to 0.6. A EG plus water mixture having mass fractions of [0.5-0.5] was used to carry out the calculations. The results are reported in Table 6. An example of ARC thermodynamic cycle is shown in Figure 22.
Note that as the heat available to the WHR system increases, the net power output also increases. However, due to the limitation of the maximum operating temperature of the EG plus water mixture, the WHR efficiency has a plateau. Since the expander is the key component to achieve higher BP improvement, a parametric study was performed to evaluate the impact of improving the expander isentropic efficiency on both cycle efficiency and BP improvement. The results are shown in Figure 23(a) and Figure 23(b). By increasing the expander isentropic efficiency from 0.6 to 0.8, the maximum BP improvement was obtained under engine operating point #4.

Table 5. Selected Operating Condition for Performance Evaluation

Parameter	*#1*	*#2*	*#3*	*#4*
$T_{EGR,in}$ (ºC)	358.4	464.2	543	661
$T_{EHX,in}$ (ºC)	274.4	326.4	354.7	389.1

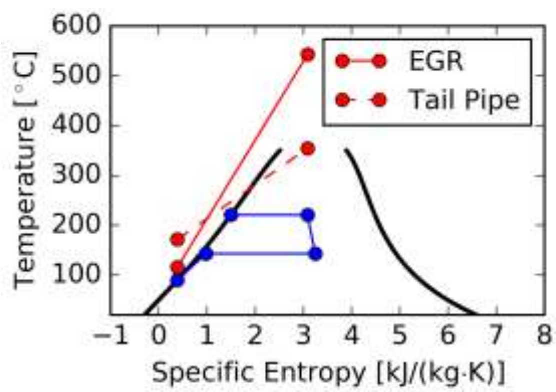


Figure 22. TS Diagram of ARC WHR Cycle

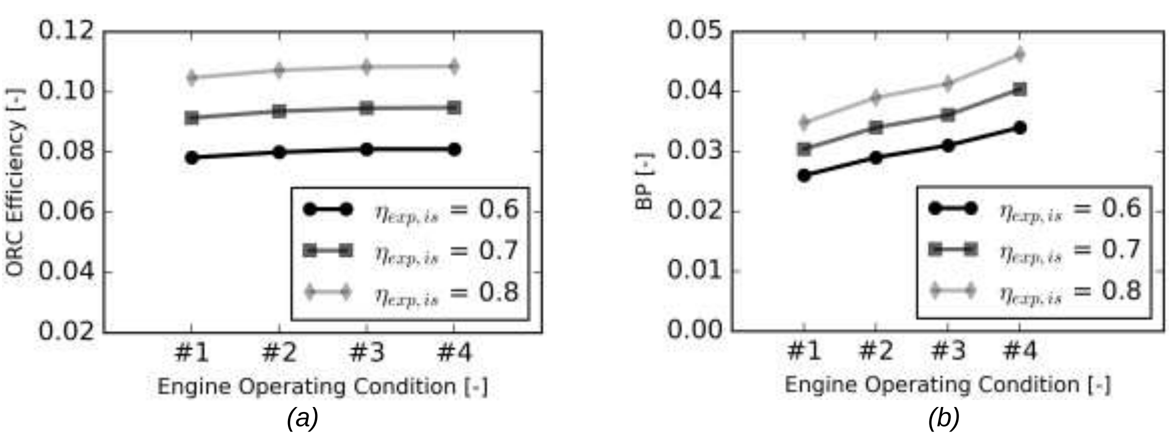


Figure 23. Effect of Expander Efficiency (η_(exp,is)) on (a) ORC Efficiency and (b) ARC BP Improvement

Table 6. ARC System Performance

Parameter	***#1***	***#2***	***#3***	***#4***
Expander Isentropic Efficiency (%)	*60*	*60*	*60*	*60*
Predicted Net Output Power (kW)	2.014	4.520	7.189	10.51
Predicted ORC System Efficiency (%)	7.82	8.0	8.1	8.1
Total Heat Input to System (kW)	25.76	56.44	88.90	129.7
Brake Power improvement (%)	2.61	2.91	3.35	3.44
Fuel Economy Improvement (%)	2.5	2.83	2.99	3.32
Expander Inlet Mixture Quality (-)	0.433	0.501	0.518	0.518

Table 7. ARC System Performance with Heat Rejection Constraint

Parameter	***#1***	***#2***	***#3***	***#4***
Expander Isentropic Efficiency (%)	*60*	*60*	*60*	*60*
Predicted Net Output Power (kW)	1.9	2.8	2.4	3.3
Predicted ORC System Efficiency (%)	7.5	7.8	7.9	7.9
Total Heat Input to System (kW)	25.76	36.29	30.42	41.93
Brake Power improvement (%)	2.61	1.82	1.03	1.09
Fuel Economy Improvement (%)	2.5	1.8	1.0	1.0
Expander Inlet Mixture Quality (-)	0.37	0.439	0.456	0.456

Table 6 and 7 show ARC system performance analysis without heat rejection limitation and with heat rejection limitation using EGR boiler and tail pipe heat exchanger as heat sources. ARC system resulted ~1.5% fuel economy improvement when appropriate weighting factors were applied for operating conditions in table 7.

ARC System Constraints:
ARC system performance was constrained by working fluid (engine coolant) properties, architecture (unable to recover engine block heat loses, CAC and oil cooler) and heat rejection limitation from existing radiator size. Figure 24 depicts the ARC system performance loss walk from the above mentioned constraints.

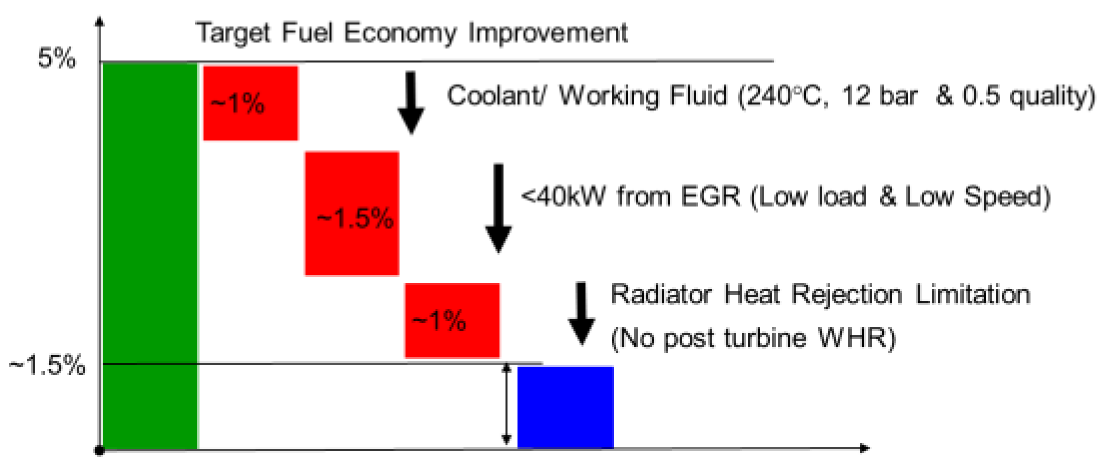


Figure 24. ARC System Performance Loss Walk

3 CONCLUSION

ARC program phase 1 efforts indicate that ARC system is capable to deliver ~4% fuel economy improvement but would require an expander operating at 80% efficiency and a vehicle without a heat rejection limitation. By utilizing only the EGR boiler and a limited tailpipe exhaust heat recovery, only ~1.5% fuel economy improvement (Table 7) is predicted. Therefore, utilizing tail pipe exhaust energy is required to meet the 5% target. However, adding a tail pipe heat exchanger will significantly increase the system cost and the fuel economy benefits at higher operating loads are reduced due to increased vehicle cooling and fan loads.

- Original system architecture does not meet the 5% fuel economy target and is constrained by the engine coolant upper specification and vehicle's heat rejection limit (based on WHR analysis for specific operating conditions)
- Model predicts the system architecture will achieve ~1.5% fuel economy improvement
- 1.5% fuel economy is insufficient to justify the increment WHR system cost
- It has been decided to stop the ARC program with phase 1 efforts by 31 March 2017