PROJECT PROGRESS REPORT

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DENALI COMMISSION BY THE ALASKA CENTER FOR
ENERGY AND POWER

PROJECT TITLE: Optimizing Heat Recovery Systems for Power Generation in Rural Alaska

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PROJECT OBJECTIVE: The objective of this project is to conduct laboratory performance test on an ORC system and performance and economic comparison of two different ORC systems in capturing waste heat from diesel generators for rural applications.
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INTRODUCTION:
The objective of this project has four folds. The first is to improve the efficiency of the diesel power plant of a village by 10%, which also results in a reduction of CO₂ by 2.2 lb for each gallon of diesel fuel saved, through the use of a 50kW Organic Rankine cycle (ORC) system to recover waste heat contained in diesel engine jacket water and exhaust. The second is to evaluate feasibility, operation and maintenance requirements, and payback time of applying the ORC system on the village power plant mentioned above. The third is to develop guidelines for ORC system selection, operation, and maintenance and to evaluate the potential impact of applying waste heat ORC systems on rural Alaska economy, fuel consumption, and green house gas reduction. The fourth is the performance and economic comparison of two ORC systems. One ORC system is the 65 kW system (an upgraded version. The original system is 50kW.), which uses screw expander, comes under emerging technology. The second ORC system is the Pratt & Whitney (P&W) 250 kW unit which user radial turbine belongs to the category of well developed technology.

CURRENT PROJECT STATUS AND EXPECTED DELIVERY DATE:
The approach of this project includes eight parts, which are procurement, modeling and simulation, installation and instrumentation, durability test for system reliability, controlled test for system performance verification, performance and economic comparison of two ORC systems (one is 65kW and the other is 250kW), and report writing and dissemination. Details of the expected delivery date, tasks of each part and its current status are listed in the following paragraphs.

After the last Quarterly report, the project team at UAF was in regular contact with the power unit (Green Machine) manufacturer (ElectraTherm) and received the Green Machine in March 2011. The UAF team contacted two reputable local contractors for system installation and obtained quotations in late April 2011. The estimated cost for installation was about $150,000, which was far beyond the available budget and the expected costs from both the ET engineers and the UAF team. One of the major reasons of the high cost was resulted from the need of following local regulations for boiler, plumbing system, and electrical system installations. In order to keep the project going, Tom and Jack of the ACEP engineers have continued searching for different ways and sites for installation. The new experimental site selected is an available space inside the UAF Power Plant. The site has steam and cooling water supplies for heating and cooling needs of the Green Machine test. The selected new site is much smaller than the original site and is estimated enough for the installation of the test system (without boiler and storage tank of the cooling tower). Project staff believes that the new site has the advantage in saving installation and testing cost due to the availability of the onsite heating and cooling media supplies, which reduces much of the concerns in following local regulations for installing boiler, cooling tower, and electrical up-loading system.

Project Time Line:
The expected delivery date for the ORC power unit and related equipment/supplies is the same as given in the previous report (late December 2011). Detailed revised timeline is given in Appendix I.

Procurement:
Considering the new test site, major equipment, which are the devices absolutely needed for testing and measurement, include the 65kW ORC system, a steam to hot water heat exchanger, a heating flow pump set (VFD and 250GPM), a by-pass system for GM emergency stop, refrigerant tanks, a 3-way valve for temperature control on heating side, an automatic on/off control valve for steam source cutoff, a manually operated valve for cooling flow rate control, a 3-way valve for temperature control on cooling side, 3 temperature mixers, and 2 Btu meters. Final check (on June 23rd) with the engineers of the ORC system manufacturer has been conducted to confirm the appropriateness of the devices listed above for achieving the project objective.

For the new testing set up, extra major components, which need to be purchased, include a steam to hot water (steam/water) heat exchanger, a manual control valve to control the cooling water flow rate (heat sink) on cooling side, one temperature mixer, a steam automatic on/off control valve, and a 3-way temperature control valve on hot side. All the existing major equipment has been moved from the Bidwell building to the new experimental site (UAF power plant building). Purchases are in process for the added equipment needed for the new test setup.

Piping control system is required to connect the steam/water heat exchanger to the ORC system and the cooling sink to the ORC system. Other auxiliary components, which need to be added to the detailed line diagram and purchase list, include rubber hose, expansion tank, adapters, etc. The detailed line diagram will be completed in a couple of weeks and components will be purchased according. On-shelf measurement devices (e.g. measurement devices for temperatures, pressures, devices related to electrical power measurement and remote monitoring), are also being purchased.

**Test System Modeling and Simulation Methodology:**
The purpose of modeling and simulation is to make the model a tool for selecting village diesel engines by matching performance characteristics, load pattern, and environment conditions of the diesel engine to the performance of the ORC system. The model will also provide information for optimal application of the ORC system for efficiency and net gain in power including generated power and parasitical power.

The model includes three components: heat source loop, heat sink loop, and the ORC system. Fluid used in heating and cooling loops is water.

**HEAT SOURCE**
The physical heat source loop for the new test site is expected to include a hot water source from the steam/water heat exchanger, a VFD pump, and pipes and fittings. Other components for measurement and control are also included. Through the pipes, the hot fluid interacts with the heat source heat exchanger of the ORC system. The loop has controllability in temperature and flow rate of the output of the steam/water heat exchanger. Information of hot water and pump power consumption corresponding to the operation condition is also included in data to be collected. The model constructed includes all the function features of the expected heating loop and storage of all the data related to the data to be collected from the physical heating loop. In addition, the model can be easily modified to cope with different types of heat source and components to be adopted.

**HEAT SINK**
The physical heat sink loop includes a cooling fluid source from the control valve connected to a fire hydrant, heat sink pipes and fittings, and needed measurement and controls devices. Through the pipes, the cooling fluid interacts with the condenser of the ORC system. The loop has controllability in temperature and flow rate of the cooling fluid. Information of fluid power consumption and water consumption corresponding to the operation condition is also included in data to be collected. The model constructed includes all the function features of the expected cooling loop and storage of all the data related to the data to be collected from the physical cooling loop. In addition, the model can be easily modified to cope with different types of heat sink and components to be adopted.

ORC SYSTEM
The physical ORC system to be used for this project is an integrated unit, for which to conduct tests on individual components, without modifying the system (modifying the system may result in losing warranty), may not be possible. Also detailed engineering information of individual components may not be available for the concern of proprietary. In other words, the ORC system may need to be considered a black box of which operation characteristics cannot be altered by the user. A general ORC unit will include at least a pump, an evaporator, a heat to power converter, and a condenser. Other components needed in modeling will depend on the versatility of the physical ORC system to be modeled. The known property of the ORC system to be used includes components of a Rankine Cycle system with working fluid flow rate control for quality adjustment of the working fluid entering the screw expander. Working fluid used for this system is R245fa.

Considering ORC system reliability test and tests of ORC system performance related to system power generation under different heat source and sink conditions, detailed engineering data of individual components may not be needed. For investigation of optimal net power generation (including power generated by the ORC system and parasitic power consumed by cooling loop and/or heating loop), detailed engineering data may also not be needed. Therefore to achieve the objectives of this project listed in the section of Introduction, detailed engineering data of the ORC system components may not be critical. However, if the purposes of the tests include verification of the ORC system performance (e.g. optimal performance constantly) as claimed by the manufacturer and provision of comments to the design of the system, detailed engineering data of components will then become required.

For this project, the plan of the modeling may be divided into two to three stages depending on how feasible and desirable to reverse-engineering the performance parameters of the ORC system is. The first stage is to model the ORC system using simple parameters for system components with quality and flow rate control of the working fluid. The purpose of this stage is to qualitatively understand the effects of operation conditions of heat source and heat sink on ORC system performance and the results will be used for test planning. The second stage is to fit the system parameters using data obtained from a limited number of experimental cases (i.e. heating and cooling conditions). If system simulation results obtained using fitted values of system parameters can qualitatively match experimental results, but without appropriate accuracy, for extra operation conditions, a more complex model will be constructed for detailed modeling as the third stage. In other words, if the preliminary ORC system parameter analysis shows feasibility, more accurate modeling effort will be conducted.
Stage One:
The simple model will include an evaporator, a screw expander, a condenser, and a VFD pump. The model allows the quality of the working fluid entering the expander adjustable via varying the flow rate of the working fluid for optimal ORC system performance. The results will be used for test planning. The expander is modeled by a single efficiency at this moment and will be modified as more information is available in publications and through experimental data, the pump is modeled with varying efficiency based on the operation condition of the pump, and the evaporator and condenser are modeled by their respective flow and heat transfer properties and heat transfer areas. Currently the evaporator is modeled with two sections; one of liquid and the other of liquid and vapor mixture. The condenser is modeled as a single section unit. If heat losses to the atmosphere are found significant, heat losses will also be included in the model.

Ranges of values of parameters used for simulation are based on specifications of the components (e.g. hot water flow rate and temperature limits, pressure and temperature limits of ORC system), properties of the fluids (i.e. heating, cooling, and ORC working fluid), asymptotes performance of sub-components (e.g. heat exchanger performance versus flow rate), etc. Known limits include maximum pressure of the ORC system (150psi), estimated heat source temperature (235F), controlled heat source capacity (2.4MMBtu), flow rates of pumps (250gpm for heating, 375gpm for cooling), and cooling sink capacity (3.0MMBtu recommended by an ORC engineer). Values of ORC components parameters adopted from publications (very limited data available) and conventional application practice [1] for system simulation include expander efficiency (e.g. 0.75), pump efficiency (e.g. 0.75), heat transfer coefficient of evaporator (e.g. 1500 W/m²-K or 265Btu/ft²-F), evaporator area (e.g. 100ft²), heat transfer coefficient of condenser (e.g. 1400 W/m²-K or 247Btu/ft²-F), and condenser area (e.g.200ft²). Some of the values will be adjusted based on the match between the simulation results and the published and experimental data. Heat exchanger simulation model are based on standard practice [2, 3]. Since working fluid (R245fa) property will affect the ORC system performance and temperature and pressure limits to be used for testing, some of the physical properties are obtained and listed in Table 1 for reference. Detailed property of R245fa can be found in NIST document. Results of stage one simulation include system performance (e.g. net optimal efficiencies, expander power) as functions of operation parameters of heat source and heat sink. Results also include effects of sizes of heat exchangers and efficiencies of expander and pumps on system performance. The first stage results will be used to help design test plan of the ORC system.

<table>
<thead>
<tr>
<th>Safety</th>
<th>Vaporization Heat (1atm.)</th>
<th>Boiling T.(1atm.)</th>
<th>Critical Point</th>
<th>Saturation Slope</th>
<th>ODP*</th>
<th>GWP** 100 year</th>
</tr>
</thead>
<tbody>
<tr>
<td>Non-Flammable</td>
<td>197.5 Kj/Kg (355.5 Btu/lb)</td>
<td>14.6C (58.3F)</td>
<td>154C (309.2F)</td>
<td>36.4 bar (527.9psi)</td>
<td>Isentropic</td>
<td>-0</td>
</tr>
</tbody>
</table>

*Ozone depletion potential
** Green house warming potential
(No listed phase out year)

Stage Two:
The second stage is to use measured data from experiment to fit the values of system parameters of the first stage model. Data to be used for fitting include inputs and outputs of the ORC system.
the testing system. Inputs include, at least, inlet temperatures and flow rates of the heat source and heat sink heat exchangers. Outputs include, at least, outlet temperatures of ORC heat exchangers and electrical power generated from the ORC system, of which the effect of mechanical to electrical power conversion efficiency is included. If other measured data become available (i.e. obtained from meters come with the ORC system or meters which are allowed to installed to the ORC system), they will also be used to fit the values of the parameters.

The fitted parameters values will then be used for model simulation of the test system under extra operation conditions of heat source and heat sink. The outputs will be compared with experimental data under the respective testing operation conditions to investigate the discrepancies and to qualitatively determine the feasibility of modeling process. If the model is qualitatively feasible, but results predicted need to be improved, a more detailed model will then be adopted as the third stage of the simulation process.

Stage Three:
A more detailed model of an ORC system may need more than just four major subcomponents (i.e. expander, condenser, evaporator, and pump) as used for stage one modeling. Pipes and control elements (e.g. valves, fittings, and expansion tank) may affect heat transfer and pressure distribution and distribution of liquid and vapor in evaporator, condenser, and expansion tank. Therefore, pipes and control elements may affect performance of working fluid in the system and need to be included in system modeling. In addition, each of the sub components may be further divided into detailed elements to get better simulation results.

An expander may be further divided into its function elements, which include pressure drop element as nozzle effect, vapor cooling down element for working fluid before entering into the screw channel, isentropic expansion element accounting for the working fluid expansion in the channel between the screws, further expansion element due to ambient pressure, leakage element for leakage of working fluid from high pressure region to low pressure region, and existing cooling down element. In addition to the six elements, mechanical loss due to friction also needs to be considered for obtaining net expander work.

Inside the condenser, working fluid may experience three different conditions: vapor phase, mixed phase, and liquid phase. To better represent the performance of the condenser using simple mathematical model, a multi-section model may be needed with each section having its own constant heat transfer properties to approximate the heat transfer behavior of the particular zone. The spaces occupied by liquid, mixture, and vapor may be estimated using the amount of heat transferred. Pressure drop element is also needed for the condenser simulation. Similar to condenser, evaporator can be modeled using the same modeling method.

Pumps used in the testing system can be modeled using their performance characteristics given in the respective manuals and experimental data. Pipes and pipe control units can be modeled using standard method recommended in handbooks for pressure drops. Heat transfer effect of the pipe may be less critical, if all the pipes are short and insulated properly as for the expected experimental setup.

If the third stage is conducted, the model parameters will be fitted and extra simulations will
be performed and results will be used to verify the model.

**Current Test System Modeling and Simulation Status:**
The first stage of test system modeling has been completed and the effects of flow properties of heat source and heat sink on net efficiencies of the ORC system and the test system (i.e. including parasitical power consumptions through heating loop and cooling loop) have been obtained through simulation.

The Organic Rankine Cycle (ORC) used in this simulation model is shown in Figure–1. The working fluid used in this simulation model is refrigerant R–245fa. The saturated liquid refrigerant from the condenser is pumped at high pressure to pre–heater. In pre–heater the refrigerant is heated to saturated liquid state and this saturated liquid then goes to evaporator. In evaporator the saturated liquid is heated to the required saturated liquid–vapor quality. This high pressure liquid–vapor mixture is converted to low pressure liquid–vapor mixture (to the condenser pressure) using a screw expander which is connected to the generator to produce power. The low pressure refrigerant from the screw expander is cooled to the saturated liquid condition in condenser which is again pumped back to pre–heater and the cycle continues.

The above mentioned system model has three major components: the heat source loop, the heat sink loop (open loop) and the ORC system. In the heat source loop for diesel generator waste heat application, the heating fluid may be from engine jacket water or 50/50 glycol/water mixture for exhaust heat exchanger or both combined. In the heat sink loop the cooling fluid may be from cooling tower, radiator, or large water body (may be from nearby river or lake).

The modeling and simulation is a continuous process. As more system component information is available from experiments and open literature, the model will be updated to give more realistic and accurate results. The system component information such as for screw expander, boiling and condensing heat transfer coefficient of refrigerant in evaporator and condenser etc. is not available in open literature and need to be approximated based on experimental data and experimental analysis (if experimental data of the ORC system is accessible). As this data will be obtained from the experimental analysis will be used to tune the model so that in future it can be applied to any waste heat source for economic and feasibility analysis of the ORC system. The intention of the third stage is to enable the model capable of comparing the performance of the ORC system operated under different diesel engine load and environmental conditions, but is not to reengineer the design of the ORC system.

In this simulation basically five system parameters are being controlled they are inlet temperature and flow rates for heat source and heat sink input loops and the quality of the refrigerant inlet to the expander. The quality of refrigerant inlet to the expander is controlled to optimize the power output and efficiency of the ORC system for given heat source and heat sink conditions.
The system simulation has been performed for different screw expander inlet refrigerant quality (from 0.75 to 0.95) for given heat source and heat sink inlet conditions. The heat source and heat sink inlet conditions are flow rate and inlet temperatures of respective fluids (here water is considered for both heat source and sink). In the current preliminary simulation following assumptions were made:

1. All the ORC heat exchangers i.e. evaporator, pre–heater and condenser, are 100% efficient.
2. The quality of refrigerant out of the evaporator in the ORC system is controlled.
3. The quality of liquid out of pre–heater and condenser are saturated liquid.
4. The isentropic efficiency of screw expander and pump (within the ORC system) are taken to be constant at 78% and 70% respectively.

The assumptions can be modified easily, once the better performance characteristics of components are known from published or experimental data.

In simulating the ORC system performance explicit formulae for heat transfer coefficients of refrigerant on one side of the heat exchanger and water on other side of the heat exchanger should be known. Generally the heat transfer coefficient of a fluid is expressed in terms of its thermodynamic and transport properties. The heat transfer coefficient also depends on the geometry of heat exchanger, material of construction etc.

All the heat exchangers considered in the present case are plate heat exchangers (PHE). A widely accepted expression for heat transfer coefficient of single phase fluids in a plate heat exchanger is given by Muley and Manglik [4]. In ORC system this expression is used for calculating heat transfer coefficient of hot water and cold water in evaporator and condenser respectively and heat transfer coefficient of hot water and refrigerant in pre–heater. In all the above cases of calculating heat transfer coefficient the fluid thermo–physical properties were taken at average fluid temperature.
The expression for heat transfer coefficient of evaporating refrigerant liquid-vapor mixture in the evaporator is given by Ayub [5]. The expression for heat transfer coefficient of condensing refrigerant liquid-vapor mixture in the condenser is given by Selvam et al [6]. All the above expressions are presented in the appendix.

SIMULATION CASE STUDY
The constructed ORC system model has been used to simulate an example 50kW ORC system with the system parameters listed in Table–IIIA of Appendix-III. Figure–2 and Figure–3 show the figures for efficiency vs. expander inlet quality. These figures also show the effect of parasitic power and heat sink supply temperature on system efficiency. The parasitic power is the power needed to pump the heat source and heat sink fluids to/from the ORC system. As the heat sink supply temperature decreases (in this case from 21°C to 5°C), to remove the same amount of heat from the condensing refrigerant in the condenser less amount of cooling fluid is required. This may decrease the parasitic power and increase the efficiency of the system. This may be one of the advantages of using the ORC system during the winter months.

The effect of heat source temperature on the ORC system efficiency can also be observed in Figure–2 (93°C) and Figure–3 (121.11°C). As the heat source temperature increases the efficiency of the ORC system increases. Here in simulating the system for different heat source temperatures the screw expander inlet pressure conditions (or evaporator exit conditions) were different though all other system parameters remained same (i.e. condenser pressure, expander and pump efficiencies, heat source flow rate etc). For heat source temperature of 93°C case the expander inlet pressure was 6.95 bar and for 121.11°C case the pressure was 15.7 bar. As more work is produced by the expander when it goes from high pressure at expander inlet to same condenser pressure, this may be the reason for increase in system efficiency for different heat source temperatures.

The components’ properties used in this simulation are based on values of common practice, which may not match the properties of the expected ORC system. To obtain good match between simulation and experimental results, improvement in performance characteristic expressions of critical components may be needed. These expressions may be obtained from fitting parameter values which describe the performance characteristics of components using published and experimental data. According to the simulation, recommended data to be measured from the test system for individual component parameters are inlet and outlet temperatures, pressures, qualities, and flow rates of the major components (i.e. evaporator, condenser, pre-heater, expander, and pumps).
Figure–2: Efficiency of ORC system with varying screw expander inlet quality for heat source temperature of 200°F (93°C).

Figure–3: Efficiency of ORC system with varying screw expander inlet quality for heat source temperature of 250°F (121.11°C).
Test Parameters:

Reliability Test:
Reliability test is conducted for the experimental setup to evaluate the durability and performance stability of the system i.e. the system can function as desired without malfunction and major failure for a long enough period of time. During reliability test the ORC system will be run at full load under the specified operation condition i.e. to run at 235°F of hot water to GM temperature with heating fluid flow rate and coolant temperature controlled to keep maximum system power output (without violating the requirement of GM). The specific tasks of this test include,

- In the early stages of installing the system, noticing the malfunctions of any system components in the heat source or heat sink loops. Those components will be repaired or replaced and the reason for malfunction will be duly documented.
- During the reliability test, if the system breakdown or does not function as desired, component involved, the cause of failure, time to repair the component and time to re-start the system will be duly reported.
- During the reliability test, maintenance requirement for the system, frequency of maintenance, skills required for operations, repair and maintenance will be noticed and documented.
- Recorded parameters from the power unit during the reliability test include power generated, heat input and flow property at heat source, heat rejected and flow property at heat sink and parasitic power consumed by the power unit, heat source and heat sink loops. These parameters will be useful in estimating the efficiency and performance stability of ORC system and the experimental setup.

Parameters related to the ORC components performance are also recommended to be recorded for component performance monitoring if they are accessible.

Performance test:
The performance test is also conducted for the experimental setup. During the performance test the system will be tested for various heat source and heat sink input conditions.

Heat source:
Heat source temperatures recommended for testing are from 155°F to 235°F with an increment of 20°F. The system will also be tested at different heat source flow rates as listed in Table 2. The heat source temperature and flow rates were selected based on the simulation results which showed significant enough difference in efficiency. This temperature and flow rate range covers the jacket water, exhaust heat recovery fluid and both combined of a typical rural Alaska diesel engine.
Table 2 Recommended flow rates and temperature combination for testing the experimental system

<table>
<thead>
<tr>
<th>Heat source temperature (°F)</th>
<th>Heat source flow rate (GPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>155</td>
<td>1. 120GPM</td>
</tr>
<tr>
<td>175</td>
<td>2. 140GPM</td>
</tr>
<tr>
<td>195</td>
<td>3. 156GPM</td>
</tr>
<tr>
<td>215</td>
<td>4. 180GPM</td>
</tr>
<tr>
<td>235</td>
<td>5. 200GPM</td>
</tr>
<tr>
<td></td>
<td>6. 240GPM</td>
</tr>
</tbody>
</table>

Heat sink:
Heat sink temperatures recommended for testing are from 40°F to 100°F with an increment of 20°F. The cooling water flow rate is 200 GPM. The heat sink temperature range was selected based on water freezing point and typical Alaska coolant temperature availability during summer months (50°F to 90°F).

Output parameters to be measured:
The output parameters that are measured from the experimental system are power generated, heat input and flow property at heat source, heat rejected and flow property at heat sink, parasitic power consumed by power unit, heat source and heat sink loops. These output parameters will be used to determine the efficiency of the system at various heat source and heat sink input conditions.

ORC system parameters:
The measurements obtained from the power unit (which are provided by manufacturer of GM) are the expander inlet and outlet pressures, temperature difference across expander, net and gross power outputs, generator RPM. All the above data which are externally measured on the heat source and heat sink side of the power unit (such as the temperatures of hot and cold liquids entering the power unit and their flow rates) are also measured by green machine data acquisition system which can be used for cross checking. According to the simulation, recommended parameters to be measured from the ORC system, if accessible, for individual component parameters are inlet and outlet temperatures, pressures, qualities, and flow rates of the major components (i.e. evaporator, condenser, pre-heater, expander, and pumps).

Installation and Instrumentation:
The task of installation and instrumentation include installation of the complete testing system (i.e. ORC system, heat source system, cooling system, electrical power consumption devices, measurement and data acquisition devices, and remote monitoring devices). Detection, rectification, and modifications (if it is needed) are also part of this task. Since regulations, safety, and reliability are important to this project, experts in related areas are being consulted for piping and control system design and installation.

The space for the experiment had been located (i.e. Bidwell Building) and reported in the forth quarterly report; since then the location was relocated (i.e. inside the UAF Power Plant) in order to largely reducing installation cost. Currently the equipment for testing has been
transported to the new test site. Since the new site has a relatively small available space, the installation will be conducted after detailed system diagram has been carefully investigated. Appendix II shows the floor plane of available space and related connection locations (steam outlet and condensate locations, hydrant and water discharge locations, and electrical power switch board and internet connection location). Alone with the floor plane are four pictures showing the available space around the green engine. Based on a discussion with ET engineer, the available space meets the requirement for ordinary maintenance.

Expected experimental setup:
For field application the experimental system has three major components: the heat source loop, the heat sink loop and the ORC system (power unit). In the heat source loop the heating source may be coolant fluid from engine jacket water or exhaust heat exchanger or both combined. In the heat sink loop the cooling fluid may be from cooling tower, or dry cooler system, or radiator, or large water body (e.g. nearby river or lake). To simulate the heat source condition the outlet hot water of a steam/water heat exchanger, which transfers heat from power plant steam to water, is used as heat source. Cold water from a fire hydrant is used as the cooling source in the experiment. The line diagram of the system with heat source and heat sink is shown in Appendix IV for the new test site. In Appendix IV pictures of some of the components used for the test are also included.

The current status of the experiment setup is that several on shelf components, hoses, and pipe line accessories need to be purchased. Installation is expected to be completed before August 20.

The main components of the heat source loop are the steam/water heat exchanger, automatic steam cut off valve, VFD pump, 3-way temperature control valve, strainer, BTU meter, expansion tank, temperature mixers, and ON/OFF solenoid valves. The steam/water heat exchanger, VFD pump, and 3-temperature control valve are used to control hot water temperature and flow rate flowing into the ORC system to simulate heat source conditions obtained from different diesel engine loads. The automatic steam cut off valve cuts off the steam flow while GM stops. Temperature mixers are used for maintaining uniform temperature throughout the cross-section of the pipe for more accurate heat input measurements. The ON/OFF solenoid valves are used when there is emergency shutdown of the power unit so that the hot water by-passes and flows back to the boiler instead of power unit. The BTU meter is used for measuring the flow rate and amount of heat released by the hot water to the working fluid in the evaporator and pre-heater of the power unit.

The components on the heat sink loop include a manually controlled flow rate valve, hydrant, 3-way temperature control butterfly valve, strainer, ON/OFF valve, and BTU meter, and two flow temperature mixers. The hydrant, manual flow rate control valve, and the 3-way temperature control valve are used for flow rate and temperature control of the cooling fluid entering the GM cooling side. The BTU meter is used for measuring the flow rate and amount of heat released by the working fluid to the cooling water in the condenser of the power unit.

The instrumentation of the system also includes the installation of remote monitoring and remote data collection devices. The remote monitoring and data collection is done from both UAF and ElectraTherm facility in Nevada. For this purpose we may require broadband internet connection cables, static IP address, and modem which are being installed at the
Bidwell experimental site.

**Reliability Testing:**
Reliability test, which is a 600-hour test, considers the durability of the system for long term operation. The test results will include details of malfunctions, repairs, maintenance performed, and other major system performance (e.g. fuel consumption, performance stability, power generated) during the test. The results will also include technologies needed for applying these systems.

**Performance Testing:**
Performance test considers the performance of the ORC system and the performance of the test system (i.e. ORC system, heating loop, and cooling loop) as functions of operation conditions of the heat source and heat sink. Performance of ORC system includes net system efficiency, system consistency, net and gross power generated and more if more measurement devices are available. Test performance result includes detailed relationships between electrical output, parasitic power, and operation parameters of heat source and heat sink. This information may be used for the selection of appropriate diesel generators for this particular ORC system. The information may also be used to estimate electrical energy, which can be generated from a particular diesel generator with given load pattern and environmental conditions. If more measurement data (e.g. information of working fluid states along the ORC loop, power consumption of the working fluid pump) becomes available, the data can also be used for performance verification of the ORC system and design parameters recommendation.

**Comparison in Performance and Economic Impact between the 65kW and 250kW ORC Systems:**
Performance comparison will include discussions of efficiencies and powers generated of the two systems; net efficiencies and net powers generated of the systems with respect to different possible cooling systems; feasibilities of the systems for different sizes and types of heat sources (e.g. exhaust, jacket water); economic analyses (e.g. total benefit, cost return time) for different heat sources. Other discussions include issues related to installation, maintenance, operation, operation stability, technology requirement in applying these systems.

**Report Writing and Dissemination:**
Detailed information of all the above mentioned tasks will be covered in regular quarterly reports and final report. In addition, technical findings will be submitted to appropriate journals and conferences.

**MAJOR PROGRESS OF THIS QUARTER:**
A. Bid for installation for Bidwell test site.
   Two quotations have been received from the contacted contractors for installation of the testing system in the Bidwell test site. The cost from both quotations is about $150,000. This cost is beyond the expectation and the budget. One part of the major cost was the concerns of matching the local regulations for installing boiler, plumbing system, and electrical system.

B. Relocation of the installation site.
ACEP engineers Tom and Jack searched for ways to make the installation less costly and found a new test site and support from the UAF Power Plant. The advantages of the new site include the existing heating and cooling supplies, fewer or no local regulations applicable for installation, and reduced cost for installation and operation.

C. Modified test system line diagram for the new test site has been completed. For the new test system, the boiler and cooling tower systems may not be used. All the other purchased components will be used in the new test system. The cooling tower system will be used later for field testing. New purchases of components are needed for the new system. Details about existing components and components of new purchasing for the modified test system are listed in Appendix V.

D. A meeting involving all participating parties. A teleconference, which included representatives of all the parties (i.e. power plant, ET, TCC, and ACEP) involved in this project, has been held on June 23. In the meeting, the following issues have been discussed:
   Installation requirements (obtained from ET) for GM installation.
   Detailed discussion of the modified line diagram.
   Power plant resources availability and their properties.
   Commissioning details.
   The meeting helps all the parties understanding the current status in test system design, procurement, and resources availability, etc.

MAJOR FUTURE WORK:
A. Completing installation and instrumentation.
B. Continuation in system modeling and simulation.
   Development of simulation methodologies for second and third stages.
C. Analyses of system performance data and economic impact.
D. Continuation of report writing and dissemination.
REFERENCES


Appendix I
ORC PROJECT TIMELINE (revised by Ross Coen and Chuen-sen Lin, 3/22/11)

PHASE 1 (January 1, 2010 to December 31, 2011)
Task 1—Procurement (01/01/10 to 03/15/11)
To purchase ORC system, cooling system (radiator, temperature control loop components), heating system (flow rate control), thermal flow measurement devices (flow meters, digital pressure gauges, thermocouples, etc.), ORC electrical power consumption and measurement devices (resistors, heat dissipation unit, and electrical power measurement devices), flow control system (pump, pipe, valves, thermal couple, etc.). Task completed.

Task 2—Installation and instrumentation (03/16/11 to 08/20/11)
To install complete testing system (ORC system, electrical power consumption system, heat source system, cooling system, instrumentation), detection and rectification, and modifications.

Task 3—Testing (08/20/11 to 10/30/11)
Part 1:
To conduct a 600-hour test to verify the performance and reliability of the ORC system to ensure that no undesired interruption occurs to the village diesel power during the demonstration phase. To record all measured data (i.e., thermal fluid data and electrical power data).

Part 2:
To conduct 50-hour performance test of the ORC system under controlled input and output conditions, which simulate varying diesel generator load and varying environment conditions. The data will be used for the design of an operation scheme to optimize the performance of the ORC system in a village power plant, which normally experience varying load and environmental conditions.

Both Parts 1 and 2:
To collect all performance data needed for analysis.
To record installation costs and O/M cost associated with the system.
To record O/M issues and failures.

Task 4—Report writing and dissemination (11/01/11 to 12/31/11)
Intermediate and final reports will include test setup and data, detailed installation and O/M issues, and analysis of system efficiency, GHG reduction, and fuel consumption. Report will also include a recommendation of how to select an ORC system, a scheme to optimize the system performance, and preliminary studies in economic impact and feasibility of applying ORC to diesel generators in rural Alaska. Part or all the information mentioned above will also be disseminated through conference proceedings and/or journal publications.
APPENDIX II
Experimental Building Space Floor Plan

1-4: Green Machine (GM) and space around GM
5: Condensate Outlet
6: Front of GM, looking towards the door
7: Way to steam source
8: Steam valve
9: Steam pressure guage
10: Flange to radiator
11: Location of flange
Green Machine

Space available in front of GM
Space available left of GM
Space available right of GM
Space available behind the GM
Appendix-III
Expressions for Single Phase and Two-Phase Heat Transfer Coefficient of Fluids in Plate
Heat Exchangers

In calculating the heat transfer coefficients of fluids we need to know the physical parameters
of the plate heat exchanger. Table-IIIA gives the physical parameters of a typical plate heat
exchanger taken from open literature [4] for a 50 kW ORC system.

Table–IIIA: Plate Heat Exchanger Physical Parameters considered in the present simulation

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Plate Width (w in m)</td>
<td>0.5</td>
</tr>
<tr>
<td>Plate Length (L_p in m)</td>
<td>1.1</td>
</tr>
<tr>
<td>Channel Spacing (m)</td>
<td>3.50E-03</td>
</tr>
<tr>
<td>Thickness of Plate (t in m)</td>
<td>6.00E-04</td>
</tr>
<tr>
<td>Cheveron angle (β in degrees)</td>
<td>45</td>
</tr>
<tr>
<td>Enlargement Factor (Φ)</td>
<td>1.29</td>
</tr>
<tr>
<td>Corrugation Pitch</td>
<td>7.00E-03</td>
</tr>
<tr>
<td>Equivalent diameter (D_e in m)</td>
<td>7.00E-03</td>
</tr>
<tr>
<td>Projected area per plate (A_p in m²)</td>
<td>0.55</td>
</tr>
<tr>
<td>Flow area on one fluid side (A_f in m²)</td>
<td>4.20E-02</td>
</tr>
<tr>
<td>Surface area on one fluid side (m²)</td>
<td>26.4</td>
</tr>
<tr>
<td>Hydraulic Diameter (D_h in m)</td>
<td>0.005426</td>
</tr>
<tr>
<td>Pressure drop in heat exchanger (MPa)</td>
<td>0.05</td>
</tr>
<tr>
<td>Plate Thermal Conductivity (K_p in W/m-K)</td>
<td>13.8</td>
</tr>
<tr>
<td>Plate Thermal Resistance (m²-K/W)</td>
<td>4.35E-05</td>
</tr>
<tr>
<td>Area of Evaporator (m²)</td>
<td>33.26</td>
</tr>
<tr>
<td>Area of Condenser (m²)</td>
<td>28.3</td>
</tr>
<tr>
<td>Area of Pre–Heater (m²)</td>
<td>15</td>
</tr>
<tr>
<td>Critical Pressure of R245fa (MPa)</td>
<td>3.651</td>
</tr>
<tr>
<td>Critical Temperature of R245fa (°C)</td>
<td>154.01</td>
</tr>
</tbody>
</table>

Heat Transfer Coefficient for Single Phase Fluids

The expression for convective heat transfer coefficient for single phase fluids in a plate heat
exchanger is given by Muley and Manglik [4] and is read as

\[
h_f = \frac{Nu \times K_f}{D_e}
\]  \hspace{1cm} (1)

\[
Nu = \frac{0.2668 - 0.006967\beta + 7.244 \times 10^{-5} \beta}{Pr^{1/3} (\mu / \mu_n)^{0.14}}
\]  \hspace{1cm} (2)

In the above equation \( K_f \) is thermal conductivity of fluid and the Reynolds number (Re) is
based on equivalent diameter of the plate heat exchanger and is calculated as,

\[
Re = \frac{(G \times D_e)}{\mu}
\]  \hspace{1cm} (3)
Where \( G \) is the mass velocity of the fluid, \( D_e \) is the equivalent diameter and \( \mu \) is the dynamic viscosity of the fluid. The mass velocity \( (G) \) is the ratio of mass flow rate of the fluid to flow area of the heat exchanger.

**Heat Transfer Coefficient for Evaporating Fluids**

The expression for convective heat transfer coefficient for evaporating fluids in a plate heat exchanger is given by Ayub [5] and is read as

\[
h_{\text{Eva}} = 0.0675(K_f / D_e)[(Re^2 h_{fg} / L_p)^{0.4124} (p / p_{cr})^{0.12} (65 / \beta)^{0.35} \text{[BTU/hr-ft}^2\text{-oF]} \tag{4}
\]

In the above equation \( K_f \) is the conductivity of the fluid, \( h_{fg} \) is the enthalpy difference between outlet and inlet of heat exchanger, \( L_p \) is length of the plate, \( p \) is pressure and \( p_{cr} \) is critical pressure. Reynolds number \( (Re) \) is based on equivalent diameter and its calculation is similar to the above equation for single phase fluids (Eq. (3)).

**Heat Transfer Coefficient for Condensing Fluids**

The expression for convective heat transfer coefficient for condensing fluids in a plate heat exchanger is given by Selvam et al [6] and is read as

\[
h_{\text{Cond}} = (Nu_{\text{Cond}} \times K_f)/D_h \tag{5}
\]

\[
Nu_{\text{Cond}} = Ge_1 Re_{Eq}^{Ge_2} Pr^{1/3} \tag{6}
\]

\[
Ge_1 = 11.22 \left( \frac{p_{co}}{D_h} \right)^{2.83} \left( \frac{\pi}{2} - \beta \right)^{-4.5} \tag{7}
\]

\[
Ge_2 = 0.35 \left( \frac{p_{co}}{D_h} \right)^{0.23} \left( \frac{\pi}{2} - \beta \right)^{1.48} \tag{8}
\]

\[
Re_{Eq} = \frac{G_{Eq} D_h}{\mu_f} \tag{9}
\]

\[
G_{Eq} = G_e \left[ 1 - x + x(\rho_f / \rho_g)^{0.5} \right] \tag{10}
\]

\[
G_e = \frac{m}{A_f} \tag{11}
\]

Where \( K_f \) is the thermal conductivity of the fluid, \( p_{co} \) is plate corrugation pitch, \( D_h \) is hydraulic diameter, \( x \) is quality of fluid entering the condenser, \( \rho_f \) and \( \rho_g \) are the density of
saturated liquid and vapor at condenser inlet condition. Here we need to observe that the Reynolds number (Re) is based on hydraulic diameter.
Appendix-VI
Experimental system line diagram and some system component pictures
Line diagram for heating and cooling loops to power unit
Pump for heat source loop

VFD for the pump (heat source loop)
3-way flow control valve on heat sink loop
## Appendix-V

List of components

### Components already purchased and delivered at power plant:

Green Machine components from ElectraTherm:

<table>
<thead>
<tr>
<th>#</th>
<th>Component</th>
<th>Port diameter</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Green Machine main body</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Cold side flanges</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>3</td>
<td>Filter dryer</td>
<td></td>
<td>2 cans</td>
</tr>
<tr>
<td>4</td>
<td>Cold side expansion joints</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>5</td>
<td>Hot side expansion joints</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>6</td>
<td>Gaskets, Bolts &amp; Nuts</td>
<td></td>
<td>1 Box</td>
</tr>
<tr>
<td>7 &amp; 8</td>
<td>Hot side Solenoid Valves</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>9</td>
<td>Generator protection system</td>
<td></td>
<td>1 Box</td>
</tr>
<tr>
<td>10 &amp; 11</td>
<td>Manual stop valve</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>12 &amp; 13</td>
<td>3-Way connection on hot side</td>
<td>3”</td>
<td>2</td>
</tr>
<tr>
<td>14</td>
<td>Refrigerant lubricant</td>
<td></td>
<td>2 cans</td>
</tr>
<tr>
<td>15</td>
<td>Flow switch and connection</td>
<td>3”</td>
<td>1</td>
</tr>
<tr>
<td>16</td>
<td>Refrigerant – R245fa</td>
<td></td>
<td>6 Tanks</td>
</tr>
<tr>
<td>17</td>
<td>Temperature mixers for hot and cold side</td>
<td>3”</td>
<td>2</td>
</tr>
</tbody>
</table>

Pipe components:

<table>
<thead>
<tr>
<th>#</th>
<th>Component</th>
<th>Port diameter</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>VFD pump for hot side</td>
<td>4”</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>VFD control system for pump</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>3-Way butterfly flow control valve on cold side</td>
<td>4”</td>
<td>1</td>
</tr>
<tr>
<td>4 &amp; 5</td>
<td>BTU meters for hot and cold side</td>
<td>4”</td>
<td>2</td>
</tr>
<tr>
<td>6</td>
<td>Temperature measuring devices</td>
<td></td>
<td>1 Box</td>
</tr>
</tbody>
</table>

### Additional components need to be purchased for project:

<table>
<thead>
<tr>
<th>#</th>
<th>Component</th>
<th>Quantity</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Steam to hot water heat exchanger</td>
<td>1</td>
</tr>
<tr>
<td>2</td>
<td>Manual flow rate control valve on cold water supply side</td>
<td>1</td>
</tr>
<tr>
<td>3</td>
<td>Temperature mixers</td>
<td>1</td>
</tr>
<tr>
<td>4</td>
<td>3” to 4” connectors</td>
<td>12</td>
</tr>
<tr>
<td>5</td>
<td>Automatic steam ON/OFF</td>
<td>1</td>
</tr>
<tr>
<td>valve</td>
<td></td>
<td></td>
</tr>
<tr>
<td>-----------------</td>
<td>-----------------------------------------------------------------</td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>3-way temperature control valve on hot side</td>
<td></td>
</tr>
<tr>
<td>7</td>
<td>Rubber hoses</td>
<td></td>
</tr>
</tbody>
</table>