PROJECT PROGRESS REPORT

PREPARED FOR TANANA CHIEFS CONFERENCE AND 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.

Disclaimer

Abstract

Diesel generators produce waste heat as well as electrical power. Considering diesel engine application in rural Alaska, in general, less than 40% of the fuel energy is used for electrical power generation and the rest is dissipated through different mediums or directly radiated into the environment in the form of low temperature heat (versus heat directly from fuel combustion). This project conducted an experimental reliability study and both analytical and experimental performance studies of an emerging heat engine technology, a low temperature organic Rankine cycle (ORC) system with a capacity of 50kW and using a screw expander to convert low temperature heat into electrical power. This project also involved performance comparison between the 50kW ORC system and a conventional 250kW ORC system. 50kW low temperature heat to power conversion system is currently considered an emerging technology, which is in the testing stage, and 250kW system, which uses radial turbine for power conversion and is commercially available, is considered a matured technology. Detailed performance characteristic charts and economic analysis have also been completed for the 50kW ORC system.

The project began with a survey of existing technologies and selection of the most promising and deliverable technology for testing. The technology has been delivered for testing was an ElectraTherm (ET) 50kW organic Rankine cycle (ORC) system. Test location was selected based on the potential cost and availability of resources (e.g., heat source, cooling source). Among two potential test locations, an available space in the power plant on the UAF campus was selected due to the availability of heating (steam), cooling (cooling water), and human resources and the potential of less total cost for installation and operation. The supporting systems (from now on it is called testing system) needed for the include thermal fluid systems for heating and cooling, an electrical circuit for power up-loading/consumption control, an instrument and signal process circuits for data acquisition/performance monitoring and control and an dedicated internet line for remote monitoring/control. Detailed designs of the supporting system were completed largely through the cooperation between the engineers of ACEP, facility services, and power plant of the UAF. Associated management personnel have also contributed a lot on schedule and resources controls.

A 600-hour reliability test was conducted after the installation and instrumentation. No major system break downs occurred during the test and the performance of the system and every component of the system appeared very consistent with the screw expander power (ORC heat to power converter) output of 50.1kW and net ORC system output of 46.4kW (i.e. a parasitic power of 3.7kW). During the 600 hours of the reliability test, the only flaw occurred was the inability to start the ORC system due to the defect of a pressure switch. The flaw was automatically detected by the ORC system software displayed on the system monitor screen. Pressure switch is one of a series of safety guards to prevent the machine from overload and its defect will not affect the system to continue to operate with any compromise in system security. According to the ET engineering group, the defect of this type of pressure switch is rarely occurs. Based on data obtained from the reliability test, the net efficiency of the ORC system is about 7.6%, potential annual energy generation 395,328 kW-hours (355 working days and 10

maintenance days), and diesel fuel saved 28,238 gallons (assumption: 14kW-hr/gallon for rural Alaska diesel power applications). The estimated annual reductions in CO_2 is 316 tons, CO 174 lbs, HC 261 lbs, PM 148 lbs, and NO_x 6,013 lbs. With an interest rate of 10% and a diesel fuel price of $5/gallon, the payback time is about 4.6 years, assuming that the diesel power plant cooling system is enough for the ORC cooling. With 0% interest rate, the estimated payback time then becomes 3.6 years.

Performance test was conducted for more than 50 hours after the reliability test. The performance test measured the electrical output of the ORC system and the performance parameters (e.g., temperatures and pressures of working fluid) of its components with respect to cases of different combinations of the temperatures and flow rates of the heating and cooling sources. The results measured from the test showed performance consistency of every ORC system component for every case. The results also showed that the efficiency for gross output of the ORC system ranges from 5.6% to 8.2% (net efficiency from 5.3 to 7.6) for a wide range of heating source and cooling source conditions.

The measured performance test results were used to construct the performance characteristic charts of the ORC system in terms of the heating and cooling source conditions. An example was given to demonstrate how these charts could be used to predict the performance, emissions and CO2 reductions, and economic effect of applying the ORC system on a village diesel generator.

The conclusion of the testing is that the ORC system is feasible and reliable for diesel generator waste heat recovery. The technology requirement in operating the system is minimal. The technology requirement in maintenance is expected feasible for rural village diesel generators, especially if the maintenance schedule of the ORC system is combined with the diesel generator maintenance schedule. Economic impact may depend on the match between the ORC system and the load pattern and capacity of the village generator set. In general, for an appropriate match, the economic impact is positive and promising.

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Chapter 1
Introduction

In rural Alaska, approximately 180 villages consumes about 370,000MWh [1] of electrical energy annually, using isolated diesel generator sets. In general, the majority of time the diesel generators are operated in partial load and low load conditions. This makes the electrical power to fuel energy ratio be less than 40% and the rest of the fuel energy become heat dissipating into the environment through engine jacket coolant, exhaust, direct radiation, etc. If part of the waste heat could be used, there would be significant fuel savings.

There are many heat recovery applications available for capturing diesel engine waste heat, including applications for general heating (e.g., space heating, city water temperature maintenance), direct thermal to electricity conversion, heat to power conversion using a heat engine, refrigeration, desalination, etc. Among these applications, waste heat for heating is considered the most efficient application; even heating is useful only for cold season. Although, in many cases, waste heat for heating is prohibited due to the village infrastructure and high construction cost, unwillingness of the villages, etc. A detailed report about waste heat for heating for Alaskan village diesel generators has been discussed in details in [2]. Waste heat for power through heat engines is also highly considered due to its acceptable efficiency (i.e., close to 10%), flexibility in electrical power utilization, and expected low maintenance (i.e. similar to steam engine or refrigeration systems). In addition, unlike heating application, power is needed yearly around.

Power usage in many of the rural villages is about or below 1MW. For these generator sets, the power to be produced using waste heat recovery is expected below 100kW. For waste heat engines belonging to this category, the power to cost ratio is expected very high if the heat engine is facilitated with a radial turbine (i.e. a type of expander) for heat to power conversion. Many different thermodynamic cycles and different types of heat to power expanders have been tried to bring the cost down. Thermal cycle examples include organic Rankine cycle (ORC) and ammonia/water (or Kalina) cycle. Examples of heat to power conversion expanders include screw expander, scrolling expander, piston expander, etc. All of them are still either in prototype and proving stage or prototype fabrication stage. It is well understood that the performance of a heat engine depends on conditions of heating source and cooling source, which largely relay on the load pattern and waste heat properties (e.g., exhaust, jacket coolant) of the diesel generator set and the available village cooling source. Therefore, to estimate the performance and economic impact of any waste heat engine on an individual diesel generator set, the performance data of the heat engine under various heating and cooling conditions are needed. In general, these data are obtainable from test of the heat engine under controlled heating and cooling conditions.

The objective of this project has four folds. The first is to prove that an improvement of the efficiency of the diesel power plant by about 10% (i.e. about 4% of fuel efficiency) is

achievable through the use of an organic Rankine cycle (ORC) system, which uses 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 a selected ORC system. 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 emissions and greenhouse gas reductions. The fourth is the performance and economic comparison of two ORC systems. One ORC system is a 50kW system, which uses screw expander, comes under emerging technology. The second ORC system is a Pratt & Whitney (P&W) 250 kW unit which uses radial turbine belongs to the category of well-developed technology. Due to an unexpected turbo charger break down of the Cordova diesel generator set, to which the 250kW ORC system will be installed, performance information of the 250kW ORC becomes unavailable at this moment. Therefore, the result of the fourth task, comparison in performance of the two ORC systems, is not included in this report and will be provided in a make-up report in the future.

This report presents details of each step performed along the process leading to the completion of this project. The rest of the report begins with a description of the project preparation (Chapter 2), which includes a survey of waste heat recovery technologies for power generation capacities lower than 100kW, the final selected technology for testing, and the layout of required elements for test plan and test site. Chapter 3 discusses ORC principles, modeling and simulation, and how the simulation results can be used for preliminary designs and selections of components and prediction of system performance under different waste heat and cooling conditions. Chapter 4 describes the layout of the final design for the ORC testing, details of the testing system and its components, and procurement. The final design layout has been obtained through a few design iterations and the design was based on the availability of the test space, estimated project cost, resources availability, codes and standards, and timing. The final selected testing location has both heating (steam) and cooling (water) sources available for the ORC system. The final testing system includes a closed steam loop coupled with a closed hot water loop for heating, an open cooling water loop for cooling, an electrical circuit for power up-loading and consumption control, instrument and signal process circuits for data acquisition/performance monitoring and control, and a dedicate internet line for remote monitoring and controls. Detailed designs of the testing system were completed through the cooperation between the engineers of ACEP, facility services, and power plant and students and faculty members of the ME department on the UAF campus. Associated management personnel have also contributed a lot on schedule and resources controls. In addition, many of them are volunteers. The last section of this chapter is about procurement, in which components purchased and receiving schedules are mentioned and some of the difficulties encountered in procurement are also pointed out.

Chapter 5 gives a description about installation and instrumentation. This includes installation methods applied to the thermal fluid system, data acquisition system for data collection and monitoring/system controls, calibrations of the measurement devices, etc. Chapter 6 briefly describes the commissioning process and related issues, such as, limitations and constraints of the ORC system, new findings, etc. during the commissioning. Chapter 7 describes equipment used and equipment arrangement for testing, scheduling, and cases included in performance testing. Chapter 8 presents details

of the reliability test. During the reliability test the ORC system was operated at its full capacity (50kW overall output) for 600 hours under manufacturer recommended temperatures and flow rates of heating and cooling sources. Chapter 9 discusses details of the performance test, which includes test cases, procedure, and results. Performance of the ORC system was investigated under numerous operation cases, the combinations of different heating and cooling conditions. Chapter 10 presents data analysis of test results. It includes the observation results of the requirements for operation and maintenance, reliability and feasibility, and potential economical and environmental benefits obtainable from applying the ORC system.

It also includes test data analysis and induced performance characteristic charts for the ORC system. An example is given to demonstrate how the characteristic charts are applied to estimate effects of applying the ORC system to a specific village diesel generator on economic and environmental impact. Chapter 11 summarizes the activities of the project from beginning to end. It includes important information of observations and findings, and suggestions to the ORC system design and recommendations for how the benefit can be optimized by applying the ORC system to rural diesel generator sets. Chapter 12 concludes this report by comparing project objectives and project accomplishment and expected future activities.

As mentioned before, discussions in data collection and analysis of the Cordova 250kW ORC system as well as the comparison of the two ORC systems are not contained in this report due to the interruption of the Cordova ORC project resulted from an unexpected break down of the turbo charger of the diesel generator. A separate make-up report about the 250kW ORC system performance and the comparison of the two systems will be provided after the performance data of the 250kW system becomes available.

Chapter 2
Project Preparation

This chapter describes the preparation work for this project. The preparation work includes three parts: the selection of an appropriate low temperature heat engine, the layout of required elements for testing plan (critical parameters, etc.), and testing site (utility, heat source, heat sink, etc.) selection.

SELECTION OF A LOW TEMPERATURE HEAT ENGINE FOR TESTING

The proposed project began with a survey (2008, by Jared Kruzek) of accessible manufacturers, who are involved in low temperature heat engine industry, about their industrial applications and potential and willingness of delivering, within a reasonable time frame, a low temperature heat engine with a power capacity between 10kW and 100kW. 18 manufacturers were contacted and their general product information covered (Appendix IA). The final manufacturers selected for further consideration included an ammonia/water system manufacturer and an ORC system manufacturer. The ammonia/water system manufacturer was contacted first due to its previous credible working experience with Alaskan power industry. In addition, the manufacturer has showed its complete design layout and willingness of spending its own budget for fabricating the system.

In 2010, the pre-shipment testing conducted to the fabricated ammonia/water system showed that a major component (the heat to power conversion unit) used for this system has two major drawbacks. These two drawbacks were hampering the overall system performance and caused repeated delays in delivery. While the manufacturer was continuing to work on improving the product, no timeline for delivery could be established. The drawbacks included: 1) migration of lubrication fluid from the power unit into the loop of working fluid, which lowered the heat transfer efficiency significantly after a short period of operation, and 2) much lower than expected heat-to-power-conversion efficiency of the power unit.

After the selection of the two manufacturers for further contact, the ACEP team has been continuing in contact with the ORC system manufacturer. In 2010, the ORC manufacturer has shown promising test results on its ORC prototype. The ACEP team decided to acquire an ORC system to keep the test project going. In order to be sure that no other promising technologies were overlooked, another survey was conducted (by Kelsey Boyer) to up-date the information in low temperature heat engine industry. The result showed that not much of changes had occurred since the first survey. The report of the second survey is given in Appendix IB.

REQUIRED ELEMENTS FOR TEST PLAN AND TEST SITE

The test plan has been decided to have two parts: one is for reliability test and the other performance test. It doesn't matter which part of test is considered, a heat engine needs a heat source, which can provide driving energy to the heat engine and emulate the heating conditions (i.e. temperatures and flow rates) receivable from the waste heat generated from village diesel generators. A heat engine also needs a cooling source, which can

absorbed the dissipated heat from the heat engine and emulate the variety of cooling conditions possibly provided by the cooling sources existing in the villages (e.g., surface and ground water, radiator, and cooling tower). In addition, heat energy transmitting devices (e.g., heat exchanger, pipe, pump, and valve) are needed to transmit heat between the working fluid in the ORC system and the media of the heating and cooling sources. Other required system elements include devices for the concerns of safety, reliable performance, etc.

The purpose of reliability test is to observe the endurance in operation and consistency in performance of the ORC system under the rated operation condition. The purpose of performance test is to find detailed performance of the ORC system and its components (i.e., efficiencies, energy consumptions, etc.) for the system under numerous operation conditions of heating and cooling. Therefore, parameters to be measured and measurement devices to be installed are many more than that for reliability test. In order to guarantee that all measurement components needed for this project be included in the final testing system design, a preliminary design line diagram, which lists out all the parameters to be measured, is considered useful. The line diagram of the preliminary design and required components are given in Appendix IIA along with components information (Appendix IIB). Based on the preliminary design and components information, requirements of space, facilities, and utilities of the test site could then be estimated; though a variety of final designs may exist and need to be screened out. The selection of the final design concept needs to be conducted possibly based on existing facility and resources of the testing site, available overall test budget, desired operation and maintenance requirements, timing constraint, etc.

Chapter 3
Modeling and Simulation

In order to obtain appropriate performance of the ORC system, properly selected components of the testing system are needed. In order to find the performance of the ORC system and the performance of its individual components, sensors and measured data of physical properties (i.e., temperatures, pressures, and flow rates) of the working fluid pertinent to the components are needed. The collected data are then analyzed to give the performance results. Preliminary designs and selections of appropriate components (e.g., sizes and types) could be obtained through the process of system modeling and simulation. Modeling and simulation may also help determining operation parameters that are critical to system performance. Combined with testing data, model can be further improved and simulation results may become good enough and useful in predicting long term performance and benefit obtainable from applying the ORC system to any individual diesel generators. The rest of this chapter describes the model constructed for the integrated system (the ORC system plus the testing system) and the simulation process and results. The model includes three components: heat source loop, heat sink loop, and the ORC system. Fluid used in heating and cooling loops is water and working fluid used in the ORC system is R245f refrigerant.

HEAT SOURCE

The physical heat source loop for the new test site is expected to include a hot water source from a steam/water heat exchanger, a VFD pump, and pipes and fittings. Other components for measurement and control are also included. Through the pipes, the heating fluid enters into the heat source heat exchanger of the ORC system and transfers heat to the working fluid, the R245f refrigerant. The loop can control the temperature and flow rate of the existing hot water of the steam/water heat exchanger. All important information along the steam and heating water loop (e.g., of fluid temperatures, pressures, flow rates, VFD rpm, pump power consumption) for each operating condition need be collected for system and components performance analysis. The model constructed corresponding to the heating loop includes all important operation parameters of all the function features. 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 a fire hydrant and, its manual control valve, pipes and fittings, and needed temperature and flow rate measurements and control devices. Through the pipes, the cooling fluid enters into the condenser of the ORC system. The loop has limited controllability in temperature and flow rate of the cooling fluid entering the condenser. Information of fluid properties along the pipes, power and water consumption corresponding to each of operation conditions need be collected. The model constructed corresponding to the cooling loop includes all important operation parameters of all the function features. In addition, the model can be easily modified to cope with different types of cooling source and components to be adopted.

ORC SYSTEM

A general ORC unit includes at least a pump, an evaporator, a heat to power converter, and a

condenser. Other components needed in modeling depend on the versatility of the physical construction of the ORC system. For example, according to publications, one known property of the ORC system with screw expander is the ability of allowing mixed vapor/liquid working fluid in the heat to power conversion unit (i.e. for this case is the screw expander), so that the system can add a control component to control the working fluid flow rate and/or quality of fluid entering the expander to optimize the system performance. The physical ORC system to be used for this project is an integrated unit, for which to conduct very accurate measurements of working fluid properties for performance analysis of individual components, without modifying the system (modifying the system may results in losing warrantee), may not be practical. Also detailed engineering information of individual components may not be available for the concern of proprietary. However, the system does allow adding more sensors for the access to approximate working fluid properties pertaining to the performance of many of the components. This is very helpful in getting better analysis results of the components.

MODELING METHODOLOTY

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 of components 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) and provision of comments to the design of the system, detailed engineering data of components will then become required.

For the purpose of being able to fully model the system with reasonable accuracy, the plan of the modeling could be divided into two to three stages depending on how feasible (including achievability and timing) and desirable to know the details of the performance parameters of the ORC system and its components. 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 and selecting components of the testing system. The second stage is to fit the system parameters using data obtained from approximated measurement results and a limited number of experimental cases (i.e. heating and cooling conditions). If system simulation results obtained using fitted (approximate) 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 this report, methodology of first stage is described in details. The methodologies of the second and third stages are given in Appendix IIIA for reference purpose only.

Stage One:

The selected ORC system has features of being able to optimize the system net output for all heating and cooling conditions and limiting the total output to a maximum of 50kW. There are other features related to conventional constraints for performance regulating and system protection, such as constraints in maximum temperature, maximum pressure, etc. In a

physical prototype, these features are performed by physical mechanisms, such as mechanical and electrical devices. To simplify the mathematical model, some of the mechanisms and related performance parameters are neglected due to the much less influence than the others to the gross performance of the ORC system. The simplified model (Figure 1) includes the function of evaporator, a screw expander, a condenser, and a VFD pump. The model also 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 simulation results are useful for test planning. The expander is modeled by a single efficiency at this stage 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. The evaporator has the capability to model liquid, liquid/vapor mixture, and vapor flows. Currently, 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.

SELECTIONS OF COMPONENT PARAMETERS AND OPERATION PARAMETERS

Ranges of values of operation 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, which mostly are based on the recommendation of the manufacturer, include maximum pressure of the ORC system (150psi), estimated heat source temperature (235°F), 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 [3] 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/m^2-^{o}K or 265Btu/ft^2-^{o}F), evaporator area (e.g. 100ft^2), heat transfer coefficient of condenser (e.g. 1400 W/m^2-^{o}K or 247Btu/ft^2-^{o}F), and condenser area (e.g.200ft^2). 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 [4, 5]. 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 efficiencies, expander power) of the ORC system, net efficiencies of the test system, etc. 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 are used to help design test plan of the ORC system.

Table 1 Thermodynamic properties and environmental date of R245fa

Safety	Vaporization Heat (1atm.)	Boiling T.(1atm.)	Critical Point	Saturation Slope	ODP*	GWP** 100 year
Non-Flammable	197.5 Kj/Kg (355.5 Btu/lb)	14.6^{o}C (58.3^{o}F)	154°C (309.2^{o}F) 36.4 bar (527.9psi)	Isentropic	-0	1020

*Ozone depletion potential

** Green house warming potential

Besides simulation results in system performance, the process also help estimating sizes and capacities of components needed for the testing system, such as sizing of pumps and heat exchangers for heating and cooling, pipes, valves, etc (Appendix IIIB).

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 from 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 as mentioned before. 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 superheated or saturated (including vapor of vapor/liquid mixture). This high pressure working fluid is converted to low pressure liquid or vapor/liquid 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 desired state in condenser and the liquid portion 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 exiting exhaust heat exchanger or both combined. In the heat sink loop the cooling fluid may be from cooling tower, radiator, or large water body (from a 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 coefficients of refrigerant in evaporator and condenser etc. is not available in open literature and need to be estimated 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, the data 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 (Appendix IIIA) 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.

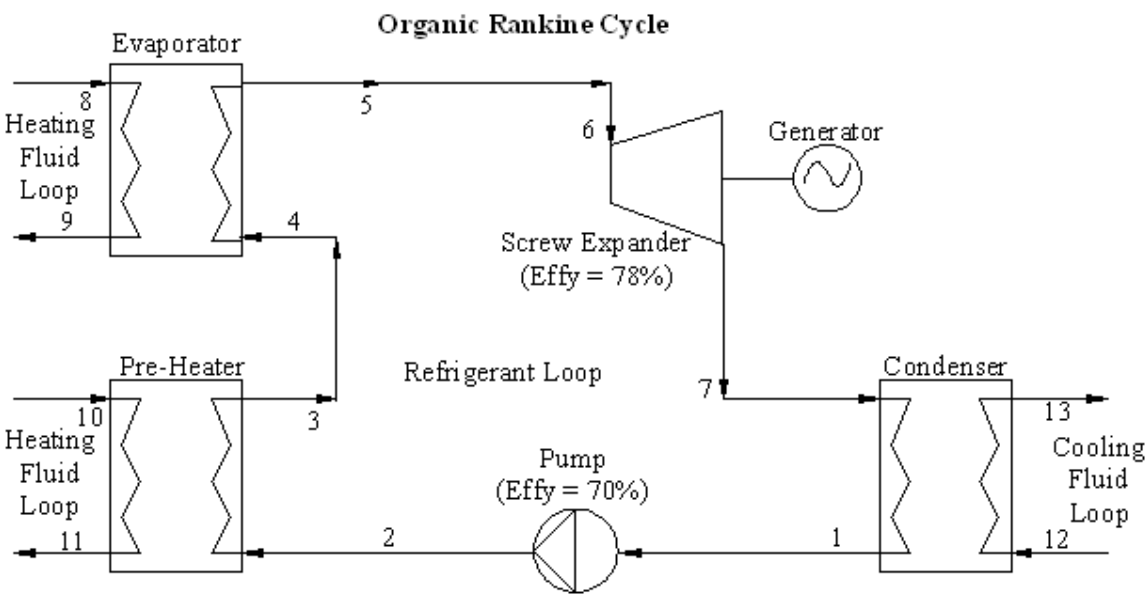
In this simulation basically five system parameters are being controlled they are inlet temperature and flow rates for heat source and heat sink input loops and the state of the refrigerant inlet to the expander. The quality of refrigerant inlet to the expander is controlled to optimize the power output 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 state 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.

Assumptions will affect the system performance characteristics. If the simulation results don't match the published or measured performance characteristics of the real system, test plan need to be designed to determine which assumptions need to be changed. The model can be modified easily, once better performance characteristics of components are obtained from experiment.



Figure–1: Schematic of Organic Rankine Cycle System

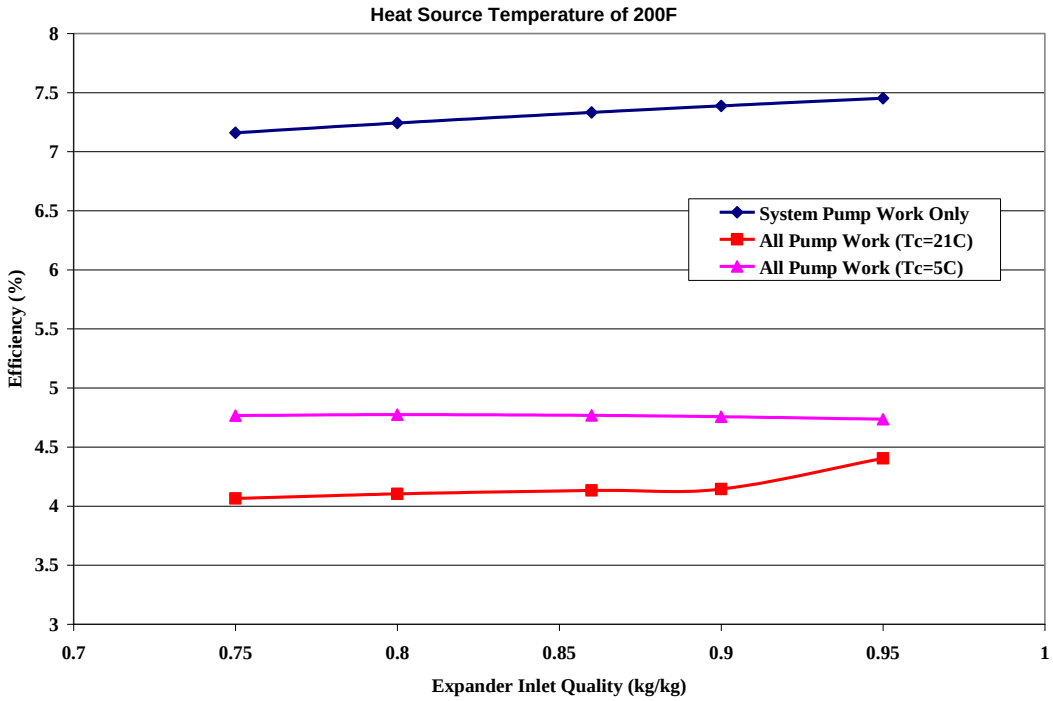
In simulating the ORC system performance explicit formulae for heat transfer coefficients of refrigerant on one side of the heat exchanger and water on other side of the heat exchanger should be known. Generally the heat transfer coefficient of a fluid is expressed in terms of its thermodynamic and transport 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 [6]. 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.

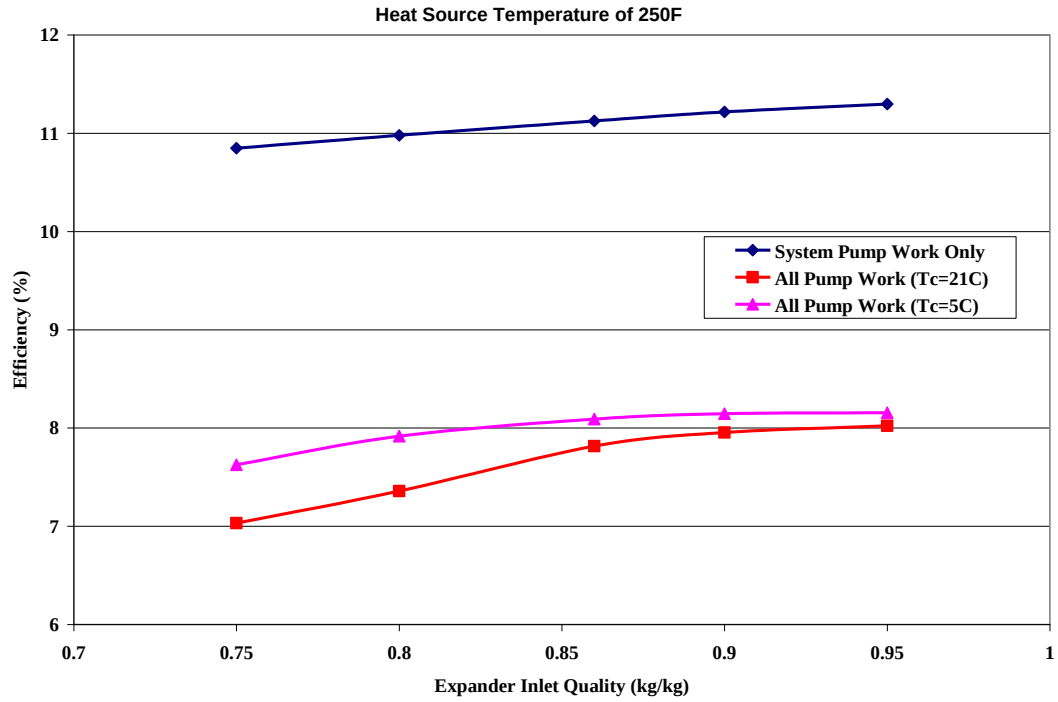
The expression for heat transfer coefficient of evaporating refrigerant liquid-vapor mixture in the evaporator is given by Ayub [7]. The expression for heat transfer coefficient of condensing refrigerant liquid-vapor mixture in the condenser is given by Selvam et al [8]. All the above expressions are presented in the appendix IIIB.

SIMULATION CASE STUDY

The constructed ORC system model has been used to simulate an example ORC system of 50kW with the system parameters mentioned above, and heat exchanger parameters and computation method listed in Table–IIIB-1 of Appendix-IIIB. 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.



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^{o}C).

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^{o}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^{o}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.

Based on the given model, simulation results show that the higher temperature of the heating fluid and lower temperature of cooling fluid give better system performance. Higher flow rates for both heating and cooling flows also give better system performance. However, the increase in system performance may be capped by the size of the respective heat exchangers.

For the current model, efficiencies of the pump and screw expander are both considered constant. This may lead to a monotonically increasing relationship between the system efficiency and the efficiencies of the working fluid pump or the screw expander. However, according to published data, the ORC system may have an optimal efficiency in terms of the working fluid flow rate at each operation condition, defined by the heating and cooling flow conditions. This suggests that, in order to better simulate the system performance, the model of the system need to include parameter of working fluid flow rate and other related parameters, such as parameters defining performance of pumps, screw expander, evaporator,

and condenser. These parameters can be estimated from matching the model simulation results and measured data, of which the best possible data are data from approximate measurements (i.e. steady state temperatures measured from the outside surfaces of bared tubes of the working fluid). For the structure of this ORC system, the data from the approximate measurements may be good enough for the purpose of this project.

Chapter 4
Test Plan, Design and Selection of Components, and procurement

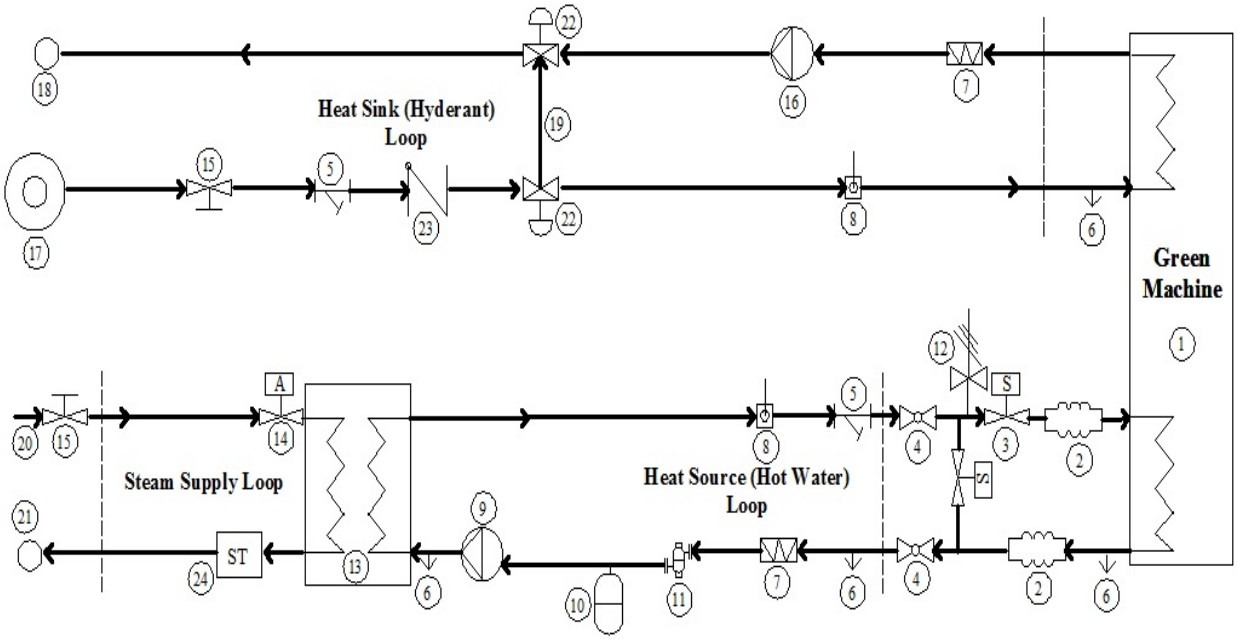
TEST PLAN AND DESIGN/SELECTION OF COMPONENTS

Test plan includes two parts: a 600-hour reliability test and a 50-hour performance test. As mentioned in Chapter 2, the reliability test focuses on system reliability and performance consistency of the ORC system for long term operation and performance test focuses on identify system and components performance characteristics under different operation conditions. The required equipment to obtain the desired information for these two parts of testing includes a heating source and a cooling source of which the fluid flow conditions are controllable, a power up loading (generated by ORC system) and consuming (e.g., by pumps) circuit, measurement and data acquisition devices for measuring and collecting operation and performance data of critical components in the system, and an internet connection between the ORC system and ElectraTherm facilities. Other accessories needed include piping pressure regulator, safety devices, filters, etc. The ORC system is a complete self regulated system with system performance measurement and control/safety/monitoring devices for some of its critical components.

The interface between the ORC system and the environment are through the heating fluid inlet and exit ports (for heating fluid), cooling fluid inlet and exit ports (for cooling fluid), the UAF electrical motor center (for power up-loading from the ORC system to the UAF power plant and power consumption of pumps and actuators), and internet connection box (for remote monitoring and control from the ElectraTherm company). Manual interface devices between human and machine include HMI, emergency switch, manually operated valves, etc.). The ORC system doesn't have enough measurement devices for performance evaluations of all the major components (i.e. expander, evaporator, condenser, and pump). If performance evaluation is required for every major component, extra sensors for temperatures and pressures pertinent to the components are needed. Due to the integrated nature in manufacturing the ORC system, temperature sensors cannot be installed in direct contact with the working fluid. Therefore, temperature sensors are installed to pipe outside surfaces immediately next to the fluids at the locations of interest. Since the measurements are for steady state operation conditions, the deviations between the measured temperatures and the respective true temperatures of fluids are expected small enough to have minimal effect on the conclusions of this project.

The final selected test site is a vacant space in the UAF Power Plant building. UAF power plant has plenty of steam supply for heating, water supply for cooling, and pressurized de-ionized water for pipe pressure maintenance. All of the available resources made the installation and operation expenses low and affordable. The final design layout of the heating and cooling loops has been obtained through several design iterations and the design was based on the availability of the test space, available project budget, accessible resources, codes and standards, and timing constraints. The final testing system includes a closed steam loop coupled with a closed hot water loop for heating, an open cooling water loop for cooling, an electrical circuit for power up-loading/consumption, instrument and signal processing circuits for data acquisition/performance monitoring and control, and an internet connection for remote monitoring/control. A detail of the available floor space for installation is given in

Appendix IVA. Detail of design layout is given in Figure 4 and the selected components for heating and cooling are list in Table 2 with brief information in component selection and specification.



#	Component	#	Component	#	Component
1	Green machine (GM)	12	Pressure relief valve	23	Check valve
2	Expansions joints	13	Steam-to-hot water heat exchanger	24	Steam trap
3	Solenoid valve	14	Steam control valve (may be automatic)		
4	Ball valve	15	Steam manual control valve		
5	Strainer or Filter	16	Pump (Constant flow rate)		
6	Drain	17	Hydrant source (Cooling water source)		
7	Temperature mixer	18	Cooling water from GM (Hydrant sink)		
8	BTU meter	19	By-pass for temperature control on coolant side		
9	VFD Pump	20	Steam inlet		
10	Expansion tank	21	Steam condensate outlet		
11	Rolairtrol air separator	22	Temperature control valve		

Figure 4 Design Line Diagram of the Testing Syste

Performance characteristics of the ORC system and its components are obtainable from measured data through data analysis. Performance characteristics are defined in terms of various input conditions of the heat source and heat sink. The performance characteristics can then be used for selecting appropriate rural Alaska diesel gen sets for this ORC system and planning waste heat distributions for optimal returns from waste heat applications. Operation and performance parameters to be detected and collected for this test are summarized into five groups and listed in Table 3. Besides the measurement devices equipped with the ORC system, measurement devices for all the other parameters listed in Table 3 need to become available through purchasing or other means.

The following sub sections discuss more details of the individual sub systems mentioned:

Heat Source Loop:
Again the heat source loop can be sub divided into two loops; they are steam loop and hot water loop. Steam to hot water heat exchanger is the component which connects both of the loops. This heat exchanger is used to transfer heat from steam to hot water.

1. The main installed components of the steam loop are steam/water heat exchanger, steam flow control valve, steam trap, condensate piping.
2. The main installed components in the hot water loop are hot water VFD pump, air separator, pressure relief valve, expansion tanks, BTU meter, temperature mixer, ON/OFF solenoid valves
3. The steam/water heat exchanger, VFD pump, and temperature control valves 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. 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 heat source 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 evaporator and pre-heater of the power unit.

Heat Sink Loop:
1. The main installed components of the heat sink loop include the hose from fire hydrant to power unit, pump (not VFD), BTU meters, check valve, two 3-way flow and temperature control butterfly valves, bypass loop, temperature mixer, hose from power unit to a ditch.
2. The fire hydrant valve, 3-way butterfly valves, pump, and bypass loop are used for flow rate control and the water inlet temperature of the GM cooling side.
3. 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 check valve is to prevent the water flow back into the fire hydrant.

Electrical system:
1. The electrical system installation is basically the electrical wiring required for various components (ORC power unit, VFD pump, cold water pump etc.) in the whole test system and for safety concerns. The wiring diagram is given in Appendix IVB.
2. Electrical wiring has been successfully completed for uploading the power generated by the GM to UAF grid. Electrical wiring has also been completed for powering the VFD pump on hot water loop, pump on cold water loop, steam control valve, two 3-way bypass valves in cold water loop.
3. A Grid protection relay is used to protect the grid and the ORC system.

Instrumentation:
Performance characteristics of the ORC system and its components are obtained from measured data through data analysis. Performance characteristics are defined in terms of various input conditions of the heat source and heat sink. The performance characteristics can then be used for selecting appropriate rural Alaska diesel gen sets for this ORC system and planning waste heat distributions for optimal returns from waste heat applications. Operation and performance parameters to be detected and collected for this test are summarized into five groups and listed in Table 2. Besides the measurement devices originally equipped with

the ORC system, measurement devices for all the other parameters listed in Table 3 need to become available through purchasing or other means.

Remote Monitoring and Control Line:
The remote monitoring and control is done from both UAF and ElectraTherm facility in Nevada simultaneously. For this purpose the broadband internet connection cables, static IP address and modem which are installed at the UAF Power Plant experimental site is completed.

PROCUREMENT
Considering the integrated testing system (ORC system plus supporting testing system), equipment purchased include the 65kW ORC system and most of the components used in steam loop, hot water loop, the open cooling water loop, electrical components used for electrical circuit, and sensors and data acquisitions components used for measuring/monitoring and control. Table 3 shows the major components used for steam, heating, cooling system, and electrical systems. Major components purchased for measurement/monitoring and control system include a National Instrument PXI board and SEXI modulus. In addition, steel and black iron hot water pipes, steam rubber hoses, wires, cables, structural materials, and related connection and construction materials were also in the purchase list. All the purchases have been completed before December 2011 except the sensors needed for measuring temperatures and pressures related to the performance of the ORC components and two Watt meters for pump power measurements, of which the purchases were completed in mid-January of 2012.

Some of the components and construction tools (steam trap, steam to hot water heat exchanger, expansion tank, pipe grooving tool, cutting tool, welder, etc.) were borrowed from UAF facilities service and UAF power plant. One lesson learned is that procurement may become extremely time consuming for a remote city in a remote state. According to our experience, sometimes, even common structure materials may take a few days to become available.

Table 2 Operation and Performance Parameters for data acquisition

Hot water loop parameters	**Cold water loop parameters**	**GM parameters (sensors installed by our team)**	**GM parameters (sensors come with the ORC system)**	**Electric power parameters**
Steam Inlet Temperature to Heat Exchanger	Cold Water temp into GM Before Check Valve	GM Ambient Temp	GM Gross Power (kW) - NOT accurate	GM net power output (Watts)
Steam Condensate temperature	Cold Water temp into GM just before GM	GM Condenser Inlet Temp (sensor on pipe surface)	Temperature difference between hot and cold water supply temperatures (F)	GM Pump power (Watts)
Hot water temp out of Heat exchanger (to GM)	Cold Water temp out of GM after bypass valves	GM Condenser Outlet Temp (sensor on pipe surface)	GM pump VFD Hz	Hot water pump power (Watts)
Hot water temp into Heat exchanger (from GM)	Cold Water temp out of GM condenser before pump	GM Evaporator Inlet Temp (sensor on pipe surface)	Expander Hi Pressure (PSIG)	Cold water pump power (Watts)

GM Hot Water Inlet Temp	Cold water flow rate	GM Evaporator Outlet or Expander inlet Temp (sensor on pipe surface)	Expander Low Pressure (PSIG)	
GM Hot water outlet Temp	Heat rejected to cold water in GM condenser in MWH	GM Expander Outlet Temp (sensor on pipe surface)	Expander Rear Temp (F) (sensor on pipe surface)	
Steam Inlet Pressure		GM Pump Inlet Temp (sensor on pipe surface)	Bearing Temp (F)	
Steam Condensate Pressure		GM Pump Outlet Temp (sensor on pipe surface)	CW Supply Temp (F) (sensor on pipe surface)	
Steam Valve Position		GM Refrigerant Tank Temp (sensor on tank surface)	CW outlet Temp (F) (sensor on pipe surface)	
Hot water flow rate		GM Condenser Inlet Pressure	HW Supply Temp (F) (sensor on pipe surface)	
Hot water heat supply to GM in MWH		GM Evaporator Inlet Pressure	HW outlet Temp (F) (sensor on pipe surface)	
		GM Pump Inlet Pressure	Generator runtime hours	
		GM Pump Outlet Pressure	Lube oil pressure (PSIG)	
		Refrigerant Tank Pressure	Generator RPM	
		GM Hot Water Bypass Temp (sensor on pipe surface)	GM pump Power (kW)	
		GM Hot Water out of PH before bypass (sensor on pipe surface)	Total GM kWH from starting	
			GM Net Power (kW)	

Table 3 Selected Components for Heating and Cooling Loops

Component	Size	Reason for selection
Green machine (GM)	Power output: Max 50kW, Min 5kW Hot water supply: 180°F to 250°F. Cold water supply: 40°F to 110°F	The only commercial unit available in market at that time which can recover heat to power from diesel engine jacket water temperatures (200°F)
Steam to Hot water heat exchanger	i. Heat exchange rate: 4000MBTU/hr ii. Area = 111.00ft^2	i. Based on heat source requirement for GM. At 5.5% efficiency (worst case scenario), the GM would need 4000MBTU/hr for upgraded GM of 65kW. ii. Area of HX was selected with communication with heat exchanger manufacturer
Steam flow rate control valve with actuator	i. Valve Spec.: Valve flow coefficient (Cv): 160; ii. 4" Normally closed Siemens valve iii. Actuator: SKC62U	i. The reason for using steam flow control valve with actuator is to manipulate hot water supply temperature to GM from 155°F to 250°F. ii. The valve flow coefficient (Cv) was calculated from standard manufacturer's handbooks. The calculated Cv obtained was about 135.

Steam trap	Specification: SpiraxSarco FT-30; Maximum operating Pressure: 30psig; Maximum Temperature: 45°F of superheat at all operating pressures	Steam trap was selected based on power plant steam supply pressure range of 18psig to 24psig.
VFD pump for hot water loop	i. Bell & Gossett 20hp pump (1750rpm) rated for VFD operation. ii. Rated for 250gpm and head of 116feet of water	i. Pump was selected based on the pressure drop in the hot water piping calculated using pressure drop across various components. ii. Pump was also selected in communication with pump manufacturer. iii. VFD is for various flow rates of hot water.
Expansion tank	Extrol SX-40V; Pre-charged to 12psi.	Expansion tank volume selection: ASRAE handbook standard procedure with a factor of safety margin (for high temperature and above atmosphere pressures of hot water). The calculated expansion volume based on handbook was 3.5gal.
Air separator	Bell & Gossett Rolairtrol R-4F air separator: 300gpm and 4" flange	Air separator selection was based on the maximum hot water flow (300gpm) in the loop.
Pressure relief valve	Bell & Gossett 45psi pressure relief (PR) valve	45psi PR is used to accommodate for above boiling water temperature of 235°F.
BTU meters	Kamstrup BTU meters on hot water loop and cold water loop	For measuring the heat supplied by hot water to GM and heat rejected to cold water by GM
3-way butterfly valve	Triad 4" 3-way butterfly valves with double acting actuator	Two three way butterfly valves were used in the cold water bypass loop to control the cold water supply temperature to GM by recirculating the warm water coming out of GM
Cold water pump	15hp Scott Pump with 3500rpm	Cold water pump will only be used during the performance testing of the GM when the recirculation of warm water is needed to test the GM at varying cold water temperatures.
Check valve	Walworth 3" 150psi check valve	To prevent cold water flow back into fire hydrant
Electrical cable for uploading power	3/0 metal-clad (MC) three conductor cable with ground (Due to unavailability of 1/0 cable and time concern)	The cable size was selected based on the distance between the GM and power plant motor center (point for power up-loading to UAF grid), voltage drop, and heat generated due to cable resistance. According to standard handbook calculation 1/0 cable was enough to serve the purpose.
Grid protection relay	Beckwith grid protection relay model # M-3410A	To protect mainly the utility (grid system) from GM and to protect the GM itself. If there is large voltage (±10%) or frequency fluctuations in the utility system the GPR sends a signal for GM shutdown.

Chapter 5
Installation and Instrumentation

Installation and instrumentation started around mid November 2011 and most of the work has been done within 3 weeks. The rest, which includes sensors to the GE components and Watt meters to the hot water and cold water pumps, were installed in January 2012. Detection, rectification, and modifications were also part of this task. All the installation work (e.g., wiring, piping) followed existing industrial standards and codes and also followed Chapter 1 (Green Machine Installation Requirements) of the ElectraTherm manual. Since regulations, safety, and reliability are important to this project, experts (e.g. contractors, UAF power plant, UAF Facilities Service, ACEP, and CEM) in related areas were consulted for piping, wiring, and control system design and installation. All the work was done internally by UAF personnel, including engineers in the units of the Facilities Service, Power Plant, and ACEP. Other individuals contributed to this work including many students and a few CEM machine shop and ME Department staff and faculty members.

A brief list is given below describing relative important information related to the installation and instrumentation (Please also refer to Figure 4, Table 2, and Table 3):

Steam Loop:
Piping: Steel pipe or steam rubber hose (300°F and 30psi).
Joints: Threaded with sealant or welding.
Steam/hot water heat exchanger: Eccentric condensate flange for outlet of steam side.
Heat exchanger: A designed frame is used to keep an appropriate height of the water outlet position.

Hot Water Loop:
Piping: 4” black iron pipe with GruLok joints. Struts structures are used to support the 150ft pipe system at many critical locations.
20 hp pump: A designed structure is used to support the potential large bending load.
Pipe entering and exiting the ORC system and by pass pipe (including a control valve): Steel pipe provided by the ORC manufacturing for entering, exiting, and by pass pipes. A designed supporting structure is required for the overhung pipes and the valve.
BTU meter (ultra sonic): 5D distance is required in the upstream side. The flow meter need to be installed in the cold side pipe.

Cold Water Loop:
Piping: 4” rubber hose and fire hydrant clamps are used. A check valve is also installed to prevent reverse flow.
Pumps: Need structure to support the bending load.
Inlet, exiting, and by pass pipes (including controlled valves) of the ORC system: Support structure is required for the overhung pipes and the valve.
BTU meter (ultra sonic): 5D distance is required in the upstream side.

Electrical Circuit:

Up Load Contactor: 3/0 MC Cable with 3 conductors and a ground. Ceiling tray is used to harness the cable.
Pump: Cable has 3 lines and a ground.

Instrumentation and DAQ:
Sensors installed: Numerous thermal couples are used for temperature measurements, two thermistors are for the two BTU meters, 6 manual pressure gauges, 7 digital pressure gauges, 2 Watt meters, and 2 flow meters (i.e. BTU meters). All digital signals connected to the NI SCXI 1001 box.

DAQ system: It include a NI PCI-MID-16E-1 board and NI SCXI modules (1102, 1124, four 1120, and Two 1121 modules).

Internet lines: One line connects to the power plant server and the other directly to the local computer.

Appendix V shows pictures of the integrated system after installation.

Chapter 6
Commissioning

The commission started on December 12, 2012 and lasted for about a week. Before commissioning, the ORC system (the Green Machine or GM) user needs to install heat source, heat sink, electrical circuit, and a dedicated internet line (as mentioned in the previous chapters) to the GM based on the requirement list provided by the manufacturer of GM, ElectraTherm (ET). The completed GM Installation Requirement list and a Pre-Commissioning Check list need to be received by ET before any engineer can be dispatched for commissioning.

During the commissioning, the ET commissioning engineer performed the following:

- Checked the installation of the supporting systems: This included investigation of all the installed heating, cooling, and electrical systems and the internet connection. There were many discussions between the ET engineer and the UAF engineers.
- Checked the Green Machine: A double check of the information and locations of the labels, installations and functionalities of the components, match of the specification received by the customer and the specification of the manufacturer's record, etc.
- Working fluid transferring: Conducted pressure test and transferred working fluid from storage bottles into GM. Details could be found in Chapter 5 (Fluid Transferring) of the GM manual.
- Parameter setup: Set up start parameters, operation parameters, default parameters, and other parameters using HMI. Details could be found in Chapter 7 (Set Up) of the GM manual.
- Starting up of the GM: Tested of emergency stop, operation mode, alarm mode, and PLC-IO, etc. Details could be found in Section 3.4 of the (Operation) and Chapter 7 (Distributor Manual) of the GM manual.
- Training: A comprehensive training was held for the ACEP engineers involved in this project. The training includes the most important materials covered in the GM manual, which include: information of installation, routine maintenance and special tools, operation, background, fluid transferring, tasks and applications of HMI. This part training is for general audience (many of the ACEP engineers, students, faculty and staff members).
- Hands on system set up and operation training: This training is for an engineer and a graduate student who would perform the GM test.
- In addition, some design issues have been discussed. Two important examples are listed below:
    1. All the components of GM have been designed for the capacity of 65kW, but the system was rated as a 50kW system. This down grade was due to the lack of an emergency by pass valve to prevent the screw expander from over speeding while the engine is shutting-off. This unit can be modified with a retrofit to make it operate at 65kW safely.
    2. To start the GM, the temperature of the heating fluid entering into the GM cannot be too high. Otherwise, the screw expander will be over speed and the

engine will be shutting down immediately. This is due to the methodology used to control the starting working fluid flow rate.

Chapter 7
Experimental Setup and Testing Schedule

EXPERIMENTAL SETUP:
The experimental site was set up in the UAF power plant. The set up includes a 50kW GM heat engine (the ORC system), a heating source system to supply heat to GM, a cooling sink loop to absorb the expelled heat from GM, electrical cables for both up loading the generated electricity and electrical power consumption to the operation pumps of the heating and cooling loops. Some of the specification information includes electrical characteristics of the GM (i.e. 450-500 VAC/2 phase/100/60Hz, 50 kW, 24VDC), hot and cold water connection (heating source 190°F-240°F and 175gpm, cooling source 40°F-110°F and 250gpm, working fluid R345f), mechanical specifications (8'x6.5'x7.5' and 7,300 lbs), environmental specifications (storage 20°F-140°F, relative humidity 0.95, noise 96db at 3'), and controls (inputs- HMI, physical switch, and emergency stop button; output- relay contacts for heating and cooling water supplies, hot water by-pass, and dry cooler control; Remote control- remote monitoring and machine control via internet). Set-ups for heating system, cooling system, external electrical system, data acquisition system, and internet connection have been discussed in details in Chapter 4 (Test Plan, Design and Selection of Components, and Procurement) and Chapter 5 (Installation and Instrumentation).

TESTING SCHEDULE:
Reliability test is a test for 600 hours with the GM under full capacity under a constant operation condition (Planned: heating source at 160gpm and 225°F; cooling source at 160gpm and 50F).

Performance test has been conducted for the GM running under different combinations of cooling and heating conditions, which are listed in Table 4. Each case was operated for 30 minutes for data collection.

Table 4 Different Conditions of Heating and Cooling Sources Adopted for Performance Test. (Cooling source temperature = 50°F)

Heating Temperature (^{o}F)	155	175	195	215	225
Heating Flow Rate (gpm)	120	160	200	250	300
Cooling Flow Rate (gpm)	120	160	200		

According to Table 4, the total number of possible cases is 75 (5x5x3). The number of cases to be investigated will be increased for another cooling flow temperature (60 ^{o}F), if the GM will stay in the current test site for another month and cooling water supply is still available.

Chapter 8
Reliability Testing and Results

PREPARATION:
In the preparation work the main focus is on setting up the system (Green machine, hot water loop and cold water loop) parameters for smooth running of the GM for reliability and performance tests.

GM SETUP PARAMETERS:
GM setup parameters already come with default factory settings which are good for normal operation conditions and need no change in the settings. GM setup has six Human Machine Interface (HMI) screens; of which only three screens are important for setup and startup. The six HMI screens are "Setup", "Startup Parameters", "Options", "Machine Defaults", "PLC I-O" and "Veris Setup". Among these six screens first three are only considered for setup and start of GM which are explained in detail in below tables, the rest of three screens are also shown for reader's reference.

1. Green Machine HMI screen: "Setup"

GM setup parameters screen is shown in Figure 5 and the parameters are explained in detail in Table 5 below.

Table 5: GM setup parameters table with range and default values

Parameter and Description	Range	Default Value
DELTA T SHUTDOWN: This is the hot water and cold water supply temperature difference	50^oF to 80^oF	80^oF
ENTER MINIMUM HZ RANGE: This is the GM pump minimum frequency	15Hz to 30Hz	22.0Hz
ENTER MAXIMUM HZ RANGE: This is the GM pump maximum frequency	30Hz to 60Hz	58.0Hz
MIN KW NET OUTPUT: This is the GM minimum net power output	5kW to 25kW	10.0kW
MAX EXP PRESSURE: This is the maximum expander inlet pressure	130PSI to 200PSI	185PSI
EXP DIF PRESSURE MIN: This is the minimum expander differential pressure	0PSI to 70PSI	32PSI
ENTER "ON GRID": This is the expander RPM at which it should go on grid	1400rpm to 1800rpm	1730rpm
ENTER MAXIMUM KW: This is the maximum gross power output required from the GM.	10kW to 50kW	50.0kW
HW SHUTDOWN TEMP F: This is the maximum hot water supply temperature	190F to 250F	235F

at which GM will shutdown		
ENTER POWER FACTOR: This is the power factor which is required to match the frequency of GM to frequency of power utility to which GM power is uploaded	0.60 to 1.00	1.00
ENTER FULL LOAD AMPS: This is the current amperes the GM needs to generate at full load	30A to 130A	75A
ENTER LINE VOLTAGE: This is the line voltage of the utility to which the GM power is being uploaded	240V to 500V	480V

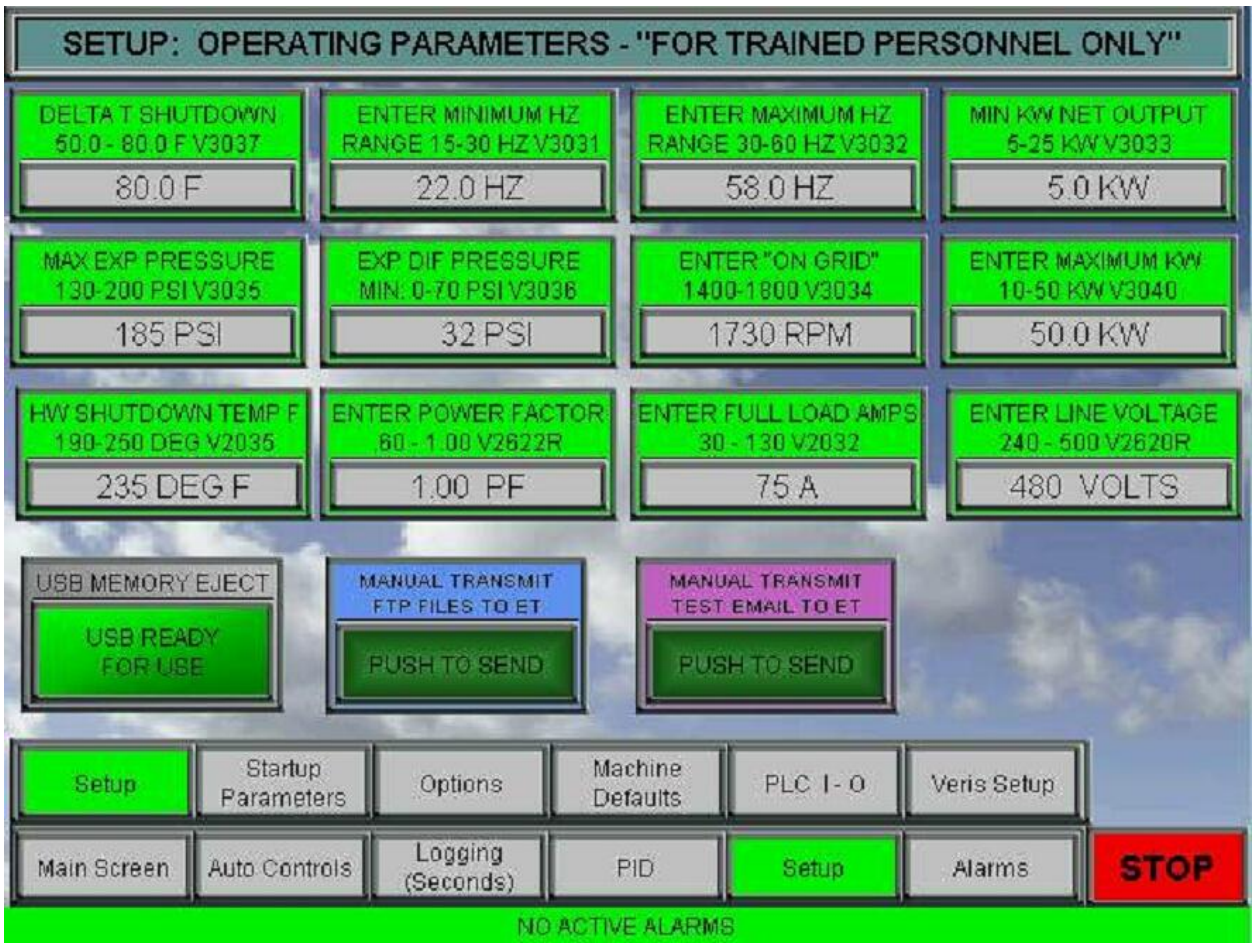


Figure 5: GM "Setup" parameters screen

2. Green Machine HMI screen: "Startup Parameters"

GM startup parameters screen is shown in Figure 6 and the parameters are explained in detail in Table 6 below.

Table 6: GM startup parameters table with range and default values

Parameter and Description	Range	Default Value
HW MIN START TEMP F: Hot water minimum supply temperature to start the GM	150°F to 230°F	150°F
HW MAX START TEMP F: Hot water maximum supply temperature to start the GM.	185°F to 250°F	230°F
HW RANGE MINUTES: This is the time required for the hot water to make one complete loop.	0 to 20minutes	20.0 minutes
MIN START DELTA T: This is the minimum temperature difference between hot water and cold water supply for starting the GM	80°F to 110°F	80°F
TIME BETWEEN STARTS: This is the time GM waits to start after shutdown due to not serious issues such as low hot water supply temp, Delta T shutdown etc.	1sec to 360secs	30secs
START DELAY: This is the time GM waits after the “TIME BETWEEN STARTS” ends and keeps on trying to start at this regular intervals	1sec to 600secs	15secs
KW STARTING SETPOINT CONFIGURATION PARAMETERS: This parameters are used to set the initial kW the GM needs to produce at the startup based on the hot water supply temperature and temperature difference between hot and cold water.	N/A	Factory default

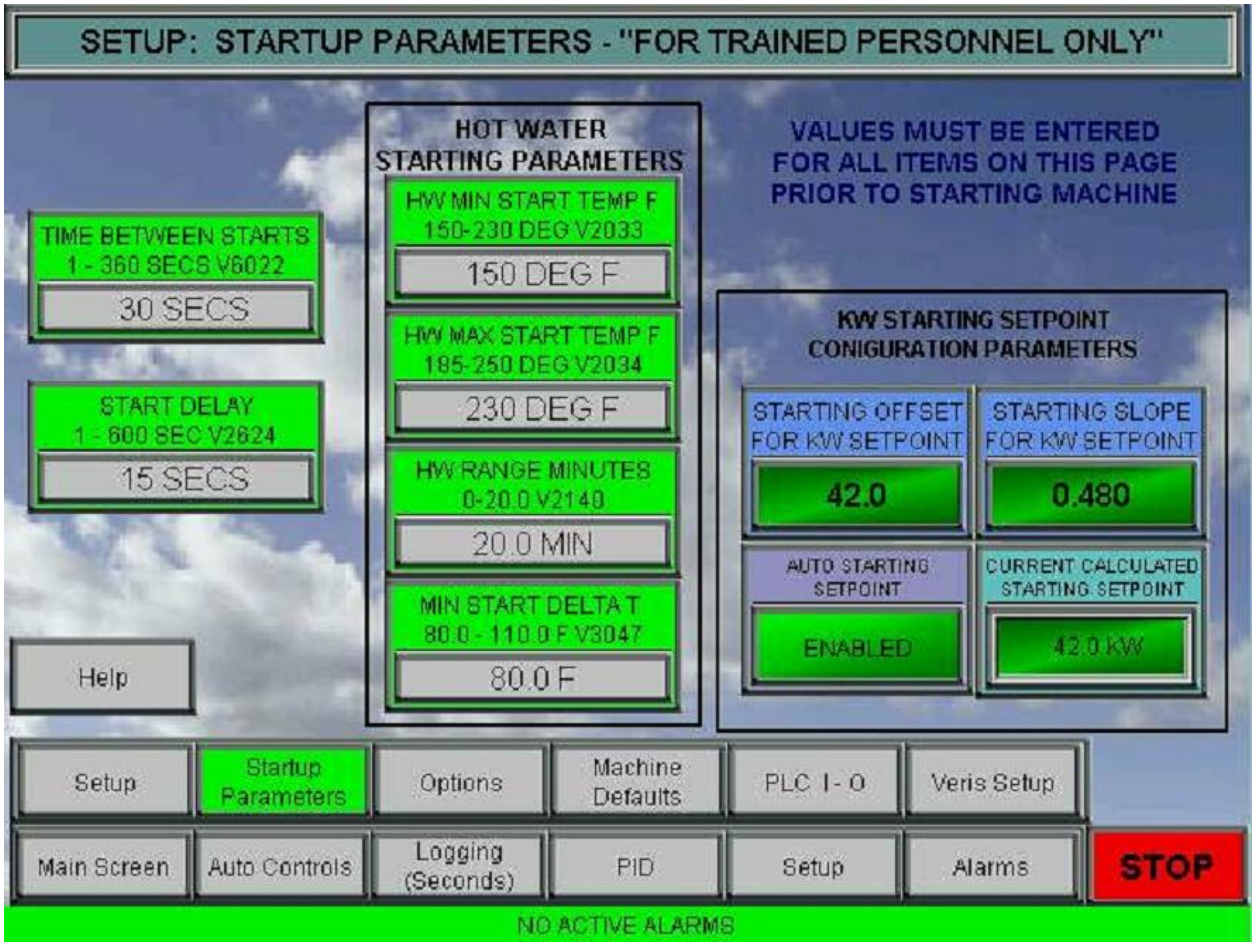


Figure 6: GM "Startup Parameters" screen

3. Green Machine HMI screen: "Options"

GM options parameters screen is shown in Figure 7 and the parameters are explained in detail in Table 7 below

Table 7: GM options parameters table with range and default values

Parameter and Description	Range	Default Value
CW FLOW SWITCH: Cold water flow switch installation	N/A	INSTALLED/NOT INSTALLED
CW FLOW METER: Cold water flow meter installation	N/A	INSTALLED/NOT INSTALLED
AIR/LIQUID COOLING: This describes the type of cooling system installed	N/A	LIQUID/AIR
GRID PROT RELAY: Grid protection relay installation	N/A	INSTALLED/NOT INSTALLED
REF BYPASS VALVE: Refrigerant bypass solenoid valve position i.e. OPEN when the GM is not running and CLOSE when GM is running	N/A	OPEN/CLOSE

VFD MANUAL RUN: It is used to check the working of GM pump manually by pressing this tab	N/A	ON/OFF
GENERATOR MANUAL RUN: It is used to check the working and direction of rotation of GM expander and generator	N/A	ON/OFF
GEN NAMEPLATE RPM: This is generator rated RPM and is factory default to 1830RPM	1500RPM to 3700RPM	1830RPM
ENTER ENCODER PPR	100ppr to 1024ppr	600PPR

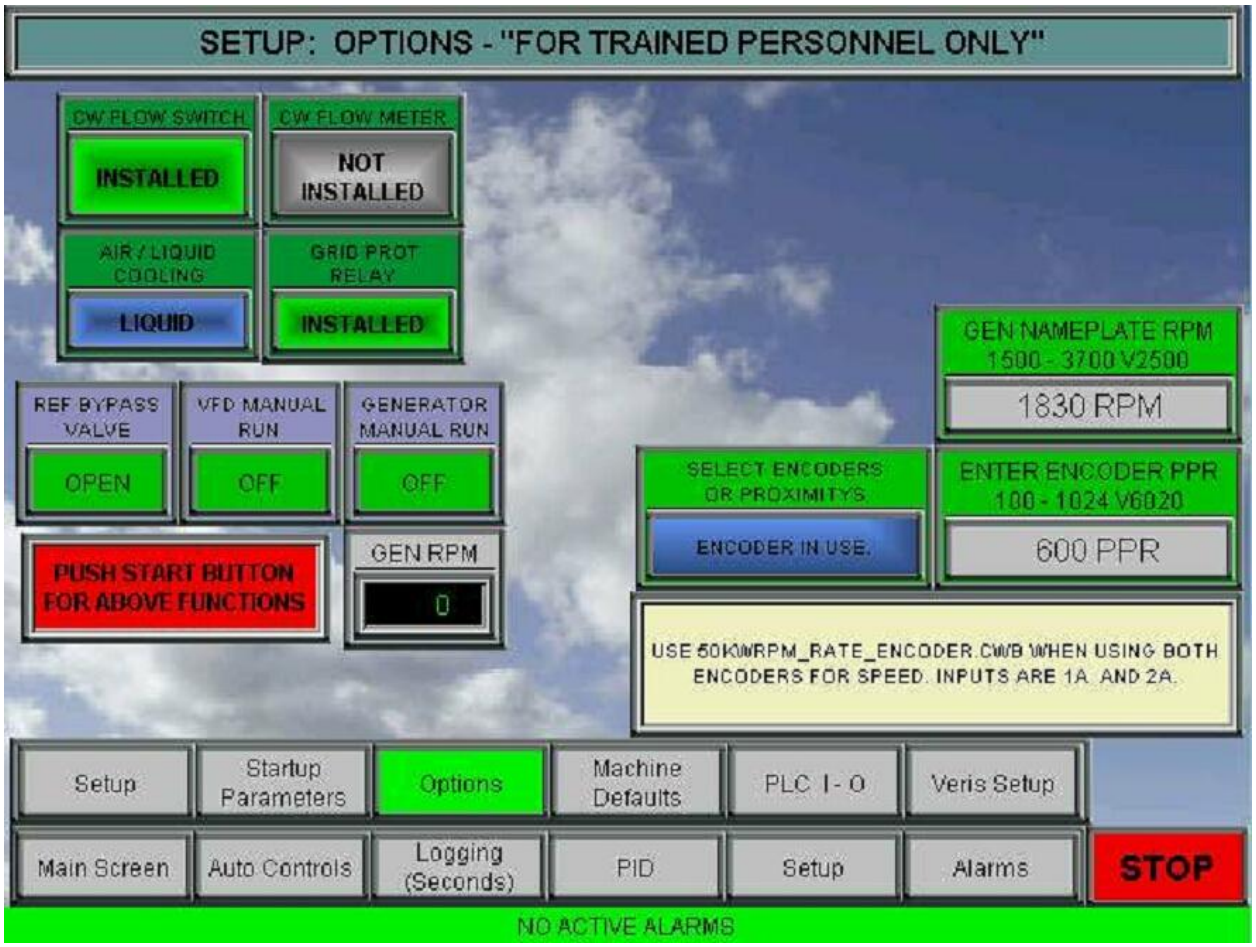


Figure 7: GM "Options" parameters screen

4. Green Machine HMI screen: "Machine Defaults", "PLC I-O" and "Veris Setup"

The following GM HMI screens are not generally used in GM setup or startup. These screens are for GM user reference purpose only. The "Machine Defaults" screen (Figure 8) comes with all factory default settings. The other two screens "PLC I-O" (Figure 9) and "Veris Setup" (Figure 10) are the GM data acquisition screens.

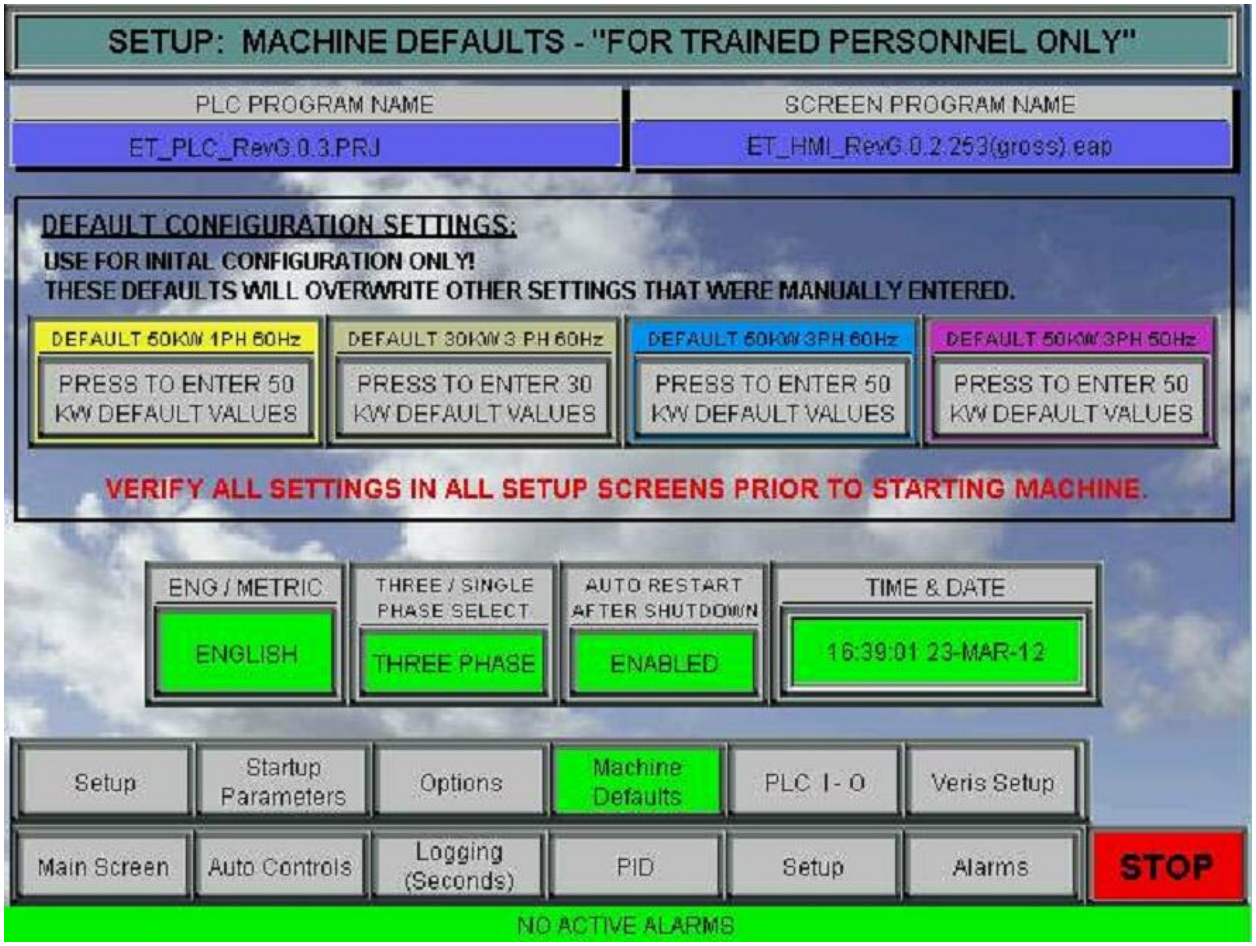


Figure 8: GM "Machine Defaults" parameters screen

Figure 9: GM "PLC I-O" parameters screen

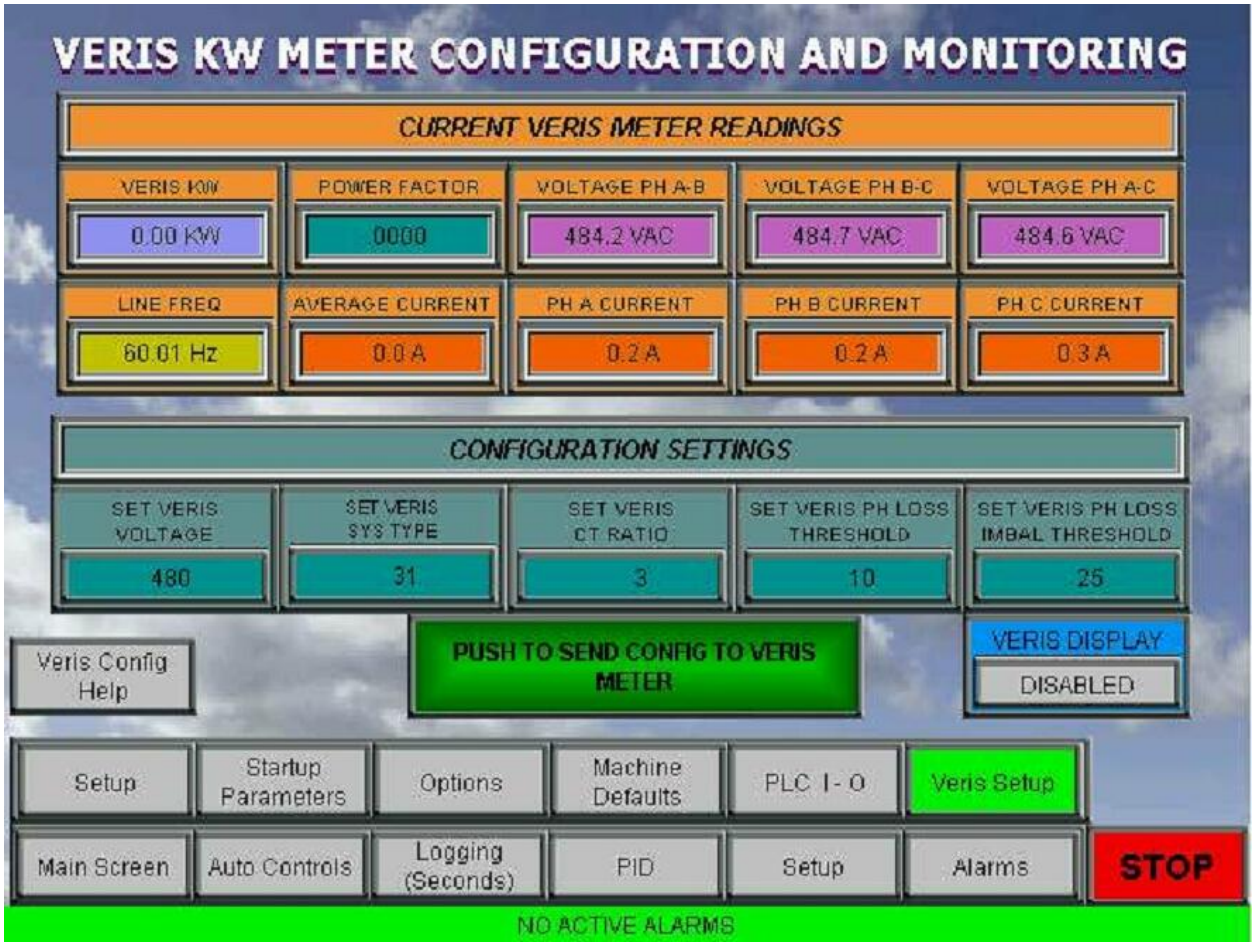


Figure 10: GM "Veris Setup" parameters screen

Hot WATER LOOP SETUP PARAMETERS:

On the hot water loop parameters to setup before starting GM are:

1. Check for leaks in the hot water system at Thread-O-Let connections, Gruvlok connections, elbows, joints etc.
2. Check the pump rotation direction by turning on the pump for very short period of time (2 seconds).
3. Check for pumps inlet and outlet pressure are within the operating range of the system i.e. pressure relief valve located at pump inlet is a 45psi.
4. Check the flow meter; make sure the direction of flow meter and flow meter gets the flow reading.
5. Check the pump VFD i.e. as we change the VFD frequency, the pump responds it by changing the flow rate.
6. Check that we can get the required temperature by opening the steam manual valve and operating the steam automatic valve using LABVIEW.

Cold WATER LOOP SETUP PARAMETERS:

On the cold water loop parameters to setup before starting GM are:

1. Check for leaks in the cold water system at Thread-O-Let connections, butterfly valves, Gruvlok connections, elbows, joints etc.

2. Check the pump rotation direction by turning on the pump for very short period of time (2 seconds).
3. Check the flow meter; make sure the direction and orientation of flow meter and flow meter gets the flow reading.
4. Check the operation of bypass valves, bypass loop for the direction of cold water flow for cold water temperature control.

OPERATION PROCEDURE:
Operation procedure explains the sequential steps in starting the GM, the main steps involved in this process are explained below.

1. Check the cold water is flowing through the GM condenser. This can be checked by looking at the GM HMI screen, which shows the "CW FL SW" (cold water flow switch) is "ON" with green tab color indicating the cold water flow.
2. On the GM HMI screen, if "MCR" is shown "OFF" with gray tab color, then press the "ENABLE" button on the GM front panel (not on HMI screen) to bring master control relay (MCR) into "ON" position with green tab color.
3. With the cold water flow switch and MCR in "ON" position the GM HMI screen should show "START" button.
4. GM cannot be started directly at hot water temperature of 225°F as it will stop abruptly due to over speed. For reliability test first set the hot water flow rate at 160gpm (by adjusting the VFD frequency) and 200°F (by operating steam valve using the LABVIEW).
5. For reliability test set the cold water flow rate at 160gpm.
6. Now the GM should be ready for start and hit the "START" button on the GM HMI screen.
7. After running the GM for 15minutes at 200°F of hot water supply temperature, increase the hot water temperature to 225°F for continuing with the reliability test.

CHECK LIST:
Check list is the sensors, gauges, joints etc., located on the Green machine itself, hot water loop piping and cold water loop piping, which needs monitoring from time to time while the GM is in operation. GM itself does not require any special inspection every day, as observed from reliability test. Following table (Table 7) gives the list of sensors and there location which could be monitored from time-to-time (weekly basis).

Table 7: Check list green machine, hot water and cold water loop

Sensor or Gauge	Location
Hot water and cold water flow rates	Respective flow meters
Hot water supply temperature	Data acquisition system (if manualthermometer is installed, than at hot water inlet to GM)
GM evaporator and condenser pressure drop on hot water and	Across GM Evaporator and Condenser

cold water loop side respectively.	
Hot water pump pressure boost	Hot water pump inlet and outlet pressure
Visual inspection of all joints	Gruvlok couplings, elbows thread-o-lets etc.,

RELIABILITY TEST RESULTS:

Reliability test is the operation of GM at full load (i.e. 50kW) for 600 hours. During this phase of test important observations would include changes in GM performance for long term run at full load, noting GM shutdowns, reason for shutdown and difficulty in solving this problem. The following table (Table 8) gives the GM gross and net power output during three different times of reliability test. This table also provides details of the hot water and cold water supply conditions. Figure 11 gives the GM HMI screen-shot at 600th hour of reliability test.

Table 8: Reliability test results at three different times of the test

Date and time	**Hot water supply temperature and flow rate**	**Cold water supply temperature and flow rate**	**GM gross output (kW)**	**GM net output (kW)**
12/23/2011; 11:15:22 AM	219.2^{o}F and 160gpm	51.4^{o}F and 160gpm	50.1	46.34
1/6/2012; 4:59:48 AM	219.4^{o}F and 160gpm	59.1^{o}F and 160gpm	50.1	46.28
2/23/2012; 5:45:45 PM	218°F and 160gpm	51.2^{o}F and 160gpm	50.1	46.51

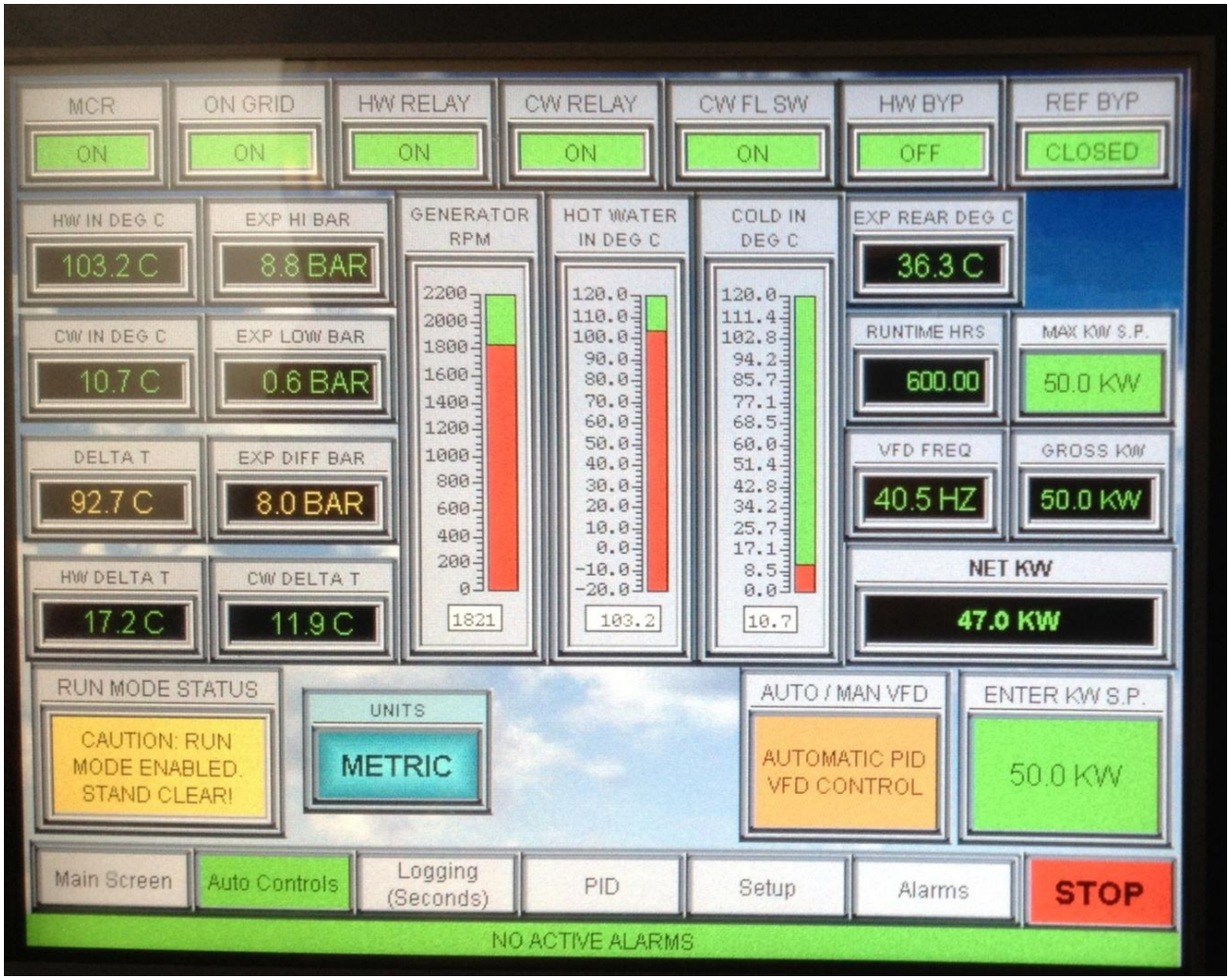


Figure 11: Green machine HMI screen-shot during reliability test operation

GREEN MACHINE SHUTDOWN DURING RELIABILITY TEST:

One January 3rd 2012, during the reliability test, GM automatic shutdown was observed. The shutdown occurred very frequently (nearly 7 times) and the "Alarms" list of the HMI screen showed the shutdown reason was due to "Expander high pressure switch limit has been exceeded". When contacted with the ElectraTherm (GM manufacturer), they noticed that the expander high pressure switch malfunctioning. Expander high pressure switch is one of many safety features of GM which will turn-off the GM when the high pressure of the working fluid exceeds beyond the designated limit (see Figure 1). To continue with the reliability test we have bypassed the expander high pressure switch and started the GM again. The rest of the reliability test was smooth without any glitch.

The frequency of "Expander high pressure switch" failure is very rare and according to experts those types of switches will work for lifetime without failure. However ElectraTherm has sent us new high pressure switch which will be installed into GM after all the tests at UAF are completed and ready to be shipped to village location for permanent installation.

Chapter 9
Performance Testing and Results

Performance test:

Performance test on GM is conducted mainly to learn the performance characteristics of the GM and its components (expander, evaporator, condenser, pump etc.) at different hot water and cold water flow rates and temperatures. The purpose of this test is also to create performance characteristic charts of the GM at different heat source and heat sink conditions. The GM was planned to be tested at 5 different hot water flow rates, 5 different hot water temperatures, 3 different cold water flow rates and 2 different cold water temperatures. The list of hot water and cold water temperatures at which GM will be tested is given in Table 9. The planned total number of performance tests that would be conducted on GM to be 150. The procedure of performing this test (i.e. method of changing temperatures and flow rate, data collection etc) is explained in "Operation Procedure" section of this chapter.

Preparation work and check list for performance test is similar to reliability test and the readers are requested to refer to the chapter "Reliability test".

Table 9: Various hot water and cold water flow rates at which GM will be tested

Hot water temperatures (^oF)	Hot water flow rate (gpm)	Cold water temperatures (^oF)	Cold water flow rate (gpm)
155	120	50	120
175	160	60 (tentatively)	160
195	200		200
215	250		
225	300		

Operation Procedure:

1. First the cold water flow rate is set at desired value by turning the manual flow rate valve near the fire hydrant. The temperature of cold water from fire hydrant is around 50°F.
2. At this cold water flow rate we than set the desired hot water supply temperature to GM by operating steam flow control valve using LABVIEW software.
3. Now by varying the hot water pump VFD frequency (e.g. VFD frequency of 24Hz corresponds to 120gpm of hot water flow), we set the desired hot water flow rate. The hot water flow rate can be read in BTU meter display is in cubic-meter/hour.
4. After setting all the four parameters (hot water and cold water flow rates and temperatures) at desired conditions, we wait for approximately 15minutes for steady state data collection.
5. Steady state data collection is done for 30minutes at one set of hot water and cold water temperature and flow rate. All the sensor data (i.e. temperatures, pressures,

flow rates etc.) will be stored in a excel format. This completes the performance test for one set of hot water and cold water flow rate and temperature.

6. Now we change the hot water flow rate to next value (e.g. 120gpm to 160gpm) by varying the VFD frequency, keeping the other three parameters same. Then Step-4 and Step-5 above are repeated. In this manner we continue performing the tests at other hot water flow rates.
7. Now we change the hot water supply temperature using Step-2 and repeat Step-3, Step-4 and Step-5 for different hot water flow rates.
8. Step-2,Step-3, Step-4 and Step-5 are repeated iteratively for three different cold water flow rates listed in Table 9

Results:

The data listed in the below tables (Tables 10 to 14) do not include cases involving cold water temperature= 60°F. This was due to the delay of the test schedule resulted from the frequent shortages of cooling water and the defect of recirculation circuit of the fire hydrant syatem (cooling water source). Since a10^{o}F difference (from 50°F to 60°F) in cooling water temperature may not cause noticeable characteristic changes of the system performance and the nature of the existing cooling source may not be manuverable to give larger temperature difference, the cases involves cooling water temperature= 60°F will not be included in this report.

Table 10A: Performance results for HW Temp= 155°F; HW flow rate = 120gpm to 300gpm; CW Temp 50°F and CW flow rate = 120gpm, 160gpm, and 200gpm

#	Hot water flowrate - Average (GPM)	Hot water Supply Temp - Average (F)	Hot water return Temp -Average (F)	Cold water flowrate - Average (GPM)	Cold watersupply Temp - Average (F)	Cold water return Temp - Average (F)	GM Net Power - Average (kW)	GM Pump Power - Average (kW)	Hot water pump power - Average (kW)
1	120.707	156.442	141.898	119.603	53.126	66.669	13.545	0.869	1.008
2	159.055	156.369	144.642	119.209	52.741	66.784	14.680	0.936	1.799
3	200.640	156.454	146.765	119.811	52.682	67.093	15.509	0.968	3.170
4	250.899	156.564	148.617	119.341	52.687	67.346	15.888	1.002	5.830
5	300.756	156.588	149.739	119.932	52.651	67.600	16.366	1.046	9.694
6	120.592	157.934	143.110	157.326	52.635	63.007	14.554	0.914	1.005
7	159.151	158.195	146.303	158.502	52.275	63.013	15.730	0.962	1.781
8	201.719	156.649	147.320	164.440	52.989	63.189	15.362	0.945	3.172
9	251.794	158.933	150.640	156.831	52.116	63.726	17.865	1.068	5.812
10	302.198	155.490	148.699	163.723	51.823	62.725	17.120	1.053	9.804
11	122.655	156.079	141.037	208.293	49.878	57.870	15.937	0.961	1.040
12	162.509	155.815	144.067	214.308	50.806	58.669	16.533	0.984	1.869
13	203.469	156.209	146.522	204.502	51.090	59.687	17.049	1.020	3.214
14	251.237	156.390	148.275	204.269	51.114	59.834	17.768	1.067	5.811
15	300.157	156.990	149.981	204.574	51.107	60.096	18.447	1.107	9.666

Table-10B: Induced performance results from measured readings of Table-10A

#	Hot water heat input (kW)	Cold water heat rejected (kW)	Cold water pump Power (kW) *	GM Overall Net Power (GM, HW, CW pump) (kW)	GM Gross Power (NO pumps) (kW)	Overall efficiency (GM, HW, CW pump) (%)	GM net efficiency (GM pump) (%)	GM gross efficiency (NO pumps) (%)
1	257.216	237.311	1.008	11.528	14.414	4.482	5.266	5.604
2	273.273	245.269	1.008	11.872	15.616	4.345	5.372	5.714
3	284.830	252.960	1.008	11.330	16.477	3.978	5.445	5.785
4	292.096	256.300	1.008	9.050	16.890	3.098	5.439	5.782
5	301.806	262.678	1.008	5.663	17.411	1.876	5.423	5.769
6	261.905	239.079	1.799	11.750	15.468	4.486	5.557	5.906
7	277.285	249.353	1.799	12.149	16.691	4.382	5.673	6.020
8	275.682	245.752	1.799	10.391	16.307	3.769	5.572	5.915
9	305.947	266.759	1.799	10.253	18.932	3.351	5.839	6.188
10	300.688	261.501	1.799	5.517	18.172	1.835	5.693	6.044
11	270.302	243.881	3.160	11.737	16.898	4.342	5.896	6.252
12	279.723	246.893	3.160	11.505	17.518	4.113	5.911	6.263
13	288.765	257.580	3.160	10.675	18.069	3.697	5.904	6.257
14	298.674	260.981	3.160	8.797	18.834	2.945	5.949	6.306
15	308.228	269.423	3.160	5.621	19.554	1.824	5.985	6.344

Table 11A: Performance results for HW Temp= 175°F; HW flow rate = 120gpm to 300gpm; CW Temp 50°F and CW flow rate = 120gpm, 160gpm, and 200gpm

#	Hot water flow rate - Average (GPM)	Hot water Supply Temp - Average (F)	Hot water return Temp - Average (F)	Cold water flow rate - Average (GPM)	Cold water supply Temp - Average (F)	Cold water return Temp - Average (F)	GM Net Power - Average (kW)	GM Pump Power - Average (kW)	Hot water pump power - Average (kW)
1	119.792	175.103	156.016	119.559	52.530	69.813	21.454	1.319	1.007
2	156.636	175.534	160.259	119.528	52.442	70.659	22.929	1.404	1.782
3	199.992	175.874	163.464	119.740	52.704	71.330	23.914	1.482	3.263
4	249.819	176.085	165.707	119.535	52.591	71.847	24.973	1.566	6.029
5	300.065	176.328	167.465	119.886	52.683	72.323	25.456	1.618	10.154
6	121.583	175.004	156.233	164.220	51.978	64.718	22.353	1.316	1.030
7	162.588	174.766	160.389	163.805	51.914	64.898	22.969	1.371	1.953
8	201.221	175.091	163.120	163.439	51.949	65.265	23.740	1.424	3.339
9	251.698	175.144	165.252	164.086	51.831	65.296	24.298	1.483	6.098
10	299.345	174.865	166.074	163.343	51.480	65.688	25.994	1.603	10.037
11	121.986	175.272	156.029	204.885	50.836	61.245	23.606	1.395	1.040
12	161.947	175.739	160.478	204.265	50.773	61.767	25.274	1.494	1.929

13	202.359	176.799	164.560	205.113	51.977	62.922	25.072	1.478	3.363
14	252.320	175.808	165.270	204.768	51.126	62.712	26.938	1.642	6.146
15	299.307	175.655	166.574	205.432	50.775	62.569	27.545	1.687	10.040

Table-11B: Induced performance results from measured readings of Table-11A

#	Hot water heat input (kW)	Cold water heat rejected (kW)	Cold water pump Power (kW) *	GM Overall Net Power (GM, HW, CW pump) (kW)	GM Gross Power (NO pumps) (kW)	Overall efficiency (GM, HW, CW pump) (%)	GM net efficiency (GM pump) (%)	GM gross efficiency (NO pumps) (%)
1	334.987	302.741	1.008	19.438	22.773	5.803	6.404	6.798
2	350.555	319.011	1.008	20.139	24.333	5.745	6.541	6.941
3	363.618	326.762	1.008	19.643	25.396	5.402	6.577	6.984
4	379.866	337.239	1.008	17.935	26.539	4.721	6.574	6.986
5	389.606	344.961	1.008	14.293	27.073	3.669	6.534	6.949
6	334.365	306.532	1.799	19.524	23.669	5.839	6.685	7.079
7	342.466	311.602	1.799	19.216	24.340	5.611	6.707	7.107
8	352.912	318.846	1.799	18.601	25.164	5.271	6.727	7.130
9	364.767	323.723	1.799	16.401	25.781	4.496	6.661	7.068
10	385.533	340.023	1.799	14.158	27.597	3.672	6.742	7.158
11	343.902	312.457	3.160	19.407	25.002	5.643	6.864	7.270
12	362.100	329.016	3.160	20.186	26.768	5.575	6.980	7.392
13	362.837	328.920	3.160	18.550	26.550	5.112	6.910	7.317
14	389.566	347.580	3.160	17.632	28.580	4.526	6.915	7.336
15	398.242	354.993	3.160	14.345	29.232	3.602	6.917	7.340

Table 12A: Performance results for HW Temp= 195°F; HW flow rate = 120gpm to 300gpm; CW Temp 50°F and CW flow rate = 120gpm, 160gpm, and 200gpm

#	Hot water flow rate - Average (GPM)	Hot water Supply Temp - Average (F)	Hot water return Temp -Average (F)	Cold water flow rate - Average (GPM)	Cold water supply Temp - Average (F)	Cold water return Temp - Average (F)	GM Net Power - Average (kW)	GM Pump Power - Average (kW)	Hot water pump power - Average (kW)
1	119.627	194.753	170.587	119.373	52.678	74.673	29.914	1.930	1.002
2	161.073	194.962	175.664	119.599	52.316	75.744	32.187	2.133	1.859
3	201.688	195.061	178.835	118.657	52.402	76.474	33.437	2.270	3.132
4	251.584	195.263	181.700	119.071	52.461	77.405	34.831	2.421	5.754
5	300.870	195.362	183.792	119.528	52.473	77.643	35.433	2.487	9.575
6	119.831	195.716	171.383	157.997	51.881	68.806	31.164	1.907	1.010
7	157.969	195.853	175.535	158.123	52.174	70.343	34.121	2.268	1.779
8	201.885	194.691	178.347	162.993	51.539	69.213	34.864	2.286	3.161
9	251.334	196.274	182.281	159.515	52.100	71.645	36.808	2.549	5.732

10	302.366	195.298	183.444	164.176	51.746	70.819	37.398	2.603	9.701
11	122.288	194.263	169.615	204.072	50.290	63.774	32.690	2.038	1.031
12	160.641	195.367	175.251	204.152	50.494	64.858	35.181	2.255	1.857
13	203.486	195.706	178.773	205.309	50.720	65.538	36.686	2.435	3.207
14	251.260	195.953	181.727	204.648	50.669	66.067	37.862	2.567	5.751
15	301.278	195.350	183.358	205.857	50.671	66.024	38.111	2.571	9.638

Table-12B: Induced performance results from measured readings of Table-12A

#	Hot water heat input (kW)	Cold water heat rejected (kW)	Cold water pump Power (kW) *	GM Overall Net Power (GM, HW, CW pump) (kW)	GM Gross Power (NO pumps) (kW)	Overall efficiency (GM, HW, CW pump) (%)	GM net efficiency (GM pump) (%)	GM gross efficiency (NO pumps) (%)
1	423.554	384.671	1.008	27.904	31.844	6.588	7.063	7.518
2	455.401	410.517	1.008	29.320	34.320	6.438	7.068	7.536
3	479.461	418.479	1.008	29.296	35.706	6.110	6.974	7.447
4	499.931	435.146	1.008	28.069	37.252	5.615	6.967	7.451
5	509.990	440.787	1.008	24.850	37.920	4.873	6.948	7.435
6	427.198	391.780	1.799	28.355	33.072	6.638	7.295	7.741
7	470.236	420.927	1.799	30.542	36.389	6.495	7.256	7.738
8	483.434	422.058	1.799	29.903	37.150	6.186	7.212	7.685
9	515.273	456.774	1.799	29.277	39.357	5.682	7.143	7.638
10	525.140	458.780	1.799	25.898	40.000	4.932	7.121	7.617
11	441.596	403.146	3.160	28.499	34.728	6.454	7.403	7.864
12	473.435	429.607	3.160	30.165	37.436	6.371	7.431	7.907
13	504.834	445.724	3.160	30.320	39.121	6.006	7.267	7.749
14	523.669	461.660	3.160	28.952	40.430	5.529	7.230	7.720
15	529.350	463.038	3.160	25.313	40.682	4.782	7.200	7.685

Table 13A: Performance results for HW Temp= 215°F; HW flow rate = 120gpm to 300gpm; CW Temp 50°F and CW flow rate = 120gpm, 160gpm, and 200gpm

#	Hot water flow rate - Average (GPM)	Hot water Supply Temp - Average (F)	Hot water return Temp -Average (F)	Cold water flow rate - Average (GPM)	Cold water supply Temp - Average (F)	Cold water return Temp - Average (F)	GM Net Power - Average (kW)	GM Pump Power - Average (kW)	Hot water pump power - Average (kW)
1	123.665	214.570	183.877	118.526	52.352	80.624	40.143	3.038	1.031
2	161.509	214.842	189.999	117.844	52.436	82.124	42.790	3.286	1.819
3	201.266	215.142	194.222	119.142	52.487	83.524	44.793	3.557	3.118
4	253.109	215.287	197.939	118.778	52.356	84.423	46.533	3.784	5.822
5	300.953	215.453	200.549	118.632	52.445	84.985	47.632	3.862	9.529
6	123.395	214.637	183.943	163.370	51.726	72.383	41.551	2.910	1.028

7	162.437	214.151	188.938	164.837	51.600	73.809	44.529	3.326	1.840
8	203.083	214.439	193.404	164.067	52.102	75.151	46.386	3.550	3.206
9	252.859	214.451	197.125	163.707	51.793	75.274	47.727	3.622	5.794
10	302.477	214.441	199.845	163.547	51.594	74.994	47.683	3.632	9.657
11	123.654	213.580	182.064	204.417	49.885	66.604	42.927	3.036	1.030
12	162.084	214.915	189.296	204.023	50.559	68.561	45.722	3.325	1.815
13	202.295	214.754	193.411	205.245	50.546	69.239	47.778	3.498	3.167
14	252.642	214.618	197.376	205.019	50.583	69.247	47.742	3.510	5.763
15	301.264	214.516	199.962	204.920	50.603	69.206	47.682	3.515	9.542

Table-13B: Induced performance results from measured readings of Table-13A

#	Hot water heat input (kW)	Cold water heat rejected (kW)	Cold water pump Power (kW) *	GM Overall Net Power (GM, HW, CW pump) (kW)	GM Gross Power (NO pumps) (kW)	Overall efficiency (GM, HW, CW pump) (%)	GM net efficiency (GM pump) (%)	GM gross efficiency (NO pumps) (%)
1	556.096	490.959	1.008	38.103	43.181	6.852	7.219	7.765
2	587.854	512.580	1.008	39.962	46.075	6.798	7.279	7.838
3	616.873	541.765	1.008	40.666	48.350	6.592	7.261	7.838
4	643.317	558.025	1.008	39.703	50.317	6.172	7.233	7.821
5	657.154	565.568	1.008	37.095	51.493	5.645	7.248	7.836
6	554.907	494.425	1.799	38.724	44.461	6.978	7.488	8.012
7	600.039	536.355	1.799	40.890	47.855	6.815	7.421	7.975
8	625.887	554.050	1.799	41.380	49.936	6.611	7.411	7.978
9	641.869	563.160	1.799	40.134	51.349	6.253	7.436	8.000
10	646.805	560.708	1.799	36.227	51.315	5.601	7.372	7.934
11	570.961	500.715	3.160	38.738	45.963	6.785	7.518	8.050
12	608.372	538.102	3.160	40.748	49.047	6.698	7.515	8.062
13	632.583	562.107	3.160	41.452	51.276	6.553	7.553	8.106
14	638.184	560.602	3.160	38.820	51.252	6.083	7.481	8.031
15	642.372	558.517	3.160	34.980	51.197	5.445	7.423	7.970

Table 14A: Performance results for HW Temp= 225°F; HW flow rate = 120gpm to 300gpm; CW Temp 50°F and CW flow rate = 120gpm, 160gpm, and 200gpm

#	Hot water flow rate - Average (GPM)	Hot water Supply Temp - Average (F)	Hot water return Temp -Average (F)	Cold water flow rate - Average (GPM)	Cold water supply Temp - Average (F)	Cold water return Temp - Average (F)	GM Net Power - Average (kW)	GM Pump Power - Average (kW)	Hot water pump power - Average (kW)
1	123.565	224.699	190.110	122.553	51.017	82.285	46.105	3.732	1.029
2	162.066	226.393	199.561	121.926	51.760	83.313	47.631	3.626	1.843
3	203.472	226.197	204.571	121.271	52.072	83.429	47.588	3.687	3.160

4	252.042	226.437	208.850	122.099	51.986	83.521	47.598	3.692	5.754
5	301.069	226.640	211.812	121.301	51.997	83.540	47.604	3.764	9.561
6	121.267	223.576	188.466	157.777	51.955	76.046	46.784	3.676	1.000
7	159.804	224.123	197.259	156.921	52.218	76.329	47.672	3.485	1.775
8	202.310	226.399	204.889	163.575	51.910	75.223	47.685	3.544	3.130
9	251.688	224.794	207.379	156.836	52.133	76.396	47.617	3.562	5.693
10	302.236	226.628	211.976	164.276	51.884	75.090	47.645	3.596	9.516
11	123.015	221.247	186.870	204.384	50.739	69.147	46.658	3.509	1.017
12	161.315	220.876	194.331	204.727	50.775	69.330	47.758	3.429	1.794
13	202.499	221.339	200.029	204.251	50.740	69.314	47.757	3.474	3.155
14	251.924	221.495	204.189	204.018	50.767	69.312	47.705	3.490	5.723
15	301.888	222.119	207.515	204.446	50.503	69.113	47.725	3.510	9.478

Table-14B: Induced performance results from measured readings of Table-14A

#	Hot water heat input (kW)	Cold water heat rejected (kW)	Cold water pump Power (kW) *	GM Overall Net Power (GM, HW, CW pump) (kW)	GM Gross Power (NO pumps) (kW)	Overall efficiency (GM, HW, CW pump) (%)	GM net efficiency (GM pump) (%)	GM gross efficiency (NO pumps) (%)
1	626.185	561.405	1.008	44.068	49.837	7.038	7.363	7.959
2	637.098	563.654	1.008	44.780	51.258	7.029	7.476	8.046
3	644.707	557.128	1.008	43.420	51.275	6.735	7.381	7.953
4	649.425	564.126	1.008	40.835	51.290	6.288	7.329	7.898
5	654.074	560.586	1.008	37.035	51.368	5.662	7.278	7.854
6	623.788	556.890	1.799	43.985	50.461	7.051	7.500	8.089
7	628.977	554.315	1.799	44.097	51.157	7.011	7.579	8.133
8	637.574	558.694	1.799	42.756	51.229	6.706	7.479	8.035
9	642.195	557.520	1.799	40.125	51.179	6.248	7.415	7.969
10	648.795	558.531	1.799	36.329	51.241	5.600	7.344	7.898
11	619.572	551.202	3.160	42.482	50.167	6.857	7.531	8.097
12	627.353	556.560	3.160	42.804	51.187	6.823	7.613	8.159
13	632.231	555.833	3.160	41.443	51.231	6.555	7.554	8.103
14	638.775	554.302	3.160	38.823	51.195	6.078	7.468	8.015
15	645.915	557.424	3.160	35.088	51.235	5.432	7.389	7.932

Chapter 10
Data Analysis

This chapter discusses the analysis results derived from experimental data and other information observed during the reliability and performance tests:

ANLYSIS RESULTS FROM RELIABILITY TEST:
Based on observation obtained from the reliability test, some inferences in reliability, feasibility, and maintenance and operation requirements are given below:

1. The system is reliable (Under normal operation condition, no foreseen technical problems are expected for long term operation).
2. To operate the system doesn't need advance technology background. The only thing that is special and needs to be remembered for operating the GM is the need of using medium temperature heating source to start the GM. It is recommended designing the heating and cooling loops with the capability of heating and cooling process control for startup. This will make the startup just as simple as pushing the Start button. The design of heating and cooling process control is a standard practice.
3. The maintenance requirements of the system don't need advanced technology background and the GM maintenance process can be combined with the routine diesel generator maintenance schedule. Since virtually no defects were found during the reliability test, the maintenance requirements were expected as simple as to follow the items listed in the Maintenance Items list recommended by the manufacturer. The routine maintenance schedule includes quarterly, semi-annual, annual, and bi-annual maintenance items. Quarterly maintenance items are basically inspections (i.e. visually inspect plumbing for leakage oil/water and brackets, connections, bolts, conduit, wire, etc.). Semi-annual (e.g., replace filter/dryer element), annual (e.g., check heat exchanger pressure- drop in hot and cold water supply), and bi-annual (e.g. replace pressure relief valves) maintenance items could be combined with the routine maintenance schedule of the diesel generator and carried out by the diesel power plant engineers.
4. The technology is feasible for rural Alaska villages (i.e. the easiness in installation, operation, and maintenance requirements).

The performance data obtained from the reliability test shows that the ORC system can consistently generate a gross power of 50.1kW (i.e. rated power of the GM) and a net output power of 46.4kW (i.e. difference between the Gross power and the required working fluid pump power of the ORC system) under the condition of sufficient heating source and sufficient cooling source. Based on this result, the potential of the GM in emission reduction, CO_2 reduction, fuel savings, and payback period are evaluated and listed below in Table 15:

Table 15 A Diesel Genset Performance Data; Estimated Reductions in Emissions, CO2 Production, and Fuel Consumption; and Estimated Payback Period Through the Application of the 50kW ORC System (GM).

	A Village Diesel Genset Data	**ORC (GM) Data 49.4 kW expander output, 46.4kW ORC net output**
	With enough heating and cooling sources	Annual Net Power Generation: 395,328 kW-hr (355 working days and 10 maintenance days)
Emissions	**lb/kW-hr**	**Reductions (lb) per Year**
NOx	0.015211	6013
PM-10	0.000374	148
CO	0.00044	174
HC	0.000661	261
Diesel Annual Saving		28,238 gallons (Based on 14kWh/gal)
Annual CO2 reduction (ton)		316 tons (based on 22.4lb CO_2/gallon of diesel)
Payback Period	Assumption: $190,000 for total cost for GM installation (110,000 GM cost and 80,000 installation cost)	3.6 years (with 0 interest rate) 4.6 years (with 10% interest rate)

ANLYSIS RESULTS FROM PERFORMANCE TEST:

Based on the data given in Table-10A to Table-14B in the previous chapter Figures 12A to 14B are plotted. Figure-12A to Figure-14B provide the analysis results from performance test of GM for various hot water flow rates and temperatures, cold water flow rates. The figures captioned with “A” gives the screw expander and GM net power output for various hot water flow rates and temperatures at a cold water flow rate. GM net power is obtained by subtracting working fluid pump power from screw expander power output. The figures captioned with “B” gives the screw expander power output efficiency and GM net power output efficiency for various hot water flow rates and temperatures at a cold water flow rate.

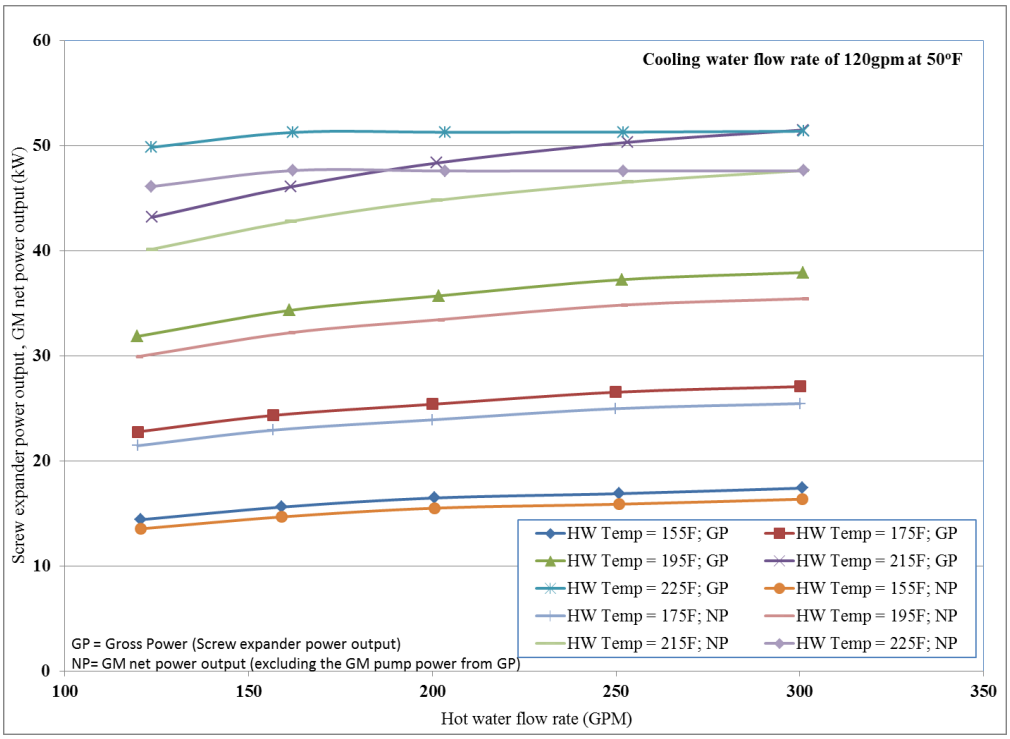


Figure-12A: Screw expander and GM net power output for different hot water flow rates and temperatures at cold water condition of 120gpm @ 50°F

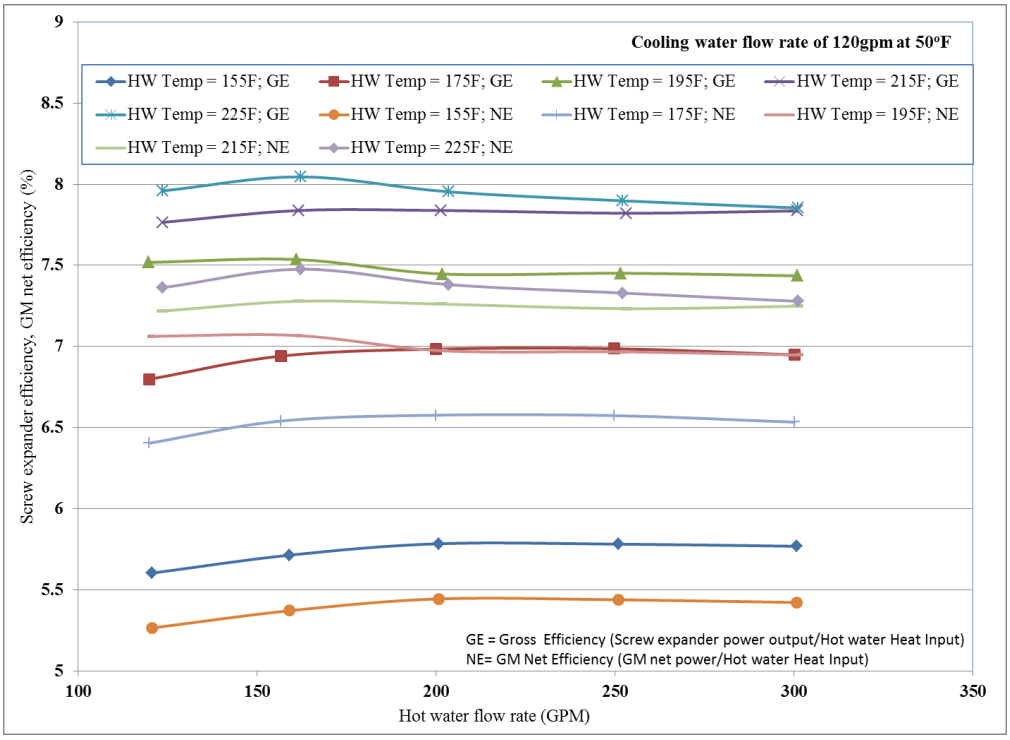


Figure-12B: Screw expander efficiency and GM net efficiency for different hot water flow rates and temperatures at cold water condition of 120gpm @ 50°F

Curves in Figure-12A and Figure-12B can be categorized into two groups. The first group includes cases which provide more than enough heat energy for the GM to generate maximum allowed power output of 50kW. For these cases, the excess heat energy is dissipated into cooling water which causes the net GM efficiency to drop (even if the net power output increases) due to the increased amount of heat lost into the cooling water. The second group contains the cases of which the maximum gross outputs are less than 50kW. The gross output of these cases are not proportional to the input heat energy, due to the limited (designed) heat transfer area of the heat exchanger, of which the effectiveness reduces while flow rate increases over certain flow rates. For these cases the gross output power are increasing at slower rate for higher flow rate but the net efficiency increases with a slower rate or even may decrease with higher flow rate. Similar trends are seen with other cold water flow rate plots and the same discussion is valid for Figures 13A & 13B as well as Figure 14A & 14B.

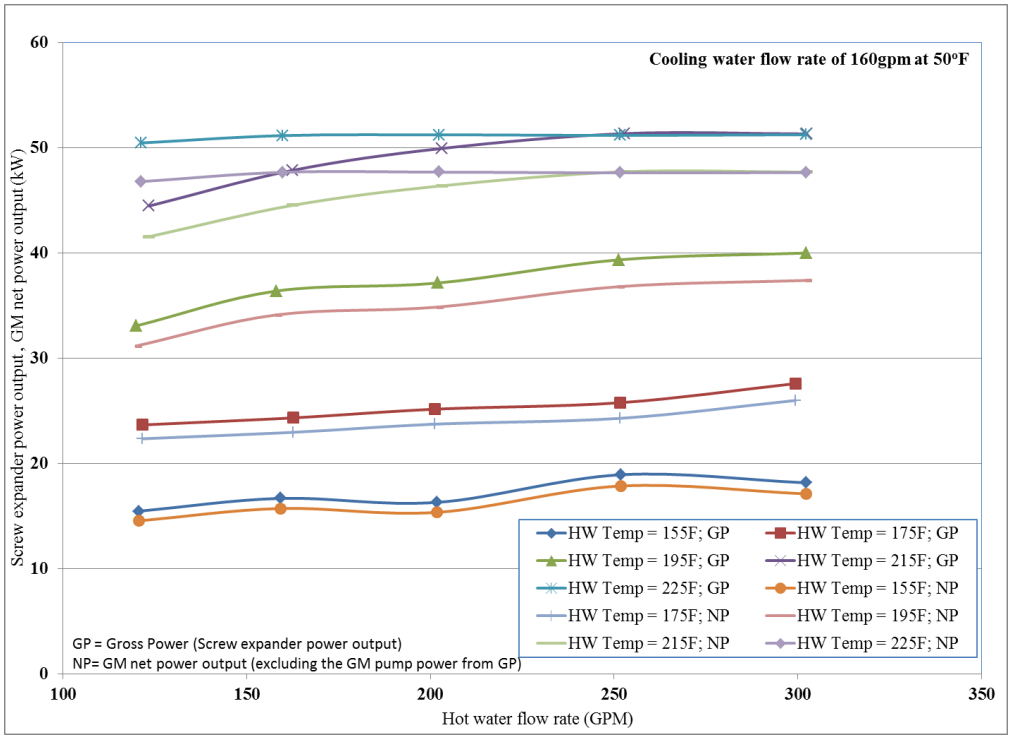


Figure-13A: Screw expander and GM net power output for different hot water flow rates and temperatures at cold water condition of 160gpm @ 50°F

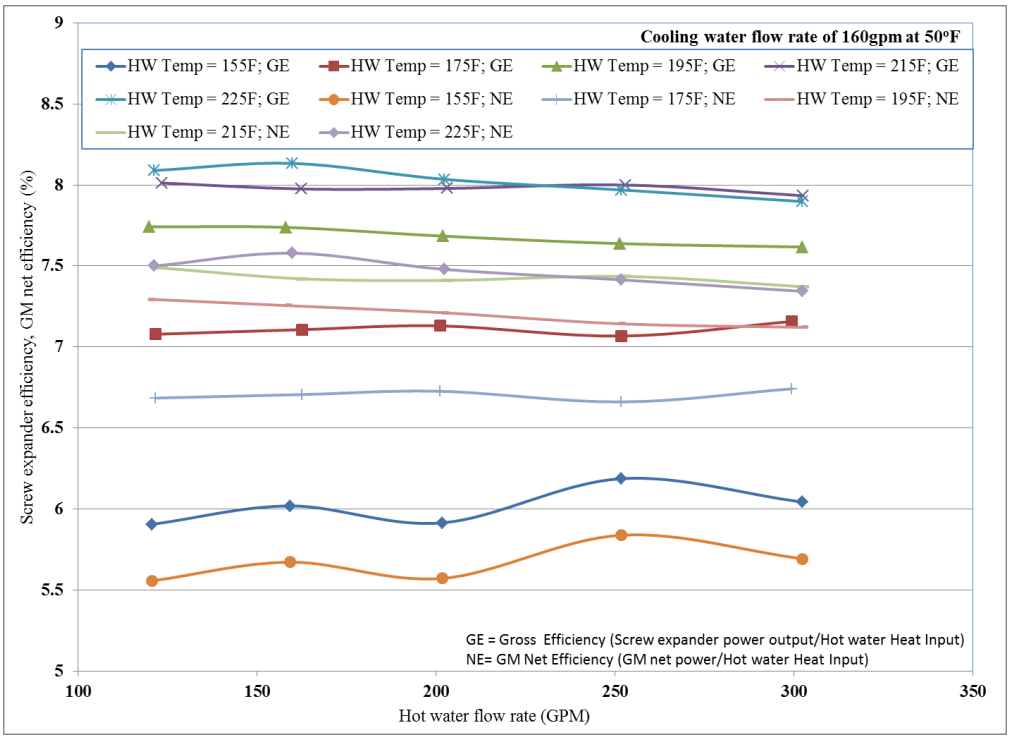


Figure-13B: Screw expander efficiency and GM net efficiency for different hot water flow rates and temperatures at cold water condition of 160gpm @ 50°F

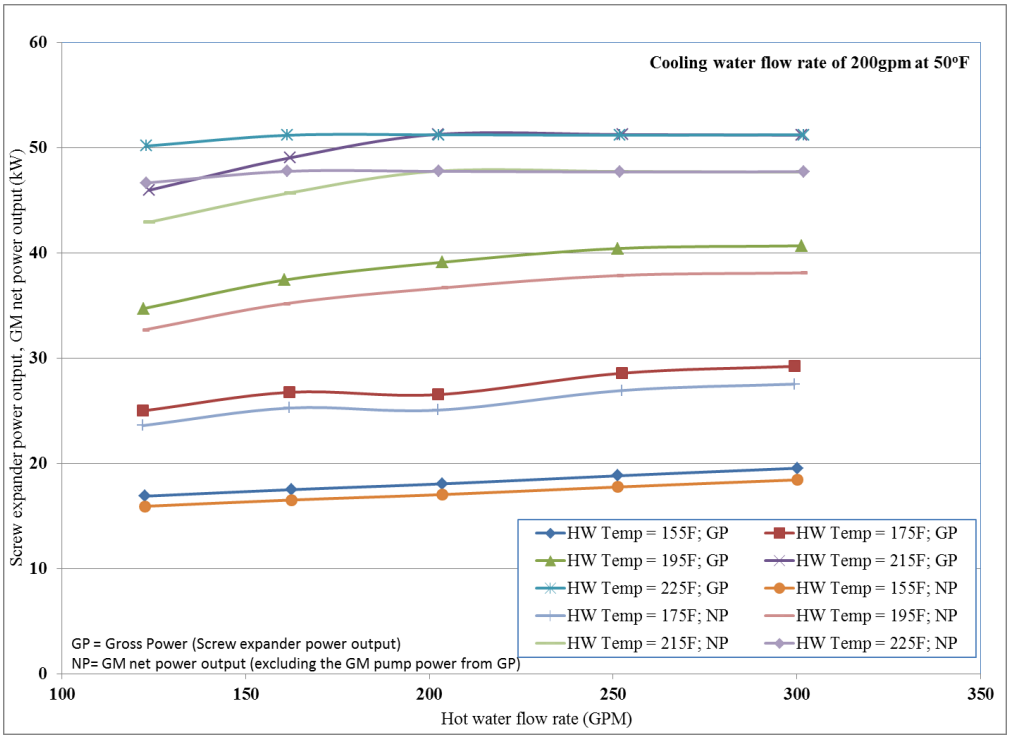


Figure-14A: Screw expander and GM net power output for different hot water flow rates and temperatures at cold water condition of 200gpm @ 50°F

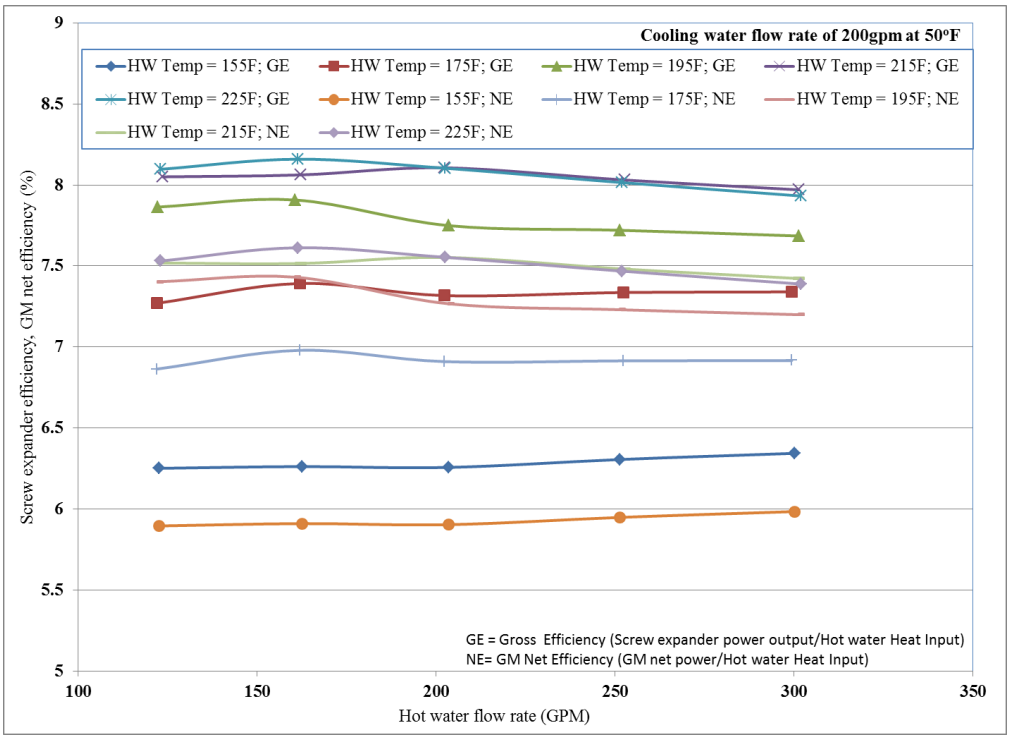


Figure-14B: Screw expander efficiency and GM net efficiency for different hot water flow rates and temperatures at cold water condition of 200gpm @ 50°F

Chapter 11

Discussions

This chapter summarizes some of the findings, which includes general information in design of the project and testing system, installation, operation and maintenance, performance characteristics and possible applications of the characteristics, etc.

FINDINGS OF GENERATION INFORMATION:
A thorough survey before completing the project proposal helps defining achievable objectives and objectives, which may lead to new findings or applicable information.

For product selection, comprehensive understandings of operation principles of existing and proposed low temperature heat engines and the properties and limitations of the major components used (or to be used) for the respective systems help making selections, which are more likely to generate successful and useful project results. Based on a second survey, among all the potential low temperature/small capacity heat engine ideas listed in the first survey report, only the ORC system with newly developed screw expander is in the stage of mixed testing and commercialization.

A thorough understanding of the selected heat engine, the ORC system (or GM), helps determining the system parameters need to be tested to make the results useful for selecting individual appropriate village diesel generators to match this particular heat engine system and to optimize the benefit obtainable from applying heat recovery. For example, the GM has a maximum power output of 50kW so it has a limitation on the conversion of heat into power. Excessive heat may become waste. Design of the heating and cooling system needs to consider this limitation of the GM and existing village waste heat application.

The preliminary modeling and simulation (model with some approximations about the properties of the system components) may help determining critical parameters of the testing system (i.e. heating loop, cooling loop, control elements, etc.) and estimating capacities of components for testing system design.

To selected testing site, the available space and existences of utilities/heating/cooling sources are important for testing system design and installation. Equally important information is if there are local codes need to be followed.

Based on the data gathered during the procurement, for efficient operation of the ORC system, the cost of the cooling system and its operation cost may become significant depending on the availability of the type of cooling source.

For an isolated small city, a careful purchase plan is needed in order to avoid large schedule delay. For example, sometimes, even common structure materials may take a several days to become available.

For this project, the installation and instrumentation (steam loop, hot water loop, cold water loop, electrical circuit, signal/monitoring/control circuits, data acquisition system) process is very smooth.

FINDINGS OF THE GM INFORMATION

During the commissioning, a couple of design issues have been discussed:

1. The selected low temperature ORC heat engine (or GM) may be easily converted from a 50kW capacity into a 65kW capacity.
2. A system computer program imperfection to cause emergency shutdown of the ORC system resulted from starting the system under undesired heating flow condition (i.e. high temperature) is not expected. It may not cause any damage to the system but is undesirable. This can be easily overcome by following correct starting procedure or adding control capability into the installed heating loop.

Some of the System Limitations:

1. GM has a maximum power output of 50kW so it has a limitation on the conversion of heat into power. Excessive heat may become waste.
2. GM may shut down if the voltage and frequency fluctuations are high for the location where it is installed (Voltage fluctuations greater than ±5V and frequency fluctuations greater than 10%).
3. GM starts only when the temperature difference between hot and cold side is greater than preset value. The pre set value can be changed between 50^oF to 80^oF.
4. GM initial (i.e. when we start the machine) power output is set by a predefined equation which is a function of temperature difference between hot and cold side (ΔT_{H-C}). If the initial temperature difference (ΔT_{H-C}) is greater than 170^oF then the starting power output of GM is greater than 50kW and the machine over speeds and shutdowns. So the initial ΔT_{H-C} should not be too high for the machine to start smoothly.
5. The GM has a hot water supply temperature limitation of 245^oF. If the temperature is above this value, GM shutdowns.
6. GM has a minimum net power output limit of 5kW.
7. The components of this ORC system are capable to stand for 65kW output. The system can be modified to increase its power capacity from 50kW to 65kW without too much of work.

The system efficiency varies not much across a wide range of input heat. The performance is better than expected and may make the system applicable to a wide range of size of engines. (Examples: At 627kW and 225^oF heat source, the engine operates with rated outputs of 50kW and the system efficiency is 8.2%. At 257kW heat source and 155^oF, system efficiency is 5.6%.)

At a given heat source temperature, the system efficiency varies not much against flow rate. This performance is also better than expected. This may make the system more suitable for engines at different loads. (Example: With the heating source at 86^oC, the efficiencies of the system are around 7.52%, 7.53%, and 7.43% for the respective flow rates of about 120gpm, 200gpm, and 300gpm.). The reduced efficiency at 300gpm is resulted from the increased parasitic power of the working fluid pump.

As observed from test data, GM may have heat input limit at each of the hot water supply temperature, which is one of the main purposes of conducting performance test and creating performance characteristic curves of the heat engine, which will be useful for determining appropriate waste heat distribution between heating and power for a given Alaskan village diesel power and selecting appropriate villages for application.

Based on reliability test result, for full rated output (gross output of 49.4kW and net output of 46.4kW), the estimated payback period is 3.6 years for a 0% interest rate and 4.6 years for a 10% interest rate, assuming that the total installation cost is $190,000. The reductions in emissions and GHG can be found in Table 15.

The experimental results (Figures 11A and 11B) show that the GM applies a limit to its maximum power output (performance curves for heating water temperatures of 215F and 225F. The curves also show that an excessive amount of hot water flow rate will not increase the net power output, but decrease the net efficiency. For lower hot water temperatures, the experimental results show that there are optimal flow rates for both net power output and net efficiency. This explanation to the optimal values can be found in the previous chapter. An example was given to show how the charts obtained from the experimental results may be used to distribute the waste heat between power application and heating application.

Chapter 12

Conclusions

This chapter discusses the accomplished tasks related to the project objectives and the future work needs to be done to the GM. It will also listed out the comments about the GM based on the experience obtained from this testing project.

The objectives of this project as mentioned in the Introduction chapter include:

1. Objective 1: To prove that an improvement of the efficiency of the diesel power plant by about 10% (i.e. about 4% of fuel efficiency) is achievable through the use of an organic Rankine cycle (ORC) system, which uses waste heat contained in diesel engine jacket water and exhaust.

   Based on experimental results, the maximum net efficiency experienced at the heating fluid temperature of 195°F (jacket water temperature) is about 7.4%. For more than 50% of fuel energy as waste heat (in jacket water and exhaust), the potential of fuel efficiency improvement is about 3.7%, which is close to the target of 4% fuel efficiency improvement. In addition, if higher temperature heat source (obtainable from exhaust) is used and the cap of the GM output restriction is moved from 50kW to 65kW (by a simple retrofitting process) to improve the maximum efficiency for high temperature fluid (245°F), the 4% improvement in fuel efficiency seems reachable.

2. Objective 2: To evaluate feasibility, operation and maintenance requirements, and payback time of applying a selected ORC system.

   Based on the observation and operation experience, the system is considered very reliable (Under normal operation condition, no foreseen technical problems are expected for long term operation) and no advanced technology background is needed for operation. Considering maintenance requirements, no advanced technology background are needed for maintenance and not much of extra cost is needed beside the consuming materials (filters, lube oil, etc.), if the GM routine maintenance schedule is incorporated into the routine diesel genset maintenance schedule. (Please refer to Chapter 10 Data Analysis for details.)

   Based on the reliability testing results (at full capacity of the GM), the estimated payback time is 3.6 years for a 0% interest rate and 4.6 years for a 10% interest rate. (Please refer to Chapter 10 Data Analysis for details.)

3. Objective 3: 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 emissions and greenhouse gas reductions.

   Guidelines for ORC System Selection:

Due to the lack of performance data of the 250kW ORC system, the available performance information of ORC system is only for the 50kW system. It may not be appropriate to use this data to develop guidelines for large diesel gensets used in rural Alaska. Guidelines for ORC system selection will be provided after the performance data of the 250kW ORC system become available.

Operations:
Most of the operations procedures for GM are described in the manufacturer's manual and the manual is up-to-date. No additional suggestions required for operation. Some of the most important one are given below

a. During working on GM, if the hot water loop does not have a bypass, then do not start the hot water or cold water supply to GM (until ready to start the GM) as it may pressurize the working fluid and rotate the expander and generator.
b. During regular check-list, check the expander high pressure value and hot water inlet temperature. The hot water inlet temperature greater than 245°F may result in over pressurizing the system which in-turn may lift the working fluid pressure relief valves causing lose in working fluid and may cause damage to components.
c. In case if a need arrives for emergency shutdown of GM, adjacent to the HMI screen there is "Emergency Stop Button" on the front panel of GM. This will shutdown the GM immediately.

Maintenance:
Similar to operations most of the maintenance procedures for GM are described in the manufacturer's manual and the manual is up-to-date. No additional suggestions required for maintenance. Some of the important ones are listed below.

a. Check for non-condensable gases in the system (procedure given in manual). Purge the non-condensable gases from the system following the procedure given in GM manual.
b. Visually inspect all the joints and connections for oil/water leak.
c. Visually check the electrical wiring frequently for any damaged connections due to excess heat or loose connections (by tug test).

Evaluation of Potential Impact on Rural Alaska Economy, Fuel Consumption, and Emissions and Greenhouse Gas Reductions:
To evaluate all the effects listed above for rural Alaska, the number of villages, which have the best match to the GM, is needed. Due to the lack of performance data of the 250kW ORC system, it is difficult to make decision about how many villages have the better match with the 50kW ORC system or the 250kW system. This information will be provided after the performance data of the 250kW ORC system become available.

4. The fourth is the performance and economic comparison of two ORC systems. One ORC system is a 50kW system, which uses screw expander, comes under emerging technology. The second ORC system is a Pratt & Whitney (P&W) 250 kW unit which uses radial turbine belongs to the category of well-developed technology.

   Performance and economic benefit of the 50kW ORC system have been evaluated in the Chapter of Data Analysis. However, due to an unexpected turbo charger break down of the Cordova diesel generator set, to which the 250kW ORC system will be installed, performance information of the 250kW ORC becomes unavailable at this moment. Therefore, the result of the fourth task, comparison in performance of the two ORC systems, is not included in this report and will be provided in a make-up report once the performance data of the 250kW ORC system becomes available.

NEAR FUTURE WORK:
Following the completion of the Phase 1 laboratory test, it is the intent of ACEP to transport and install the GM in a power plant in the Tanana Chiefs Conference (TCC) region in order to conduct a field test under real-world conditions. ACEP maintains a partnership with TCC, a non-profit consortium of forty-two Alaska Native tribal communities in the interior region of the state, for rural energy research and development. The project location has been tentatively determined, but discussions with Alaska Power & Telephone regarding the power plant in Tok are still ongoing. It is anticipated the Phase 2 field test will provide actual (not projected) emissions reductions results. The field test is funded with support from the AEA Renewable Energy Fund.

The expected date for the ORC system relocation is tentatively determined on April 30st, 2012. Major activities for relocation and preparation of field testing include:

1. Selection of the village for relocation: Tok is the tentative candidate due to the size and load of the power plant and easiness in transportation.

2. Procurement and layouts of required heating and cooling piping systems, electrical circuit for uploading, and instrumentation and monitoring system: The emphases are on optimizing the benefit to the village and avoiding negative effect of the added system on the performance of the diesel power plant, such as emissions and combustion efficiency, stability and effectiveness of the local electrical grid, etc.

3. Preparation of the ORC system for relocation and starting: This may include discharge of refrigerant from the ORC system, packaging and shipping, recharge and starting of the ORC system, etc.

COMMENTS:
1. Modify the GM to increase its power capacity from 50kW to 65kW:
   Reason: This may increase the output of the GM and also the maximum net efficiency of the system (projected maximum efficiency is about 8.5%). All the components of this ORC system are designed and selected for 65kW operation. The modification is only a minor operation.

2. Design a control algorithm to avoid automatic emergency shutdown of the ORC system resulted from starting the system under undesired heating flow condition (i.e. temperature). This is also a minor operation.

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Appendix IA
Survey of Low Temperature Heat Engine Companies (2008)

Introduction

In today's energy crisis people are looking at almost any means in hopes to lower their energy bill. Some of these technologies are already in the market as cost effective options and others still need more development before they make economic sense. Waste heat recovery at an industrial level is a proven technology with many instances of it being successfully used to power plants operations more efficient. Recovering the waste heat from diesel engines, on the other hand, is not a mature field. There are companies out there looking to address this problem but they are not quite there when it comes to a cost effective solution. The technology is there but the main problem is in developing a product that produces a positive net energy. Because the waste heat is a low energy source it is easy for parasitic power from the recovery unit to be higher than what is being produced. For purposes of this report three main technologies will be examined, viz the Organic Rankine Cycle, the Kalina Cycle and the Stirling Engine. The basics of these technologies will be examined in brief as well as a more detailed look at some of the companies involved in each and what the status of their product. A survey summary is attached to the end of this report.

Organic Rankine Cycle

Diesel engines lose about 30-40% of the input energy in the exhaust. Organic Rankine Cycles (ORC) look to take this low temperature heat source and improve the overall efficiency of the engine or produce electricity directly. Some estimate the efficiency of an ORC unit to be about 10-20%, which would improve the overall engine efficiency by about 3-5%.

An ORC unit works very much like that of a traditional steam Rankine Cycle: a working fluid is pumped through a heat exchanger where it is vaporized and passed through a turbine and then re-condensed, and then the process is repeated. The big difference is in that of the working fluid. The traditional Rankine Cycle uses H_2O as its working fluid with has a high boiling point. Since the waste heat from the engine is a low temperature source H_2O will not work. So an ORC unit uses some organic fluid as its working fluid. Choice of the working fluid will depend on the heat source and some companies have their own special blend that allows them to capture heat at low temperatures.

ORC units have been installed in various places. Ormat, a leader in the ORC technology, has units in North America, Europe and Asia[1]. These units, however, are being installed on an industrial scale, ranging from 200 kW to 22 MW. For use on a diesel generator one would be looking for a unit less than 100 kW for the most part. Not many units have been built at this size, although there are companies with them in the works. In order to get a better grasp of the state of this technology it is will be helpful to look at these companies individually. Editorial comments will be left out as much as possible with the hope that the facts gleaned from the companies websites and through personal contact will be enough to show how close we are to seeing an ORC unit installed with a diesel generator as a way to capture waste heat.

ORC Companies

After a thorough web search, seven companies were found that use ORC technology in their products. Some of these companies specifically targeted the waste heat from diesel engines while others were broader in their application. The companies looked at were Global Energy, Barber-Nichols, ElectraTherm, TransPacific Energy, Deluge Inc., Ormat, Turboden, UTC Power and GMK. Each company will be looked at in turn.

Global Energy

Global Energy, a Madison, WI company under the leadership of Greg Giese, has developed the Infinity Turbine ®. This is an ORC turbine built for waste heat and geothermal applications. While there are potentially numerous uses for this turbine one that is specifically being targeted is the diesel engine exhaust. According to the website, the Infinity Turbine consists of a single skid-mounted assemble that fits in the standard 20 or 40 foot ISO standard shipping container. All the equipment required for the power

skid to be operated (i.e. heat exchangers, piping, working fluid feed pump, turbine, electric generator, control and switch-gear) fit into the container. In early mid-July 2008 a 30 kW unit and an 80 kW unit were being built in Toronto. By July 29, 2008 the 30kw unit had been sold to a geothermal application in Casper, WY. It is hoped that in a month or so data will be ready for analysis. While the website does list some performance specifications this are only theoretical calculations. Hopefully this test site in WY will show that the theoretical calculations where right. A price of $60,000 was quoted over the phone for the 30 kW unit although it is not know how much it sold for to the geothermal plant in WY. Also a delivery time of 11 weeks was quoted, presumably from day of purchase.

Barber-Nichols

Barber-Nichols is a leader in the field of turbomachinery. Thermodynamic Cycle Systems including the Organic Rankine Cycle and the Steam Rankine Cycle are part of their core competency. They have already built waste heat applications but on an industrial scale. These units are much too large to be used with diesel engines but they demonstrate that Barber-Nichols is competent in the technology. They are currently working with some Canadian companies, however, to develop smaller units that could be used with diesel engines. Since these are custom designed units based on the size of the generators an interested buyer would need to send in specifications regarding their application. During a phone call with a company representative a price range from few hundred-thousand dollars up to $1M was put forward as an estimate of the total cost, including initial research, NRE, and unit itself.

ElectraTherm

Nevada based ElectraTherm has recently come out with their own ORC unit that captures waste on a smaller scale. They have plans for units ranging from 30-500 kW. It is not clear how many units they have indeed sold and that have been field tested, but one unit at SMU in Texas is known to be running and undergoing field tests, and it is most likely that this is their only one. This unit was demoed at the Geothermal Conference at SMU in June of 2008. Dr. Dennis Witmer from the Alaska Center for Energy and Power at the University of Alaska, Fairbanks traveled there to get a first hand view of the unit. Overall, the demonstration was not very impressive according to Dr. Witmer. The unit was very loud and there seemed to be some grinding sound, as well some of the parts looked like they were used parts. More importantly, though, it is unclear whether the unit is making any net power when parasitic power is taken into account. On a follow up phone call with Michael Paul from SMU in July 2008 he mentioned that an availability test had not been done since it was not a waste heat application. So it is unclear how long these units can be up and running before maintenance needs to be done. Currently, SMU is not planning on buying the unit after what they saw in the tests they did. The latest price that was available was $2400-$2700 per kW.

TransPacific Energy

This Nevada bases company is developing a heat recovery/energy conversion system using ORC technology. They claim on their website that their system can use heat sources at temperatures as low as 80°F and up to 900°F. This is a larger range than most

other ORC units which usually limited to a low temperature of around 200°F. They can do this, they say, because of their own specially designed refrigerant. They believe that this refrigerant is their key advantage over competitors. In an email response from Jim Olsen, a company representative, it was revealed that they do not actually have any units installed yet as they are a new company. This most likely means that any specifications they are stating are simply theoretical calculations and not actual measured results. The company website says this about the sizing of their units:

> Systems are sized from 20 kW up to 20 MW+ modules, containing all the equipment required for the units to be operated (i.e. heat exchangers, piping, working fluid feed pump, turbine, electric generator, controls and switch-gear). Larger units are composed of multiple modules, pre-assembled at the factory.

The price for a 115 kW unit was quoted as running up to $250,000 for the complete unit. Delivery time is expected to be 6-9 months depending on their fabricator, Concepts NREC, out of Massachusetts.

Deluge

The Deluge Natural Energy Engine is not an ORC unit but rather a thermo hydraulic engine that produces mechanical energy by heating a fluid so that it expands and moves a piston. The heat source can be solar, geothermal, or waste heat. The main components of the engine are the piston/cylinder and the heat transfer system. The cylinder contains the piston and the working fluid, usually $C0_2$. The heat transfer system is made up of heat exchangers and a system to circulate the heat exchange fluid, usually H_2O. A three step process creates a back and forth movement of the piston, in turn generating mechanical energy. The technology has been independently verified by university and government studies. In an email from July 2008 they reported they finished building their first 250 kW unit and are shipping it to a jobsite in Hawaii for installation. They claim that 250 kW can be produced from a 150 gpm flow of water at 190°F and that temperature drop through their engine would eliminate the need for the radiator cooling. These flow rate and efficiency numbers are based on calibrated tests in the shop. The price for a 250 kW engine/generator set is $400,000 when buying one at a time. The delivery timeframe is 90 days from purchase order.

Ormat

Ormat is the world leader when it comes to ORC technology. They have successfully installed ORC units all around the world. They specialize in Geothermal Power, Recovered Energy Generation, and Remote Power Units. Their units range from 200 kW to 22 MW for the Recovered Energy Generations units that cover waste heat recovery. Their remote power units range in size from 200-4500 watts. When given the specifications for a 125 kW diesel generator the company said that the application was too small for their Recovered Energy Generation units. Currently, Ormat is looking into developing a smaller unit that could be used with diesel generators, but as of now none are available. They will let ACEP know if they decide to pursue a smaller unit.

Turboden

Turboden is an Italian company that specializes in ORC technology. They have Combined Heat and Power systems in set sizes ranging from 200 kW to 2000 kW. They also have Heat Recovery systems that come in set sizes ranging from 500 kW to 1500 kW. They are also able to build custom sizes but currently do not manufacture any under 500 kW for applications requiring a single unit. They have installed many units, mostly in Europe and in the biomass industry. They company did not respond to emails so there is little details as to cost and delivery time.

UTC Power
UTC Power is a division of United Technologies Corporation based in Connecticut. They provide environmentally responsible power solutions and recently have developed an ORC unit. The PureCycle® Power System is an electric power generating system and runs off of any hot water resource at temperatures as low as 195°F. The hot water can come from a geothermal source or some other waste heat source. Currently this ORC unit is sized at 280 kW (gross) of electrical power. One of these is commercially running at Chena Hot Springs Resort in Alaska and they have sold 75 units to date. The average price per kW is $1250 with a delivery time of 8 weeks. They are currently working on a 1 MW unit to be completed in the early part of 2009. There are also plans underway for a possible smaller unit with the size to be determined.

GMK
Gesellschaft für Motoren und Kraftanlagen (GMK) is a leading ORC-module developer and producer in Europe. They have three different products bases on application. Their INDUCAL is an ORC used in power plants to recover waste heat. The electrical output of these units varies from 0.5-5.0 MW. The two other products are GEOCAL and ECOCAL which use geothermal energy and biomass respectively to produce electrical power in similar amounts as that of the INDUCAL. They have various installations of all three products that are currently running in Germany. The company seems to be focusing on larger, industrial power plants rather than on units that would be useful in heat recovery from small diesel generators.

The Kalina Cycle

The Kalina Cycle was developed by the Russian engineer Aleksandr Kalina in the early 1980s. It is a thermodynamic cycle that allows for the converting of thermal energy to mechanical power which can then be used to create electrical power. It is similar to the ORC in how it works with one major difference. The Kalina Cycle uses a binary working fluid. Usually the fluid is a mixture of ammonia and water. This allows for a broader range of boiling points because ammonia has a much lower boiling point than water. Studies have shown that the Kalina Cycle performs better than the Organic Rankine Cycle at moderate pressures. Some believe that it is in general more efficient than the Organic Rankine Cycle, but more testing needs to be done to support this claim.

Commercially tested Kalina Cycle units are not common with only a few ever built. One is used in a geothermal power plant in Iceland. There are two plants in Japan, one a waste heat plant and the other a waste-to-energy demonstration plant where the Kalina Cycle

was used. The first ever demonstration of the technology was in California, which proved to be very successful. Currently the geothermal plant in Iceland and the steel mill waste heat power plant in Japan are still running.

Kalina Cycle Companies

The companies involved in this technology have gone through various mergers and acquisitions so that today there is currently only one company with technological rights to the Kalina Cycle. Global Geothermal was created in 2007 and now owns all those rights as well as the over 200 international patents associated with the proprietary technology. They are a licensing company and sell the rights to the technology to companies wishing to use it in power plants or other applications. The following is a more detailed look at the companies that made up Global Geothermal and some of the current companies who are licensing the technology.

Original Company: Exergy

In 1992, Exergy, the company founded by the inventor of the Kalina Cycle, Aleksandr Kalina, saw the first Kalina Combined Cycle power plant start running at Canoga Park, CA. The 6.5 MW Canoga Park power plant was built as a demonstration plant to show that the technology is commercially viable. The plant ran up until 1997 and was deemed a success, as much useful data was gathered showing that the Kalina Cycle is an efficient way of generating power. Run as both a waste heat power plant and as a combined cycle power plant, it was designed to be tested at extreme conditions. In total it logged about 9,000 hours of operation over its five year life span, which corresponds to about 21% availability. [2]

In 1997 Exergy teamed up with Ebara Corporation, a Japanese company that specialized in advanced industrial systems, to build another demonstration plant in Fukuoka, Japan. [3] This demonstration was a 4.5 MW waste-to-energy plant that ran from 1998-1999 and was again seen as a success.

In 1998 another project in Japan was begun at the Sumitomo Metals Kashima Steelworks in Kashima, Japan as a waste heat recovery application. This was the first commercially installed Kalina Cycle power plant and it currently is still running. The 3.4 MW plant has had great availability and runs off 208°F (98°C) hot water.

Combining of Licensees

It is not very clear how all the mergers and acquisitions worked out and who all the parent companies are that own the companies with the rights to the Kalina Cycle, but an attempt will be made to shed a little light on where this industry has come from and where it is now.

Recurrent Engineering

Recurrent Engineering became a global licensee in 2002. They owned by Wasabi Energy. Not much is available as too what they did with the technology from 2002 to 2007, but in 2007 Global Geothermal was created to buy up Recurrent Engineering and Exergy and to consolidate other companies licensed to use the Kalina Cycle worldwide. Wasabi had

been in some disagreements with their co-venturer, AMP Capital Partners LLC as to how to handle the Kalina Cycle technology and the solution was to incorporate Global Geothermal, which is owned 70% by Wasabi and 30% by AMP

Global Geothermal

Due to these changes in the industry, Global Energy now owns all the rights to the Kalina Cycle as well as the over 200 international patents associated with it. The company's primary business model is to license the technology for up-front fees as well as follow-on fees bases on size and use of the licensee's power plant. They also provide engineering services, equipment procurement, and project management services through their subsidiary Recurrent Engineering

Current Licensed Companies

A number of companies have received licenses from Global Geothermal or the previous companies with licensing power, and are working with Global Geothermal to make the best use of the Kalina Cycle technology.

Exorka

A geothermal energy company in Iceland, Exorka specializes in low temperature geothermal sources. In 1999 they built the first geothermal power plant using the Kalina Cycle in Husavik, northern Iceland. According to their website the plant produces 2MW of power from a flow of 90 kg/s at 248°F (120°C) geothermal brine. Exorka acquired the rights to the Kalina Cycle in 1999, with these rights extending to Iceland and most of Western Europe. They are currently in the late development and finance stages of five projects in Germany. Each project is about a 5 MW binary geothermal power plant. It is expected that these projects will come online in 2010.

Geodymanics

Geodynamics is an Australian geothermal energy company that focuses on hot fractured rock geothermal energy. In 2004 they acquired the rights to use the Kalina Cycle technology in Australia and New Zealand. As of now they do not have a plant running that uses the technology. In 2007 they merged with Exorka to form Exorka International Limited.

Raser Technologies

A publically traded technology licensing company, Raser, was formed in 2003 with a goal to improve the efficiency of rotating electromagnetic and heat transfer applications. They have also got into the geothermal energy industry as well as into waste heat recovery with the goal to make it as efficient a process as they can using their skill and technological knowledge in heat transfer and motors. To this end they become a Kalina Cycle technology licensee in 2006. Raser controls large amounts of geothermal land in Nevada and Utah where they are developing a 20 MW binary geothermal plant. They also are working on a 10 MW geothermal plant in New Mexico. It is hoped that these plants will go online in 2010. Raser not only has Kalina license rights to geothermal applications but also to industrial waste heat applications. They are working closely with

Recurrent Engineering to use Kalina technology in waste heat projects in both the US and Europe, primarily in the cement industry.

Siemens

A well known Germany engineering firm, Siemens purchases Kalina Cycle license rights in 2000 and is able to use them for geothermal power projects in Germany, up to a total of 10 MW in size. Even before it gained the rights to the technology, Siemens has been evaluating and testing it. They have been working with Recurrent Engineering to complete the construction of the first Kalina power plant in Europe. The 3 MW plant was completed in December 2007 as a binary geothermal project built as a turn-key project for the City of Unterhaching, Germany. Siemens will provide long term O&M for the plant. The plant was expected to go through commissioning and acceptance testing during the first quarter of 2008 and come online in mid-June 2008. It is expected to have a lifetime of 30-plus years. Siemens is looking to add more plants in the next decade and currently has two more contracts for turn-key power plants in Germany.

Other Companies Related to Kalina Cycle

There are a few companies that do not fit nicely into the above breakdown but that are either using Kalina Cycle Technology or something very similar. They are discussed below.

Energy Concepts

Energy Concepts is a Maryland based engineering company that focuses on heat-activated absorption systems and the associated fluid contact equipment. The Absorption Cycle was invented by Ferdinand Carre in 1846. The idea is to take heat out of a system by running a cycle that uses a heat input. When ammonia is absorbed in water the vapor pressure decreases, and so according to the laws of thermodynamics, the temperature will drop also, *ceterus paribus.* The absorption cycle has the benefits of requiring little electric input and using natural substances, ammonia and water, instead of halocarbons. Although this in not a true Kalina Cycle, it takes advantage of the properties of the binary fluid made of ammonia and water. Energy Concepts has a way of also using this cycle to convert exhaust heat from prime movers to electric power. The Heat Activated Dual Function Absorption Cycle is capable of taking a heat source ranging from 250°F to 750°F to and converting it to electric power, refrigeration and/or air conditioning. According to the company website when using the Absorption Cycle a 1 MWe gas turbine with an exhaust temperature of 750°F can produce 400 kW. They claim this cycle works well with distributed power generators sized from 1 to 15 MWe.

Rexorce

An Ohio based thermal energy company, Rexorce is looking to find ways to better harness existing thermal resources and also to recover waste heat and use it in an efficient manner. They have developed a thermal engine, Thermafficient, which is designed to recover thermal energy from a range of sources and convert it into electric power, cooling and heating. Rexorce's engine uses supercritical CO_2 and other working fluids for their power generating cycle. This binary fluid cycle has many of the same benefits as that of the Kalina Cycle. According to information from a company contact, they are working on

a 250 kW generator and claim 25-30% efficiency when working in diesel exhaust at 500°F. As of July 2008 they are still working on the expansion device and hope to have it completed in a few months. The price for the unit is expected to cost less than $1500 per kW. With a higher volume of orders that price could drop significantly.

Survey Summary

Company	Unit	Price	$$/kW	Delivery	Contact Info	Current Status/Notes
ORC Units						
Global Energy	30 kW	$60,000	$2,000	11 weeks	Greg Giese; greg@infinityturbine.com	30 kW unit sold and should be up and running within a month. Check back in early September 2008
	80 kW	No price quote	N/A	11 weeks		As of July 14, 2008 being built in Toronto
Barber-Nicholes	custom	$200K - $1M	N/A	N/A	David K.; davidk@barber-nichols.com;	Would require a custom design that would take time. It is unclear how long this would take and when a unit could be delivered
ElectraTherm	30-500 kW	Depends on size	$2,400-$2,700	N/A	Bill Olsen; bolson@electratherm.com	Much to prove before it becomes commerically viable
TransPacific Energy	115 kW	$250,000	$2,174	24-36 weeks	Jim Olsen; jolson@transpacenergy.com	No units installed anywhere yet. Only designs so far.
Delgue Inc.	250 kW	$400,000	$1,600	13 weeks	Brian Hageman; bhageman@delugeinc.com;	Not ORC but thermal hydraulic engine. Built first 250 kW unit and is being installed in HI now.
Ormat	> 200 kW	No price quote	N/A	N/A	Colin Duncan; cduncan@ormat.com	As of now recovering heat from a diesel generator is not an application they have units for. They are looking into it though for the future.
Turboden	>200 kW	No price quote	N/A	N/A	info@turboden.it	Have not been able to get in contact with company so details are scarce.
GMK	0.5 - 5.0 MW	No price quote	N/A	N/A	info@gmk.info	Deal with industrial power plants
UTC Power	280 kW	350,000	$1,250	8 weeks	bierderbp@utc.com	Commercial unit of Chena Hot Springs unit. 75 units sold to date. Working on 1 MW unit to be complete in early '09 and plan on possible smaller unit.
Kalina					-	

Cycle Units						
Energy Concepts	400 kW absorption cycle	No price quote	N/A	N/A	Don Erickson	Not clear if a specific unit is available or if their product would be custom designed per application
Rexorce	250 kW Thermal Engine	No price quote	$1,500	N/A	Michael Gurin 847-962-6180	Modified Kalina Cycle. Expect to be completed in a few months

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Appendix IB
Survey of Low Temperature Heat Engine Companies (2010)

Intro:
After a complete web search, 16 different companies were found to either produce ORC waste heat recovery units, or to have a plan to produce them in the near future. While the products themselves mostly focused on recovering waste heat some sort of power generation source, a lot of the companies only build units on a very large scale designed with multiple waste heat sources in mind. The companies reviewed were: Global Energy, Barber-Nichols, ElectraTherm, TransPacific Energy, Deluge Inc., Ormat, Turboden, GMK, UTC Power, BEP, KGRA Energy, BIOS Bioenergie, Calnetix Power Solutions, Cryostar, Energy Recovery Systems, and Energy Concepts. A general overview of the companies followed by a technical summary of their products follows:

Barber-Nichols
Barber-Nichols is one of the leaders in the ORC market for industrial sized ORC's. When communication was pursued, it was revealed that they are only interested in industrial sized ORC's at the moment and have no plans to produce anything small (10-100kw). They mention on their website that they have three major units in use right now, a 500kW unit, a 700kW unit, and a 2MW unit using waste heat and geothermal resources.

BEP Energy
BEP (Burke E. Porter) Energy is a subsidiary of BEP Corp. based out of Belgium. BEP Corp. produces a large variety of automotive and industrial testing equipment. BEP was contracted by ElectraTherm to produce the "Green Machine", a 50kW ORC unit that recovers waste heat through thermal oil or jacket water.

BIOS Bioenergie
BIOS Bioenergie is an Austrian company that specializes in producing energy from biomass and energy efficiency upgrades. They seem to produce a variety of different products, but on a custom built basis. No specific information on what the sizes of their products are/could be was given when asked.

Calnetix Power Solutions
Calnetixis a company based out of California that specializes in motors and generators, but has a subsidiary Calnetix Power Solutions that produces the "Clean Cycle" ORC unit. The unit is in use in three different locations around the world. Unfortunately, a large generator is necessary to produce enough power to run the "Clean Cycle" ORC (approximately 2.5MW). There is no such generator in Alaska, which could present a problem for use.

Cryostar
Cryostaris a company that specializes in industrial gas, clean energy, and hydrocarbon applications and is based out of France. On their website they advertise about an ORC unit that could run off of geothermal energy. However, there are no links to any

specifications, pictures, or further applications. After further investigation, it was revealed that the company does not actually have a product, but they do have plans to produce one. There was no size estimate as of now.

Deluge Inc.
Deluge is a company that makes one model of ORC unit and a lot of information about how their product is better than any other ORC unit. However, no specifications or pictures could be found. When asked if a product had been produced yet it was revealed that they have an idea for an ORC but no actual ORC in production. There has been no change in products or technology since the last survey was completed in 2008.

ElectraTherm
ElectraTherm is a company that has continued to grow and expand over the years and seem to know what they're doing with ORC technology. They patented the "Green Machine" and have since installed it at a couple different sites. They claim that the reason their product is superior to other ORC's is that they use a "Twin Screw Expander" which is more efficient than other expanders used in industry.

Global Energy
Global Energy is a company that specializes in building Organic Rankine Cycle (ORC) units ranging from a power output of 10kw to 250kw. The contact at the company Greg Geise mentioned in some email contact that he could potentially build an ORC unit anywhere in between his smallest unit (IT10-10kw unit) to the largest unit (IT250-250kw unit) He currently has a working model of the IT10 and has some video online displaying the power output. Global energy is unique in the way that they create their own turbines for their ORC's called the Infinity Turbine®. This turbine is a modular turbine that is easily assembled at the site of the ORC unit.

GMK
GMK (GesellschaftfürMotoren und Kraftanlagen) is a German company that specializes in ORC units in waste heat, geothermal, and biomass applications. They claim that they can produce a system that will use the waste heat from anywhere from a 500kw to 5MW Generator and have case studies that prove they have a product that is in use. Unfortunately, as of yet, no units have been installed or shipped to the United States.

KGRA Energy
KGRA Is a more unique company than most of the companies that produce ORC units. KGRA is willing to install the unit for you; however, you are not the sole owner of the unit. KGRA will continue to operate the unit and sell you power for a discounted price. Instead of free power you will receive discounted power. The specifics of the units they could install were vague but it seemed like the mission statement was directed at an industrial source.

Ormat
Ormat is the industry leader in the field of ORC units, however, they have discovered that larger products are more marketable and efficient and directed all their efforts towards

large scale industrial applications. Currently they have no plans to design or look into a smaller ORC unit that could be applicable in rural Alaska.

TransPacific Energy

TransPacific Energy is a company based out of California that specializes in ORC units in waste heat recovery and geothermal applications. Their claims are that their working fluid is the reason their product is so much better than the competition. They provide plenty of specifications on their working fluid. However, there are no specifications on their actual product.

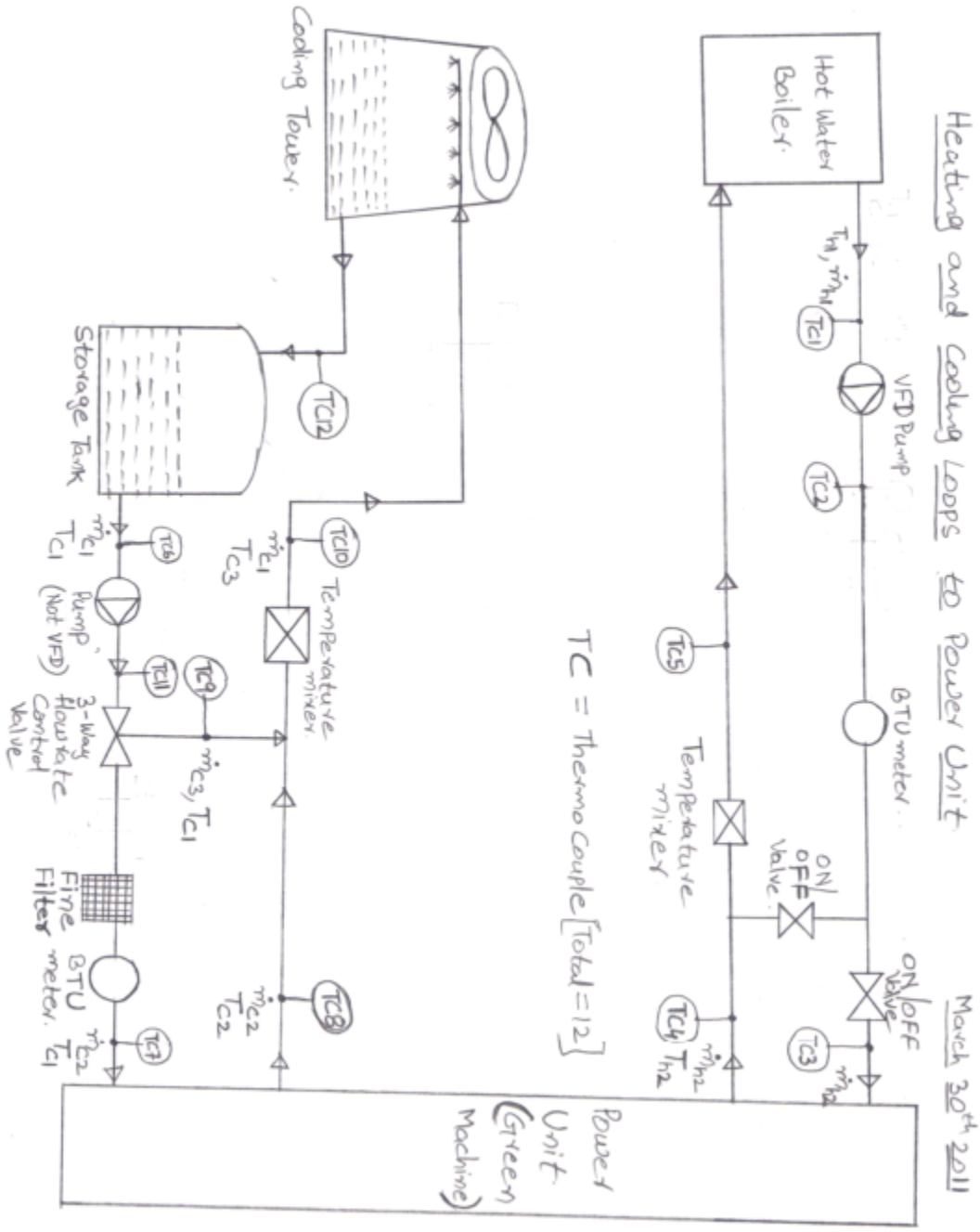
Turboden

Turboden is a company that is based out of Italy and sells a number of ORC units. They are known throughout Germany, Austria, and Italy as the world leaders in ORC technology. At the current moment they have nearly 150 units installed worldwide including a unit in Canada. Unfortunately the scale of the units is designed for a large industrial application. However, plans to reduce the size of the units are in place.

UTC Power

UTC (United Technologies Corporation) is the mother company of Turboden and Pratt & Whitney who produce the pure cycle ORC unit. The pure cycle unit is an ORC built for an industrial application and does not scale down very well. As Pratt & Whitney are a turbine company, another attempt at a smaller ORC is unlikely.

Appendix IIA
Preliminary Line Diagram of the Testing System



Appendix IIB
Preliminary Components Selected for the Testing System

Component	Size	Reason for selection
Green machine (GM)	Max power output: 50kW Min power output: 5kW Hot water supply temperature range: 180°F to 250°F. Cold water supply temperature range: 40°F to 110°F	Only commercial unit available in market at that time which can recover heat to power from stationary diesel engine jacket water temperatures (200°F)
Hot water boiler	Boiler was rated for 2400MBTU/hr output with hot water temperature variation from 155°F to 235°F.	i. Based on the heat load requirement for GM. At this stage the GM was supposed to be 50kW. At 7.0% efficiency of GM the GM would need 2400MBTU/hr of heat input. ii. Ability to change hot water temperature supply to GM from 155°F to 235°F
Cooling tower	Delta cooling tower model number T-150I, Induced draft cooling with cooling capacity of 150ton (1800MBTU/hr)	i. Based on the cooling load requirement for GM of 1800MBTU/hr
VFD pump for hot water loop	i. Bell & Gossett 20hp pump (1750rpm) rated for VFD operation. ii. Rated for 250gpm and head of 116feet of water	i. Pump was selected based on the pressure drop in the hot water piping calculated using pressure drop across various components (e.g. boiler, 4”pipe, GM heat exchanger etc.) ii. VFD pump is used to test the GM for various flow rates of hot water.
BTU meters	Detailed description of size and reason for selection is given in Table-2	
Cold water pump		
3-way butterfly valves		
Check valve		

Appendix IIIA
Methodology Proposed for Stage 2 and Stage 3 Modeling

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 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 simulation results for the extra operation conditions (using fitted parameter values) 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.

Appendix-IIIB
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-IIA gives the physical parameters of a typical plate heat exchanger taken from open literature [1] for a 50 kW ORC system.

Table–IIIB-1: Plate Heat Exchanger Physical Parameters considered in the present simulation

Plate Width (w in m)	0.5
Plate Length (L_p in m)	1.1
Channel Spacing (m)	3.50E-03
Thickness of Plate (t in m)	6.00E-04
Cheveron angle (β in degrees)	45
Enlargement Factor (Φ)	1.29
Corrugation Pitch	7.00E-03
Equivalent diameter (D_e in m)	7.00E-03
Projected area per plate (A_p in m^2)	0.55
Flow area on one fluid side (A_f in m^2)	4.20E-02
Surface area on one fluid side (m^2)	26.4
Hydraulic Diameter (D_h in m)	0.005426
Pressure drop in heat exchanger (MPa)	0.05
Plate Thermal Conductivity (K_p in W/m-K)	13.8
Plate Thermal Resistance (m^2-K/W)	4.35E-05
Area of Evaporator (m^2)	33.26
Area of Condenser (m^2)	28.3
Area of Pre–Heater (m^2)	15
Crltical Pressure of R245fa (MPa)	3.651
Crltical Temperature of R245fa (^oC)	154.01

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 [1] and is read as

$$h_f = (Nu \times K_f)/D_e \quad (1)$$

$$Nu = [0.2668 - 0.006967\beta + 7.244\times10^{-5}\beta]\times \mathrm{Re}^{[0.728+0.0543\sin[(\pi\beta/45)+3.7]]}\,\mathrm{Pr}^{1/3}(\mu/\mu_w)^{0.14} \quad (2)$$

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,

$$Re = (G \times D_e)/\mu \quad (3)$$

Where G is the mass velocity of the fluid, D_e is the equivalent diameter and μ 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 [2] and is read as

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

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 [3] and is read as

$$h_{Cond} = (Nu_{Cond} \times K_f)/D_h \quad (5)$$

$$Nu_{Cond} = Ge_1 \, \mathrm{Re}_{Eq}^{Ge2} \, \mathrm{Pr}^{1/3} \quad (6)$$

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

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

$$\mathrm{Re}_{Eq} = \frac{G_{Eq} D_h}{\mu_f} \quad (9)$$

$$G_{Eq} = G_c[1 - x + x(\rho_f / \rho_g)^{0.5}] \quad (10)$$

$$G_c = \frac{\dot{m}}{A_f} \quad (11)$$

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, ρ_f and ρ_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.

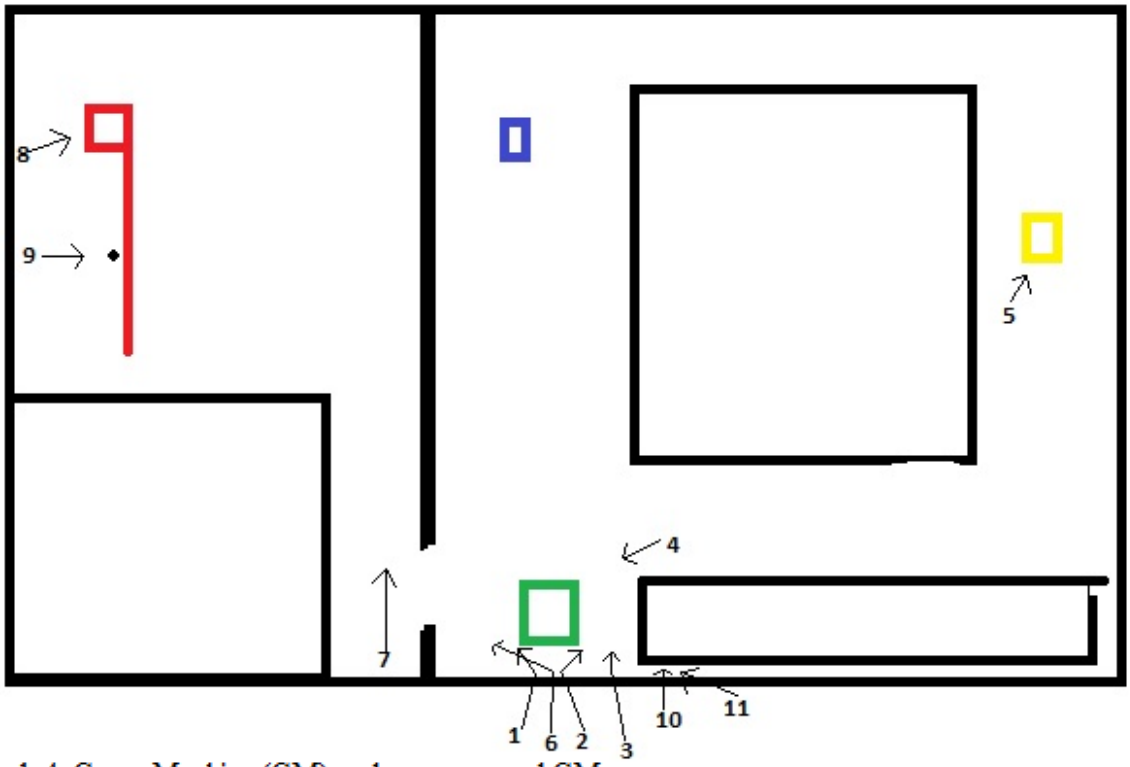
REFERENCE:

[1] Muley, A., Manglik, R. M., "*Experimental Study of Turbulent Flow Heat Transfer and Pressure Drop in a Plate Heat Exchanger with Chevron Angle*", ASME Journal of Heat Transfer, February 1999, Vol.121, pp.110–117.

[2] Ayub, Z. H., "*Plate Heat Exchanger Literature Survey and New Heat Transfer and Pressure Drop Correlations for Refrigerant Evaporators*", Journal of Heat Transfer Engineering, 2003, Vol.25 (5), pp.3–16.

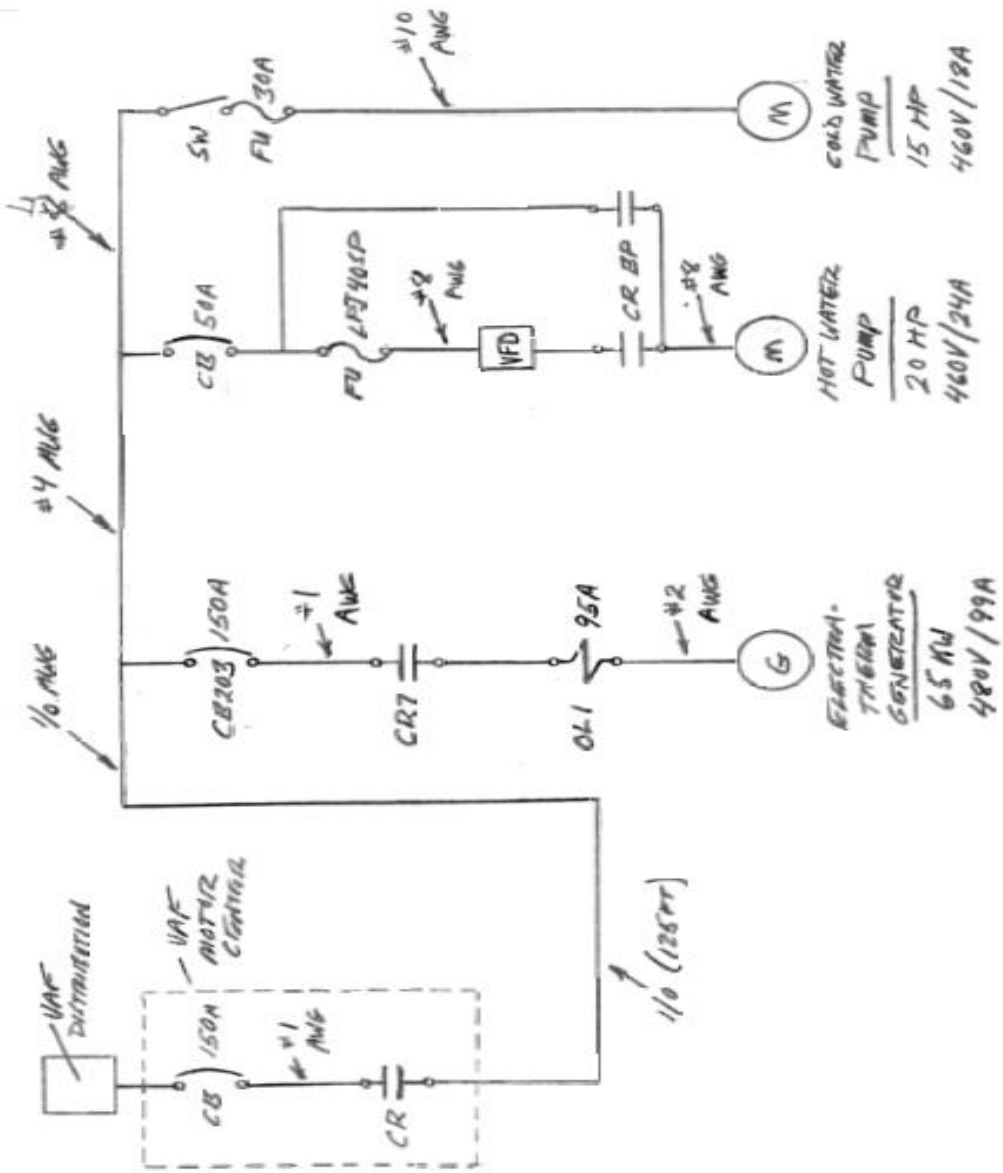
[3] Selvam, M. A. J., Senthil, K. P., Muthuraman, S., "*The Characteristics of Brazed Plate Heat Exchangers with Different Chevron Angles*", ARPN Journal of Engineering and Applied Sciences, December 2009, Vol. 4 (10), pp.19–26.

Appendix IVA
Available Floor Space for Test System Installation



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

Appendix IVB
Electrical Circuit Diagram



Appendix V
Pictures of the Integrated System after Installation

Steam flow control valve connected to steam-to-hot water heat exch*an*ger

Steam trap connected to steam-to-hot water heat exchanger

Hot water piping (with Gruvlok fittings) connected to steam-to-hot water heat exchanger

Hot water supply and return pipes to/from GM

Hot water pump (VFD pump) in hot water piping loop

Expansion tanks in hot water piping loop

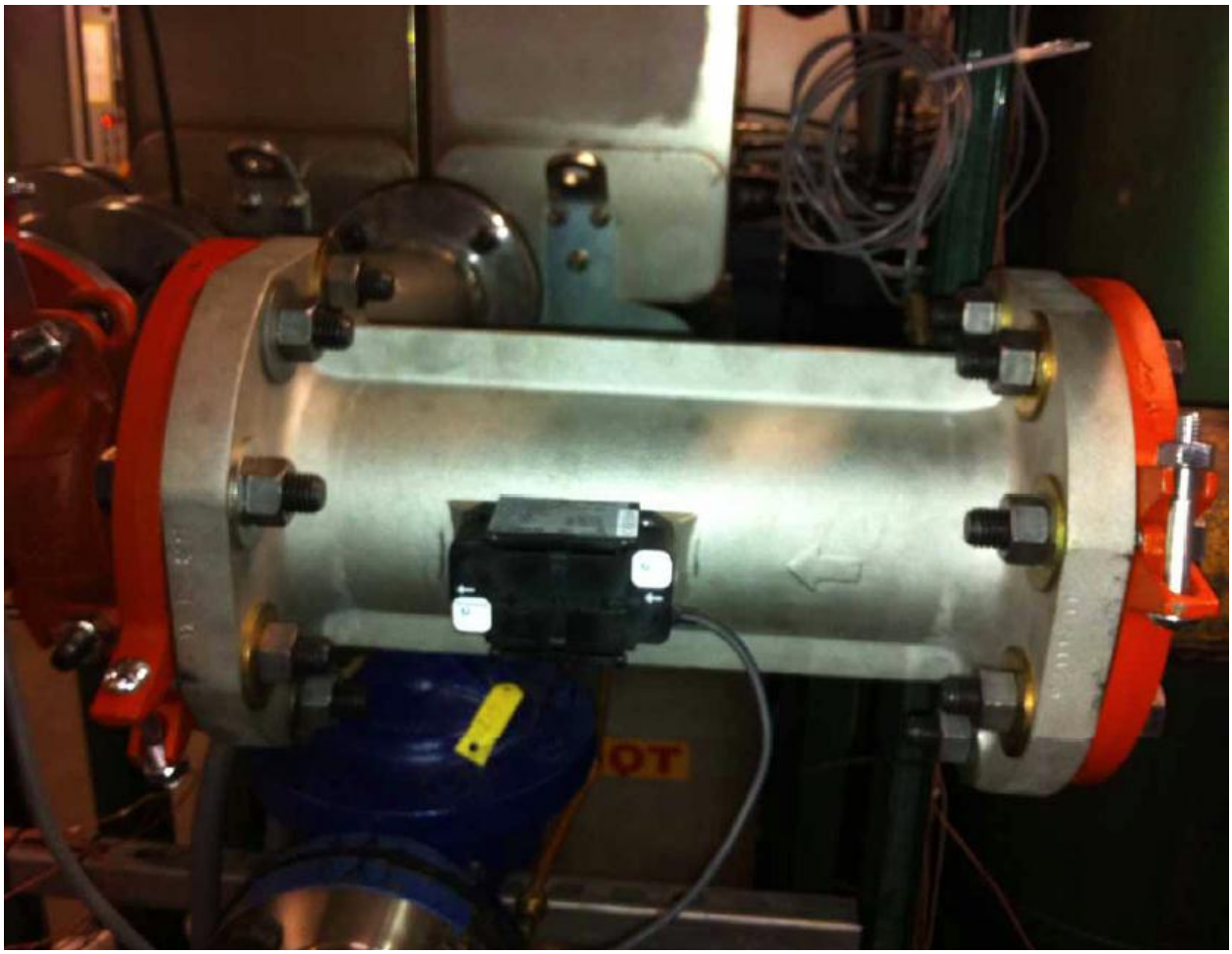
Air separator and pressure relief valve in hot water piping loop

More hot water supply and return piping to/from GM

BTU meter in hot water piping loop

Hot water supply and return piping connections on GM side

Cold water (heat sink fluid) supply and return piping to/from GM

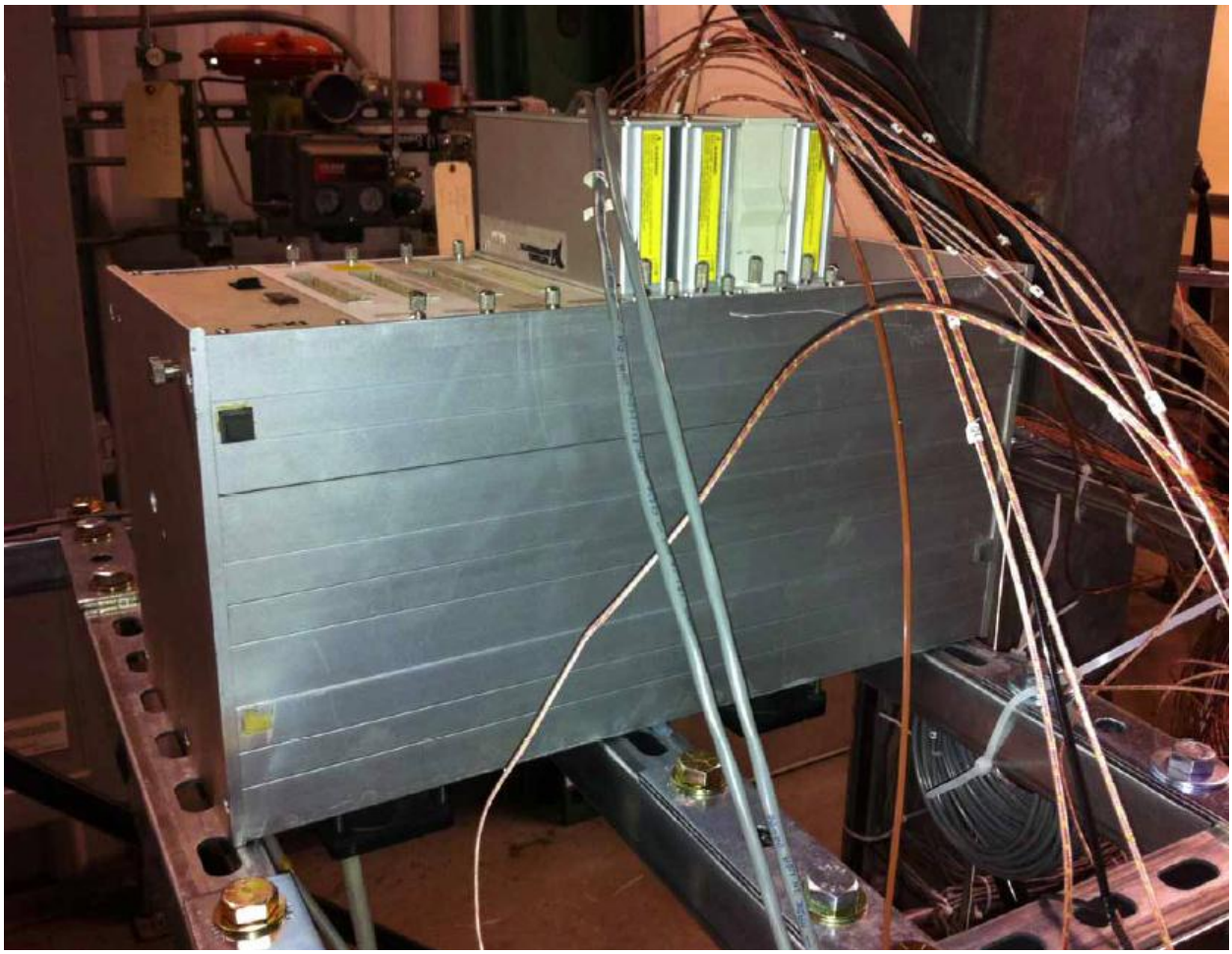


BTU meter on cold water supply piping to GM

Cold water bypass valves for temperature and flow control

Cold water pump

Check valve on cold water supply piping to GM to prevent back flow of water into fire hydrant



National instruments DAQ system being installed for data collection and performance analysis