ABSTRACT

In 2007 the electrical power consumption of 180 rural Alaska villages was 370,000MW-h, generated using isolated diesel gensets. From a stationary diesel engine considerable amount of heat energy at an elevated temperature is released into the atmosphere from engine jacket liquid and exhaust gases. In rural Alaska, due to the infrastructure, economic impact and needs of the villages, many of village gensets may not be appropriate for applying heat recovery for the purposes other than electrical power generation. Other appropriate types of heat recovery applications in Alaska may include desalination, refrigeration, and district heating. Also due to the varying sizes and electrical loads of most of the diesel gensets (from 100kw to 1MW); small sized heat recovery power systems (80kW or less) are preferred instead of industrial scale systems. In typical village diesel genset application most likely waste heat source could be the hot liquid from engine jacket and/or from exhaust-to-liquid heat exchanger. In the present work performance test was conducted on a 50kW ORC power unit under different heating and cooling conditions. The experimental setup consists of heat source loop, heat sink loop, electrical system and instrumentation (for data collection) for testing the ORC power unit. The ORC power unit was tested for hot water supply (heat source) temperatures varying from 68.3°C (155°F) to 107.2°C (225°F) and flow rate varying from 27.2m³/hr (120gpm) to 68.1m³/hr (300gpm); cold water supply (heat sink) temperatures of 10°C (50°F) and 20°C (68°F) and flow rate varying from 27.2m³/hr (120gpm) to 45.4m³/hr (200gpm). The performance test results will be used to make performance maps for ORC system which are in form of system characteristic plots for efficiency, operating power output, parasitic pump power consumption etc. with respect to different heating and cooling conditions. The data can be used in predicting long-term electrical power generation, efficiency, fuel savings, economic benefit (i.e. payback period) for a given heating and electrical load patterns. In addition emissions and CO₂ (GHG) reductions can also be estimated based on ORC electrical energy generation and fuel savings. If the ORC power unit is to be installed to recover waste heat from village diesel engines, it should be noted that power unit performance varies due to electrical load pattern, heat energy pattern, environmental conditions (e.g. for cooling source), infrastructure availability from village to village. The performance maps also provide power plant personnel with information that may be used in heat distribution for different heating and cooling conditions to optimize the benefit obtainable from diesel power plant waste heat. Different waste heat distribution applications may include heating, power, refrigeration etc. With the help of village power plant data an example is given in this paper for predicting the electrical power generation, efficiency, economic benefit etc. using the developed performance maps.

INTRODUCTION AND LITERATURE REVIEW

From a stationary diesel engine generator, which is the main source of electricity in circumpolar regions, about 60% of fuel energy is lost in the form of waste heat through charge air cooler (after cooler), jacket liquid cooler, friction and exhaust. This waste heat has low heat flux value (amount of heat rate) and low grade (i.e. low temperature) form of heat energy. Of the total fuel energy, diesel engine jacket liquid and exhaust account for about 20% and 30% respectively. If this low-grade heat is recovered for some useful applications, otherwise lost to atmosphere, considerable annual savings in diesel fuel and increase in power plant efficiency as a whole
could be achieved. Waste heat recovery applications may include heating (space heating, domestic hot water, or for warming municipal water supplies to prevent freezing), power generation, refrigeration and desalination.

In rural Alaska there are about 180 villages that run independent electrical power systems using diesel generators. In 2007 their electrical consumption was 370,000MW-h [1]. Taking 38% fuel efficiency of diesel engine, nearly 486,800MW-h amount of heat energy at an elevated temperature was lost to the atmosphere from engine jacket liquid and exhaust. Here it should be noted that the size of diesel gensets vary from about 100kW to 1MW in electrical capacity. Considering the jacket liquid heat recovery for heating, in rural Alaska this is a well-established technology and about 50% of rural villages in Alaska are equipped with jacket water heat recovery systems. Among them many are facilitated with combined jacket water and charge air heat recovery systems for heating purposes. The exhaust heat, due to concerns about cost, reliability, and possible maintenance problems, the exhaust heat was rarely recovered for useful applications at any of the village diesel power systems.

As stated earlier the applications of recovered heat from diesel engines may include desalination, refrigeration, space heating, and power generation. According to the Alaska Department of Environmental Conservation in March 2005, the mineral content of ground water for most Alaska villages was well within acceptable limits [2], therefore using diesel engine waste heat for desalination in Alaska villages is not justifiable. Applying diesel genset waste heat for refrigeration is not economical because, in Alaska ice is needed locally, such as in coastal villages with fishing industry, but only during 4 months of summer (from May to mid-September) and also unless a large commercial user for ice in the summer (such as a local fishing industry) exists, the application is not justifiable for whole Alaska.

Heating is required for 6 to 8 months of the year for all village residents in Alaska. Recovered heat can be used for space heating, domestic hot water, or for warming municipal water supplies to prevent freezing. Net heat energy recovered is highest for heating among all the applications. In general, about 50% or more of heat present in exhaust may be recoverable for heating. These factors led to the selection of heating as the diesel engine exhaust heat recovery application as part the previous work [1, 3, and 4]. Due to high cost of arctic piping, high installation costs, long distance between diesel power house and near-by buildings, the application of heat recovery for heating is not a viable option for every village.

Diesel engine waste heat to power conversion system may include thermodynamic systems or direct heat-to-electricity conversion systems. Examples of thermodynamic systems are Organic Rankine cycle (ORC) system or Ammonia water vapor absorption power system (e.g. Kalina cycle) and direct heat-to-electricity is thermoelectric generators (TEG). In industry, exhaust heat recovery using thermoelectric generators (TEGs), is still in the research stage [5]. The main drawback of a TEG is its temperature dependence. TEGs have better performance in a specified temperature range. Their efficiency falls off rapidly if temperature is below or above that specified range. Cost of an efficient TEG, one that works well over a varying temperature range, is prohibitively high. Maintenance of TEG units is also very high, making them unsuitable for remote Alaska village generators with large load fluctuations between day and night usages and summer and winter usages.

Diesel engine waste heat for power conversions using the thermodynamic cycles is a promising possibility for increasing the efficiency of Alaska village power plants. ORC has been implemented at industry level, and 8% to 18% recovery of heat has been achieved [6, 7, 8, 9, and 10] depending on heat source temperature, heat source type, heat flux rate etc. This is a considerable amount. The thermodynamic efficiency of the ammonia-water absorption cycle at industry level was reported to be 11% to 26% [11, 12, and 13]. Industry level power systems are the one which have power output more than 100kW and at 10% thermal efficiency (heat to power conversion efficiency) of the heat recovery system they would require 1MW of heat input or more. Therefore for the present work the stationary diesel engine genset waste heat recovery application selected was power generation using thermodynamic systems.

From the authors literature review for ORC and ammonia/water absorption power cycles, it was observed that no proven (i.e. for reliability and performance) physical system exists for low grade (low temperature) and low heat flux value i.e. less than 800kW of heat input (i.e. less than 80kW diesel waste heat for power systems @ 10% thermal efficiency) at the time of beginning of this work. If a system exists for low grade heat and low heat flux then they are only experimental systems [14, 15 and 16] which have not been proven for long term reliability test and readily available for installation (e.g. to recover heat from stationary diesel engines). In most of the rural Alaska, due to isolation of villages, no highly trained operation and maintenance personnel are available. Also due to isolation shipping fee (by air year round or by barge in summer) and travel are extremely expensive. Therefore the reliability, easiness in operation and fewer requirements in fabrication, installation and maintenance become the most critical factors in selecting the heat recovery application units. Based on these concerns, a retrofit unit with fewer requirements in fabrication of supporting systems is preferable. This led the research team to search for a semi-commercially available waste heat recovery unit which could be potentially used to recover heat from rural Alaska village diesel gensets. Semi-commercial unit is the one which is ready to deliver a working or
prototype unit but is lacking field testing data for performance, verification and system improvement. Table-1 gives the search results for companies which may be potentially have a semi-commercial power unit that could be used for diesel engine heat recovery. Of all the available commercial units for waste heat to power conversion, ElectraTherm® was the only potential candidate which manufactures a 50kW ORC unit and the manufacturer readily agreed to deliver a power unit. But then there were no published reliability and performance results of the 50kW ORC system in open literature which could be used to evaluate the performance (i.e. power output, efficiency, fuel savings, emissions reductions etc.) of the power system for rural Alaska conditions.

The main focus of this paper is to give performance test results in the form of maps and a method to predict the long-term outcome of installing this 50kW power unit on a village diesel engine for waste heat recovery. With the help of village power plant data an example is given in this paper for predicting the electrical power generation, efficiency, economic benefit etc. using the developed performance maps. This paper also gives a brief description of experimental setup used for testing a 50kW ORC power unit for reliability and performance. Reliability test is conducted at full load gross power output of the power unit (i.e. 50kW screw expander power output) for 600hrs to see the long term performance of the machine and the whole system. Performance test on power unit is conducted mainly to know the performance of power unit as a whole and its components (expander, evaporator, condenser, pump etc.) performance in particular at different hot water and cold water flow rates and temperatures. For performance test the power unit is 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, which are given in table-2. The total number of performance tests that were conducted on power unit was around 150. As explained above the performance test results will be used to make performance maps for this ORC system which are in form of contour plots for efficiency, operating power output, parasitic pump power consumption etc. with respect to different heating and cooling conditions. The data can be used in predicting long-term electrical power generation, efficiency, fuel savings, economic benefit (i.e. payback period) for a given village heating and electrical load patterns.

**EXPERIMENTAL SETUP**

This section presents brief description of experimental setup, with the help of line diagrams. This section also gives the components used to control heat source and heat sink flow rates and temperatures to power unit, method of uploading electrical power to grid and instrumentation for data collection. Here it should be noted that experimental setup was located in University of Alaska Fairbanks (UAF) power plant which was a coal fired CHP plant. As abundant amount of low pressure saturated steam (205.7kPa) was readily available, it was used as heat source for the ORC power unit.

Basic principle of ORC system is similar to that of steam Rankine cycle system except that the working fluid is organic (such as R134a, R245fa, R123, ammonia etc.) instead of steam. The basic components of ORC are pump, evaporator, expander, and condenser. The liquid refrigerant from the condenser is pumped at high pressure to evaporator. In evaporator the refrigerant is heated to the required superheated or saturated condition. This high pressure working fluid is converted to low pressure liquid or vapor/liquid mixture (to the condenser pressure) using an expander which is connected to the generator to produce power. The low pressure refrigerant from the expander is cooled to the desired state in condenser and the liquid portion is again pumped back to evaporator and the cycle continues. In the diesel generator waste heat application, the heating fluid used to heat ORC working fluid in evaporator may be from engine jacket liquid or 50/50 glycol/water mixture exiting exhaust heat exchanger or both combined, and is called heat source loop. In condenser the ORC working fluid rejects heat to cooling fluid (usually water) which may be from cooling tower, radiator, large water body (from a nearby river or lake), or underground well, and is called heat sink loop.

In the present ORC system, the working fluid was R-245fa. R-245fa is a non-flammable fluid with ozone depletion potential of zero and no listed phase-out year. R-245fa is used due to the match between the refrigerant properties, range of operation temperature and temperature range of available heat sources.

Figure-1 below shows the experimental setup for testing 50kW ORC power unit. The experimental setup consists of four major components: (i) Heat source loop, (ii) Heat sink loop, (iii) Electrical system (shown in figure-2), and (iv) Instrumentation (not shown in figures). In this experiment heat source loop is further divided into two loops viz. steam supply loop and hot water loop, hot water which exchanges heat with steam in steam-to-hot water heat exchanger was supplied to evaporator of the power unit as heating fluid (explained in “Heat source loop” section below). Cold water from a fire hydrant was used in heat sink loop. Electrical system is the wiring required to upload power to grid and wiring various power consuming components. Instrumentation is installing various data collection components such as flow meters, thermocouples, electrical meters etc.

**Heat Source Loop**

In the present testing of 50kW power unit the heat source was low pressure steam instead of waste heat from a diesel generator set. The reason for using steam as heat source was ease in designing the heat source loop and controlling heat
source temperatures to mimic temperatures and flow rates of engine jacket water condition for testing the power unit. For performance test as we will need to test the power unit at different heat source conditions (i.e. hot water flow rates and temperatures) which would not be possible by using a field engine waste heat, since a village field diesel generator set constantly needs to meet the village electrical load demand which may not cover the wide ranges of flow rate and temperature spectrum desired for the present testing (table-2).

Figure-1 shows the heat source loop which is further divided into two loops viz. steam supply loop and hot water loop. The major components used to build the heat source loop which on steam supply loop include steam-to-hot water heat exchanger, steam flow control valve with actuator and steam trap, and on hot water loop include 4" SCH40 black iron piping with Gruvlok fittings, hot water VFD pump, expansion tank, air separator, pressure relief valve.

In this experimental setup, power plant low pressure steam supplies heat to the hot water in a steam-to-hot water heat exchanger and this high enthalpy hot water exiting the heat exchanger is supplied as heating fluid to the evaporator of the ORC power unit. The low enthalpy hot water exiting the evaporator is again returned back to the heat exchanger to gain heat, thus the hot water is looping between steam-to-hot water heat exchanger and power unit evaporator. The outlet of the steam condensate is connected to the power plant condensate piping through a steam trap.

As explained above, for the purpose of performance test in which the ORC power unit is tested for different hot water flow rates and temperatures, it is necessary to design the hot water loop to accommodate for performance test. As shown in figure-1 on hot water loop piping, for flow rate control of hot water in the loop a variable frequency drive (VFD) pump was used. VFD was used to control the frequency input to the pump motor, thereby controlling the pump speed and hot water flow rate in the loop.

For temperature control of the hot water a steam flow control valve with electronic actuator was used. As shown in figure-1, steam flow control valve was used to obtain the desired hot water temperature exiting the heat exchanger, which is supplied as heating fluid to evaporator of power unit. By varying the amount of steam flow rate through heat exchanger the desired hot water temperature was obtained. Electronic actuator was used to remotely control the steam valve opening through National Instruments LabVIEW program. LabView VI software program takes in desired hot water temperature as input and based on the actual hot water temperature a control signal is initiated by LabView which controls the valve opening position. Normally closed (NC) valve was used for safety concerns.

Heat Sink Loop

Due to location of experimental setup at UAF power plant, the cooling source was water from a fire hydrant which was located just outside the power plant building.

Figure-1 shows the heat sink loop which is an open cold water loop. The major components of the loop are manual flow control valve, check valve, pump, and two 3-way butterfly valves with bypass line for temperature control. Temperature of water from fire hydrant is always around 10°C (50°F). The working principle of the loop was the cold water from the fire hydrant flow through the condenser of the power unit extracting excess heat from the refrigerant, there by cooling the refrigerant to condenser pressure, the warm water out of the condenser is diverted to a heat sink. The bypass line is used to test the power unit for different cold water temperatures other than 10°C during the performance testing. The 3-way butterfly valves were operated such that a portion of warm water from the condenser was recirculated through the bypass and mixed with the cold water from fire hydrant and supplied to power unit. The warm water flow through the bypass line was achieved by operating the butterfly valves and turning on the pump. The position of 3-way butterfly valves were adjusted until the desired cold water temperature into the power unit is reached. Manual flow control valve mounted directly on the fire hydrant outlet was used to control cold water flow rate to the power unit.

The water pressure from the fire hydrant was enough to move water all along the loop. The pump, 3-way butterfly valves and bypass were used only when the power unit was tested for different cold water temperatures (other than 10°C) i.e. only during the performance test otherwise they are shutoff.

Electrical System

This section mainly covers the general description of electrical wiring system which is, with the help of line diagram, the method adapted to upload generated power to power plant, powering main electricity consuming equipment viz. hot water and cold water pumps.

As the ORC power unit was located within the UAF power plant, the generated power from power unit was tied into the UAF motor center (located in the power plant) where it is uploaded to University power distribution system. Figure-2 shows the line diagram for electrical wiring from power unit generator to motor center and wiring for both hot and cold water pumps. Here the electrical wiring is done in such a way that when ORC power unit is generating power, it uploads power to motor center as well as it powers hot and cold water pumps. When the power unit is not generating power, the hot and cold water pumps can still be operated by drawing power from motor center. As shown in figure-2, from the power unit circuit breaker (CB) box 3/0AWG (American wire gauge) metal clad (MC) 3-conductor with ground cable was connected to UAF motor center CB box to upload power to University power system. From the same power unit CB box
# Parameters Measured, Instrumentation and Data Collection

This section mainly discusses about the parameters measured, instruments installed for measuring and data collection. The main focus of this paper is to give performance test results of 50kW ORC power unit, which include heat input, heat rejected, system operating power output (i.e. the power uploaded to grid), efficiencies, emissions, fuel savings, and payback period. Parameters measured are the direct measurements taken from installed measuring equipment (i.e. instrumentation) which will be used in data reduction process for further analysis of the system.

The various parameters measured during the test of 50kW ORC power unit were (i) hot water flow rate, inlet and outlet temperatures to power unit ($V_{HW}$, $T_{HW,in}$, $P_{HW,in}$), (ii) cold water flow rate, inlet and outlet temperatures to power unit ($V_{CW}$, $T_{CW,in}$, $P_{CW,in}$), (iii) electrical power output of power unit ($P_{Net}$), (iv) electrical power consumed by power unit pump ($P_{Pump,p}$), (v) hot water pump power ($P_{Pump,HW}$), and (vi) cold water pump power ($P_{Pump,CW}$). Here it should be noted that electrical power output of power unit ($P_{Net}$) already considers the power unit pump electrical power consumption (figure-2). Cold water pump power consumption was estimated based on hot water pump power (for same flow rate) due to use of fire hydrant as cold water source.

### Instrumentation and Data Collection

For flow rate, as shown in figure-1, Kamstrap Ultraflow® ultrasonic flow meters were used to measure the hot water and cold water flow rates supplied to power unit. Kamstrap Multical-601® calculator, which has flow rate display, was used to manually note the flow rates. Omega® type-K thermocouples were used to measure the inlet and outlet temperatures of hot water and cold water. Temperature measurements were stored in excel format using LabView VI program (see next paragraph). For electrical power measurement, as shown in figure-2, EKM-353 EDM electrical meters were used to measure electrical power generated by power unit, power consumption by power unit pump and hot water pump. Electrical meter manufacturer had custom software which was used for reading real time electrical power measurement and this real time data was stored in text format at every 30s interval for future data reduction.

Data acquisition and control (DAQ) functions were performed using a LabView virtual instrument program (VI) operating on a National Instruments (NI) PCI-MIO-16E module. LabView VI software was used to read the real time data and to store this data at one second interval in excel format for future data reduction. For temperature measurement NI SCXI-1120 analog input board was used. Steam valve position was controlled by simple LabView VI software program and SCXI-1121 analog I/O board. LabView VI software program takes in desired hot water temperature as input and based on the actual hot water temperature a control signal is initiated by LabView which controls the valve opening position.

### EXPERIMENTAL PROCEDURE

Experimental procedure is the operation procedure followed to operate the power unit for each case of performance test. As stated in “Introduction” section, the performance test on power unit is conducted mainly to know the performance of power unit and its components performance at different hot water and cold water flow rates and temperatures. The procedure of performing this test (i.e. method of changing temperatures and flow rate, data collection etc.) is explained in sequential steps below.

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 ORC power unit 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 27.2m³/hr (120gpm) and 55Hz to 68.1m³/hr (300gpm) of hot water flow), we set the desired hot water flow rate. The hot water flow rate can be read in flow meter display 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 30minutes for steady state condition for data collection.

5. Steady state data collection is done for 30minutes at one set of hot water and cold water temperature and flow rate. 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-5.

During steady state data collection (Step-5 above) for 30 minutes, the hot water and cold water temperatures are stored by LabVIEW in excel format at frequency of 1 sec. The hot water and cold water flow rates are manually noted from the respective flow meter calculator displays. The electrical power generated by power unit and parasitic power consumption by pumps are stored in text format at frequency of 30 sec. For each case of hot water and cold water flow rate and temperature, all the steady state measured parameters are averaged for data reduction process.

**DATA REDUCTION**

This section gives the mathematical expressions used in obtaining the derived parameters from measured parameters, which will be useful in further analysis of the power unit. This section also discusses the procedure and methodology adopted to estimate the reductions in emissions and CO₂ and the economic impact of installing an ORC power unit on village diesel power plant.

Heat supplied \( Q_{HW, Su} \) by hot water to evaporator of power unit is obtained by,

\[
Q_{HW, Su} = V_{HW} \times \rho_{HW} \times (h_{HW, in,P} - h_{HW, out,P}) \quad (1)
\]

Here density of hot water \( \rho_{HW} \), inlet enthalpy \( h_{HW, in,P} \) and outlet enthalpy \( h_{HW, out,P} \) of hot water to power unit were obtained based on evaporator hot water inlet and outlet temperatures and using NIST REFPROP 8.0 [17] program. \( \rho_{HW} \) is the average density of hot water obtained at inlet and outlet evaporator hot water temperatures.

Heat rejected \( Q_{CW,Rej} \) to cold water by condenser of power unit is obtained by,

\[
Q_{CW,Rej} = V_{CW} \times \rho_{CW} \times (h_{CW, out,P} - h_{CW, in,P}) \quad (2)
\]

Here density of cold water \( \rho_{CW} \), inlet enthalpy \( h_{CW, in,P} \) and outlet enthalpy \( h_{CW, out,P} \) of cold water to power unit were obtained based on condenser cold water inlet and outlet temperatures and using NIST REFPROP 8.0 [17] program. \( \rho_{CW} \) is the average density of cold water obtained at inlet and outlet condenser cold water temperatures.

System operating power output \( P_{OP} \) is the power generated by power unit which was uploaded to university power system, given by Eq.(3), which considers the power unit and cold water pump powers. Here in calculating system operating power output \( P_{OP} \), the ORC power unit and cold water pump power consumptions were only considered because in general a stationary diesel engine is equipped with jacket water pump to dissipate heat to atmosphere using air coolers and as stated in “Introduction” section that most of the rural Alaska diesel gensets are equipped with jacket water heat recovery system which may have a pump already installed. Taking this into account, the electrical power consumed by hot water pump is neglected assuming the already installed jacket water pump can be used to overcome the ORC power unit evaporator pressure drop. \( P_{OP} \) will be used in annual diesel fuel saved, emissions reductions and economic outcome calculations discussed in following paragraphs. Here both \( P_{Net} \) and \( P_{Pump, CW} \) are measured parameters explained in “Parameters measured” section above. Eq.(4) gives the expression for system operating efficiency \( \eta_{OP} \) which is the ration of \( P_{OP} \) and \( Q_{HW, Su} \).

\[
P_{OP} = P_{Net} - P_{Pump, CW}
\]

\[
\eta_{OP} = \frac{P_{OP}}{Q_{HW, Su}}
\]

Liters (or gallons) of diesel fuel saved per year \( (F_{S/Y}) \) was calculated using Eq.(5) which was based on system operating power output \( P_{OP} \), 355 power unit working days per year with 10 days of maintenance, and stationary diesel engine specific fuel consumption. Stationary diesel engine specific fuel consumption of 3.7 kWh/lit (14 kWh/gal) [18, 19] is a reasonable value for rural Alaska village diesel gensets. Dollar amount saved on diesel fuel per year \( (F_{S/Y}) \) was calculated based on diesel fuel saved per year \( (F_{S/Y}) \) and diesel fuel cost of $5.0/gal, which is a reasonable value for rural Alaska stationary diesel generator power plants.

\[
F_{S/Y} = \frac{P_{OP} \times 355 \times 24}{3.7}
\]

**Economic Analysis**

The economic impact of installing an ORC power unit on rural Alaska village power plant was evaluated based on payback period calculations. The payback period is determined when enough money has been accumulated at given simple interest rate to offset the total initial investment cost \( (T_{ini, Cap}) \) and annual maintenance cost based on annual cost savings. Here annual cost savings is the dollar amount saved on diesel fuel per year \( (F_{S/Y}) \) in operating the ORC power unit on recovered waste heat from a rural Alaska diesel.
engine power plant. Note that dollar amount saved on diesel fuel per year \((F_{SV})\) was calculated based 355 power unit working days per year with 10 days of maintenance explained in above section.

The total initial investment cost \((T_{ini\text{-}Cap})\) can be further divided into component costs \((C_S)\) and installation costs \((I_S)\). Component costs \((C_S)\) are the material and instrumentation cost incurred on building the whole heat recovery system and data acquisition system. For the present case the component costs \((C_S)\) include cost of purchasing ORC power unit, steam-to-hot water heat exchanger, steam valve, hot and cold water pumps, air separator, expansion tank, pressure relief valve, pipes for hot water and cold water, flow meters, thermocouples, Gruvlok fittings, supporting structural material (e.g. struts, pipe hangers), electrical cables, other miscellaneous parts (e.g. nuts, bolts, tees, pipe couplings), freight charges etc. Table-3 gives the categorized component cost incurred on building the present experimental system and component cost \((C_S)\) for the present experimental setup was estimated to be $191,000.

The installation cost \((I_S)\) may include the number of days for installation, number of personal required for installation, cost of labor per hour per person, travel cost (if any) and other installation costs. Based on our present experience on installing the experimental system it would require 5 personal and 30 days for complete installation of hot water loop, cold water loop, electrical system and instrumentation to ORC power unit (assuming all the components are available for installation). Assuming a labor cost of $70/person/hour, and $5,000 for travel, the total cost of installing \((I_S)\) the whole system comes to $89,000 and this value is used in the present payback period calculations.

Therefore the total initial investment cost \((T_{ini\text{-}Cap})\) is estimated at $280,000, which is the sum of component costs \((C_S)\) and installation costs \((I_S)\).

According to power unit manufacturer and from our reliability test experience, the maintenance requirement for the present ORC machine is similar to the maintenance requirement for air-conditioning and refrigeration systems, and minimal in economic concerns. The expected maintenance is mostly visual inspection and simple measurements, small changes (e.g. belts, lubricant, filters, batteries) and simple cleaning jobs. Considering the maintenance requirement for exhaust heat recovery system, Lin [1] and Raghupatruni [3] have determined that 2 days of maintenance per year is required. The effect of 10 maintenance days (considered in this paper for payback period estimation) on economics is estimated to supersede the effect of maintenance requirements in estimated real machine turnoff days plus labor and parts needed.

### Reductions in Emissions and CO₂

As the ORC power unit was designed to operate on waste heat from a village diesel genset (i.e. free heating source); it would offset some of the power needs directly from the diesel generator of the village and intern lead to emissions reduction. Annual emissions reductions were estimated based on the annual system operating power output by power unit (355 power unit working days per year with 10 days of maintenance) and stationary diesel engine emissions given in Table-4. Annual CO₂ reductions were based on Liters (or gallons) of diesel fuel saved per year \((F_{SV})\). Table-4 gives the Tier-4 interim emissions standards set by EPA for non-road diesel engine gensets [20, 21].

### RESULTS

The purpose of this paper is to present the performance test results conducted on 50kW ORC power unit which would be used to estimate the economic effect of application of this unit on individual gensets. To make the integrity of the performance description for this ORC power system, the outcome of the 600 hour reliability test results are briefly discussed in the below section. The outcome of the performance test results are discussed in detail in the following section.

#### Reliability Test Results

Reliability test was completed on the ORC power unit at full load (i.e. 50kW expander power output) for 600hrs to know the long-term endurance and performance of the unit. For reliability test the average hot water supply temperature was 104.2°C and flow rate of 36.28m³/hr similarly the average cold water conditions were 9.7°C and 37.15m³/hr which were based on manufacturer's specification for full load operation. The following observations were made from reliability test results:

1. No major problems were observed with the ORC power unit, such as drift in power output during long-term operations, power unit shut downs etc., during the reliability test.
2. During reliability test the average electrical power output \((P_{Net})\) by power unit was obtained to be 47.8kW (expander power less ORC pump power) with system operating efficiency \((\eta_{OP})\) of 7.5%.
3. ORC power unit achieved screw expander efficiency (ratio of expander output to heat input) of 8.4% at full load operation was well within the manufacturer's claim of 8.5%.
4. Payback period of 2years and 2.3years was obtained with 0% and 10% interest rate on capital respectively.

#### Performance Test Results

Performance test on 50kW ORC power unit was conducted varying four different input parameters viz. hot water flow
DISCUSSION

For a given hot water supply temperature and cold water flow rate, heat supplied by hot water \( Q_{HW,Su} \) to power unit evaporator increased with the increase of hot water flow rate as shown in figure-3. For example, at hot water supply temperature of 79.4°C (175°F) and cold water flow rate of 36.34m³/hr (160gpm), heat supplied by hot water increased from 327.4kW at 27.17m³/hr of hot water flow rate to 380.7kW at 68.43m³/hr of hot water flow rate.

In some cases of hot water supply temperature and cold water flow rate the irregular nature in increase of \( Q_{HW,Su} \) is due to occasional disturbance in hot water supply temperature resulted from the surge in power plant steam supply condition. For example at hot water supply temperature of 90.5°C (195°F) and cold water flow rate of 36.34m³/hr (figure-3), the actual hot water supply temperature was 91.03°C at 36.34m³/hr to 90.38°C at 45.4m³/hr. It could be observed from figure-4 and figure-5 that same trends (of heat supplied in figure-3) are followed for heat rejection to cold water and operating power output if there is any disturbance in hot water supply temperature.

As the hot water flow rate increased for a given hot water supply temperature, the heat input to power unit reached asymptotic condition (figure-3) i.e. for a given hot water supply temperature the heat absorption by working fluid in the evaporator reached a limiting value for higher hot water flow rates. The same trends were observed for system operating power output as it reached asymptotic condition for higher hot water flow rates (figure-5). The reason for this asymptotic condition is the ORC power unit evaporator reached its design capacity. There is another limitation from ORC unit PLC software, which prevents the screw expander from generating more than the rated load of 50kW. The PLC software limitation, which limits the R-245fa flow entering the screw expander, is one of the many safety features which protect the screw expander from over-speeding.

From figure-3, figure-4, figure-5, it could be observed that, for a given hot water flow rate and hot water supply temperature the effect of cold water flow rate on heat input to power unit evaporator, heat rejected to cold water in condenser and system operating power output is minimum. For example at hot water flow rate of 45.4m³/h (200gpm) and hot water supply temperature of 90.5°C (195°F), for cold water flow rates of 27.2m³/h (120gpm), 36.34m³/h (160gpm) and 45.4m³/h (200gpm), heat supplied by hot water was 467.75kW, 471.67kW and 492.42kW respectively, heat rejected to cold water was 418.4kW, 422.37kW and 446.29kW respectively, and system operating power output \( (P_{OP}) \) was 32.43kW, 32.93kW and 33.23kW. Therefore in plotting the performance curves, figure-6, figure-7, figure-8, figure-9, figure-10, figure-11, for a given hot water flow rate and hot water supply temperature it was determined to average the heat input, heat rejected and \( P_{OP} \) over 3 cold water flow rates. That is in the above example, at hot water flow rate of 45.4m³/h (200gpm) and hot water supply temperature of 90.5°C (195°F) heat supplied by hot water was 477.28kW, heat rejected to cold water was 429.0kW, and system operating power output \( (P_{OP}) \) was 32.86kW.

Figure-6, figure-7, figure-8, figure-9, figure-10, figure-11 gives the heat supplied by hot water, heat rejected to cold water, system operating power output, efficiency, payback period and reductions in \( CO_2 \) emissions for different hot water supply temperatures, hot water flow rates and cold water supply temperatures of 10°C and 20°C. In each of the plot (figure-6, figure-7, figure-8, figure-9, figure-10, figure-11), the top plot is for 10°C cold water temperatures and the bottom plot is for 20°C cold water temperatures. All the six plots are presented on the same hot water supply temperature scale and with same color coding for ease of reading i.e. for example at hot water supply temperature of 101.6°C (215°F) and hot water flow rate of 45.4m³/h, from figure-6 the hot water heat input to the evaporator is 606.6kW and 588.8kW for 10°C and 20°C cold water temperature respectively; from figure-7 heat rejection to cold water is...
552.61kW and 468.9kW for 10°C and 20°C cold water temperature respectively; from figure-8 the system operating power output is 44.18kW and 41.8kW for 10°C and 20°C cold water temperature respectively; from figure-9 the system operating efficiency is 7.3% and 7.1% for 10°C and 20°C cold water temperature respectively; from figure-10 the payback period of 2.4years and 2.6years for 10% interest rate on capital could be achieved for 10°C and 20°C cold water temperature respectively; and from figure-11 CO₂ reductions of 300short-tons/year and 282.3short-tons/year for 10°C and 20°C cold water temperature respectively could be achieved. Payback periods and CO₂ reductions were calculated based on equations and procedure discussed in above section of “Data Reduction”.

In Figure-6, figure-7, figure-8, figure-9, figure-10, figure-11, for cold water temperature of 20°C, the results were presented only up to the maximum hot water supply temperature of 101.6°C (215°F). This is because of the low saturated steam pressure in power plant which prevented the hot water supply temperature reaching the expected maximum of 107.2°C (225°F) during the test.

Example Based on Above Performance Curves

From the Power Cost Equalization (PCE) program data [18] published by Alaska Energy Authority for fiscal year 2011 and based on available diesel engine data at the location, Tok, Alaska was selected for evaluating diesel engine waste heat recovery for power generation using present ORC system. Based on PCE data Tok annual electrical load is 10,902,597kWh and all of this power is generated using isolated Caterpillar 2MW diesel engine. The specifications of the engine are given in table-5. Table-6 gives the diesel engine power output, specific fuel consumption, exhaust temperature, heat rejected by engine to jacket water and exhaust at different loads of the engine. Here note that the heat present in exhaust is based on lower heating value of exhaust i.e. cooling exhaust up to only 176.6°C (350°F) to avoid acid formation in exhaust manifold.

From the annual electrical load consumption, the average electrical load on diesel engine is 1250kW (1676.2hp). Considering 1.3MW (1700hp) as average load on diesel engine, the average percent load on diesel engine is 65.7%. By interpolation with between 50% and 75% engine data for 65.7%, table-6 also gives the diesel engine data at this load.

For evaluating the ORC performance for waste heat recovery from stationary diesel engines two cases were simulated, first being the jacket water heat recovery system only and second being the combined jacket water and exhaust heat recovery. For both of the simulation cases it was assumed that a water cooling source as heat sink is readily available at 10°C, which is about the year round ground water temperature in Tok, with flow rate ranging from 27.2m³/h (120gpm) to 45.4m³/h (200gpm).

Table-5 gives the engine jacket water temperature at 99°C (210.2°F). Assuming 200gpm jacket water is bypassed to be supplied as heat source for ORC power unit, table-7 gives the results for operating this ORC power system on waste heat from jacket water of 2MW diesel engine.

It can be observed from table-6 that the exhaust temperature at 65.7% engine load is 402°C, which is well above the 107.2°C (225°F) required for ORC to generate maximum system operating power. For the simulated case of combined jacket water and exhaust heat recovery system, if the heat recovery system is designed such that the jacket water from the engine is first passed through exhaust heat exchanger, it is possible to achieve 107.2°C (225°F) as hot water supply temperature for ORC power unit evaporator. Table-7 also gives the ORC power unit performance for both jacket water, and combined jacket water and exhaust heat recovery system is installed together. In table-7, the heat input to power unit, system operating power output, efficiency, payback period and CO₂ reductions can be obtained from Figure-6, figure-7, figure-8, figure-9, figure-10, figure-11. For combined jacket water and exhaust heat recovery the system operating power output is 45.7kW with payback period of 2.3years could be achieved. Considerable reductions in emissions could be achieved, as listed in table-7, which were calculated based on the EPA TIER-4 interim reduction standards discussed earlier.

CONCLUSIONS

The testing system for heat source loop, heat sink loop, electrical system and instrumentation has been designed and installed. The installation process was smooth and the power unit was tested for performance according to the experimental procedure discussed. Based on the experimental test results following conclusions could be drawn:

1. Application of this 50kW ORC power unit for waste heat recovery application from stationary diesel gensets is expected reliable and feasible in rural Alaska as the maintenance requirement and level of expertise required to operate the power unit is expected minimal.

2. It was observed that the effect of cold water flow rate on heat input, heat rejection, power output was minimum for a given cold water supply temperature, hot water flow rate and hot water supply temperature.

3. For a given hot water supply temperature with the increase of hot water flow rate, the heat input to power unit and system operating power output reached asymptotic condition.

4. Performance curves were plotted for heat input to evaporator, heat rejected to cold water, system operating power output, efficiency, payback period and CO₂ emission
reductions with respect to hot water supply temperature for 10°C and 20°C cold water supply temperatures respectively.

5. For all hot water supply temperatures except for 68.3°C (155°F) (or lower), the payback period of less than 6.5 years and 8 years could be achieved for 10°C and 20°C cold water temperatures respectively.

6. An example to evaluate the present ORC system using the field diesel engine data is presented for jacket water heat recovery, and combined jacket water and exhaust heat recovery systems using the developed performance curves. The example shows that the performance data obtained from this experiment can be used to simulate and evaluate the application of this ORC system to Alaska village genset for power output, efficiency, payback period, emissions reductions etc.

7. For jacket water temperature at 99°C (210.2°F), 41.7kW system operating power output was achievable with 7.2% efficiency and 2.6 years payback. From our observation of example results, it is possible to generate 45.7 kW system operating power output with 7.4% efficiency and 2.3 years payback using this ORC power unit working on waste heat from stationary diesel engines if the waste heat is from both jacket water and exhaust heat exchanger.

8. Considerable amount of annual emissions and CO₂ (GHG) reductions could be obtained if the ORC power unit was operated year round on waste heat from diesel engines.

9. Considering the 370,000MW-h of electrical consumption of whole Alaska and taking 38% fuel efficiency of diesel engine, nearly 486,800MW-h of heat energy is present in jacket water and exhaust heat. Using this waste heat, at 7% ORC efficiency, about 34080MW-h of electricity possibly can be generated which would increase the diesel engine fuel efficiency to 41.5%, with CO₂ reductions of 27000short-tons/year, fuel savings of 9214800lit/year (2434300gal/year) and fuel cost savings of $12,171,500/year.

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20. U.S. Environmental Protection Agency, “Control of
emissions of air pollution from non-road diesel engines and
fuel”, Rules and Regulations, Vol.69 (124), pp. 38980, June

Average carbon dioxide emissions resulting from gasoline

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tools and equipment for installation, UAF Power Plant for
providing lab space for performing the experiment and UAF
Facilities Services for providing personnel during installation
of tough and heavy components.
Figure 1. Experimental setup for testing 50kW ORC power unit

Figure 2. Line diagram for electrical wiring for uploading power to University system and powering hot water and cold water pumps
Figure 3. Heat input to power unit evaporator vs. hot water flow rates at different hot water supply temperatures and cold water flow rates
Figure 4. Heat rejected to cold water in power unit condenser vs. hot water flow rates at different hot water supply temperatures and cold water flow rates
Figure 5. System operating power output vs. hot water flow rates at different hot water supply temperatures and cold water flow rates
Figure 7. Heat rejected vs. hot water supply temperature
Figure 8. System operating power output vs. hot water supply temperature
Figure 9. System operating efficiency vs. hot water supply temperature
Figure 10. Payback period vs. hot water supply temperature
Figure 11. CO₂ reductions vs. hot water supply temperature
### Table 1. Review conducted for available heat-to-power conversion systems

<table>
<thead>
<tr>
<th>Company</th>
<th>Unit</th>
<th>Price</th>
<th>$/kW</th>
<th>Status/Notes in Summer-2008</th>
</tr>
</thead>
<tbody>
<tr>
<td>Global Energy</td>
<td>30kW</td>
<td>$60,000</td>
<td>$2,000</td>
<td>30kW unit sold, expected to be online within by September 2008.</td>
</tr>
<tr>
<td>Global Energy</td>
<td>80kW</td>
<td>No price quote</td>
<td>N/A</td>
<td>As of July 14, 2008 being built in Toronto</td>
</tr>
<tr>
<td>Barber-Nicholes</td>
<td>Custom design</td>
<td>$200K - $1M</td>
<td>N/A</td>
<td>Would require a custom design. It is unclear how long this would take for a unit to be delivered.</td>
</tr>
<tr>
<td>ElectraTherm</td>
<td>50kW to 65kW</td>
<td>$120,000</td>
<td>$2,400-$2,700</td>
<td>Unit available but must be proven before it becomes commercially viable.</td>
</tr>
<tr>
<td>TransPacific Energy</td>
<td>115kW</td>
<td>$250,000</td>
<td>$2,174</td>
<td>No units installed, only designs.</td>
</tr>
<tr>
<td>Deluge Inc.</td>
<td>250kW</td>
<td>$400,000</td>
<td>$1,600</td>
<td>Not ORC but thermal hydraulic engine. Built first 250kW unit and is being installed in HI.</td>
</tr>
<tr>
<td>Ormat</td>
<td>&gt; 200kW</td>
<td>No price quote</td>
<td>N/A</td>
<td>As of summer-2008 recovering heat from a diesel generator is not available.</td>
</tr>
</tbody>
</table>

#### Ammonia/water vapor absorption power systems

<table>
<thead>
<tr>
<th>Company</th>
<th>Unit</th>
<th>Price</th>
<th>$/kW</th>
<th>Status/Notes in Summer-2008</th>
</tr>
</thead>
<tbody>
<tr>
<td>Energy Concepts</td>
<td>400kW absorption cycle</td>
<td>No price quote</td>
<td>N/A</td>
<td>Not clear if a specific unit is available or if their product would be custom designed per application.</td>
</tr>
<tr>
<td>Rexorce</td>
<td>250kW Thermal Engine</td>
<td>No price quote</td>
<td>$1,500</td>
<td>Modified Kalina Cycle. Expect to be completed in a few months.</td>
</tr>
</tbody>
</table>

#### Sterling engine systems

<table>
<thead>
<tr>
<th>Company</th>
<th>Unit</th>
<th>Price</th>
<th>$/kW</th>
<th>Status/Notes in Summer-2008</th>
</tr>
</thead>
<tbody>
<tr>
<td>ReGen Power</td>
<td>500kW</td>
<td>No price quote</td>
<td>$1000 - $2000</td>
<td>3 months into 18 months development cycle. Hope to have 10kW prototype at end of 2008. Also hope to have 250kW and 1MW units in future.</td>
</tr>
</tbody>
</table>

### Table 2. Various hot water and cold water flow rates at which power unit was tested

<table>
<thead>
<tr>
<th>Hot water temperatures, °C (°F)</th>
<th>Hot water flow rate, m³/hr (gpm)</th>
<th>Cold water temperatures, °C (°F)</th>
<th>Cold water flow rate, m³/hr (gpm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>68.33 (155)</td>
<td>27.25 (120)</td>
<td>10 (50)</td>
<td>27.25 (120)</td>
</tr>
<tr>
<td>79.44 (175)</td>
<td>36.34 (160)</td>
<td>20 (68)</td>
<td>36.34 (160)</td>
</tr>
<tr>
<td>90.56 (195)</td>
<td>45.4 (200)</td>
<td></td>
<td>45.4 (200)</td>
</tr>
<tr>
<td>101.67 (215)</td>
<td>56.8 (250)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>107.22 (225)</td>
<td>68.1 (300)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Table 3. Total component cost incurred on building the experimental system

<table>
<thead>
<tr>
<th>Component</th>
<th>Cost ($)</th>
</tr>
</thead>
<tbody>
<tr>
<td>50kW ORC Power unit</td>
<td>119,388.00</td>
</tr>
<tr>
<td>Steam loop</td>
<td>8,997.22</td>
</tr>
<tr>
<td>Hot water loop</td>
<td>16,762.03</td>
</tr>
<tr>
<td>Cold water loop</td>
<td>14,613.57</td>
</tr>
<tr>
<td>Electrical system</td>
<td>3567.91</td>
</tr>
<tr>
<td>Instrumentation</td>
<td>21,246.25</td>
</tr>
<tr>
<td>Structural material</td>
<td>5,409.22</td>
</tr>
<tr>
<td>Miscellaneous parts and other costs</td>
<td>948.38</td>
</tr>
<tr>
<td><strong>Total component cost</strong></td>
<td><strong>190,932.58</strong></td>
</tr>
</tbody>
</table>
Table 4. TIER-4 interim EPA emissions standards for non-road diesel engines [20, 21]

<table>
<thead>
<tr>
<th>NOX (g/kWh)</th>
<th>Particulate matter (PM) (g/kWh)</th>
<th>CO (g/kWh)</th>
<th>HC (g/kWh)</th>
<th>CO2 (kg/lit)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.5 (0.0077161)</td>
<td>0.10 (0.0002204)</td>
<td>3.5 (0.0077161)</td>
<td>0.40 (0.0008818)</td>
<td>2.66 (22.2)</td>
</tr>
</tbody>
</table>

Table 5. Diesel engine specifications

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diesel Engine</td>
<td>Caterpillar C175-16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Brake power</td>
<td>1.9MW (2588BHP)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>16</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Compression ratio</td>
<td>16.7</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Speed</td>
<td>1200rpm</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jacket water temp.</td>
<td>99°C (210.2°F)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Jacket water flow rate</td>
<td>120 m³/h (528gpm)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>Aspiration</td>
<td>Turbocharged (no EGR)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6. Diesel engine specifications at various loads

<table>
<thead>
<tr>
<th>Percent load</th>
<th>Brake power, hp (MW)</th>
<th>Brake power consumption, lit/s (gpm)</th>
<th>Heat rejection to jacket water, kW</th>
<th>Exhaust temperature, °C</th>
<th>Heat present in exhaust at 176.6°C (350°F), kW</th>
</tr>
</thead>
<tbody>
<tr>
<td>100</td>
<td>2588 (1.9)</td>
<td>0.131 (2.08)</td>
<td>1009.3</td>
<td>417.8</td>
<td>869.9</td>
</tr>
<tr>
<td>75</td>
<td>1941 (1.4)</td>
<td>0.101 (1.61)</td>
<td>753.9</td>
<td>404.1</td>
<td>686.7</td>
</tr>
<tr>
<td>50</td>
<td>1294 (1.0)</td>
<td>0.070 (1.12)</td>
<td>510.4</td>
<td>398.6</td>
<td>478.3</td>
</tr>
<tr>
<td>25</td>
<td>647 (0.5)</td>
<td>0.040 (0.63)</td>
<td>294.1</td>
<td>340.4</td>
<td>231.3</td>
</tr>
<tr>
<td>65.7</td>
<td>1700 (1.3)</td>
<td>0.090 (1.42)</td>
<td>663.2</td>
<td>402</td>
<td>609.1</td>
</tr>
</tbody>
</table>

Table 7. Estimated ORC performance for operating on waste heat recovery from diesel engine

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Jacket water heat only</th>
<th>Jacket water + Exhaust heat</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot water supply temperature to ORC power unit</td>
<td>99°C (210.2°F)</td>
<td>107.2°C (225°F)</td>
</tr>
<tr>
<td>Hot water flow rate to ORC power unit</td>
<td>45.4m³/h (200gpm)</td>
<td>45.4m³/h (200gpm)</td>
</tr>
<tr>
<td>Heat input to evaporator of ORC power unit</td>
<td>576.5kW</td>
<td>617.7kW</td>
</tr>
<tr>
<td>System operating power output</td>
<td>41.7kW (355.7MWh/year)</td>
<td>45.7kW (390MWh/year)</td>
</tr>
<tr>
<td>System operating efficiency</td>
<td>7.2%</td>
<td>7.4%</td>
</tr>
<tr>
<td>Diesel fuel saved</td>
<td>96190.4lit/year (25410.8gal/year)</td>
<td>105382.5lit/year (27840gal/year)</td>
</tr>
<tr>
<td>Dollar amount saved on diesel fuel</td>
<td>$127060/year</td>
<td>$139200/year</td>
</tr>
<tr>
<td>Payback period @ 0% interest</td>
<td>2.2 years</td>
<td>2 years</td>
</tr>
<tr>
<td>Payback period @ 10% interest</td>
<td>2.6 years</td>
<td>2.3 years</td>
</tr>
<tr>
<td>Reductions in CO2 emissions</td>
<td>282short-tons/year</td>
<td>309short-tons/year</td>
</tr>
<tr>
<td>Reductions in NOX emissions</td>
<td>1245kg/year</td>
<td>1364kg/year</td>
</tr>
<tr>
<td>Reductions in HC emissions</td>
<td>142.3kg/year</td>
<td>156kg/year</td>
</tr>
<tr>
<td>Reductions in CO emissions</td>
<td>1245kg/year</td>
<td>1364kg/year</td>
</tr>
<tr>
<td>Reductions in PM emissions</td>
<td>35.5kg/year</td>
<td>39kg/year</td>
</tr>
</tbody>
</table>