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Design of a Testing Setup for an Organic Rankine Cycle System for Rural Alaska Diesel Power Plant Applications

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Abstract

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 diesel generator sets (gensets) may not be appropriate for applying heat recovery for the purposes other than electrical power generation. In addition, many other villages that do not have enough public buildings to utilize all the waste heat may also choose to apply the extra heat energy for electrical power generation. Due to the varying sizes and electrical loads of most of the diesel gensets (from 100kw to 1MW); small sized heat recovery power systems (100kW or less) are preferred instead of industrial scale systems. A 50kW Organic Rankine Cycle (ORC) was selected for this study. The selection was based on availability and cost. The purpose of this work is to design an experimental setup and testing procedure for collecting reliability and performance data of the ORC system, which are then usedto estimate the potential economic impact ofthe systemon most of rural Alaskan diesel power plants. The paper presentsdetails of the design and selection processes for input and control parameters of the testing system, experimental setup outline, test results,and estimatedpotential economic impact of the system on individual Alaskan village diesel power plants.

During the reliability test, the power unit was run for 600hrs at the full load. During performance testing, the power unit was tested for different hot water supply temperatures corresponding to the generally recognized Jacket water temperatures of diesel generators at different load conditions and different cooling water temperatures corresponding to the range of ground water temperatures.For hot water temperature of 90.56ºC (i.e. 195ºF) and cooling water temperature of 10ºC (i.e. 50ºF), the power unit consistently generated system operating power output and efficiency of 32.3kW and 7.0% respectively. Based on the cost spent on this experiment, the corresponding payback periodwas estimated of less than 3.5years with 10% interest on capital. Based on the annual power generation obtainable from the ORC system, the corresponding estimated emissions reductions were 975kg/year, 112kg/year, 975kg/year, 28kg/year and 220short-tons/year of NO_X, HC, CO, PM and CO_2 respectively.

Keywords: Organic Rankine Cycle (ORC), low grade heat, isentropic efficiency of screw expander, system operation efficiency, payback period, reliability test, and performance test.

1. Introduction and Literature review

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.

Considering the jacket liquid heat recovery, in rural Alaska this is well-established and only about 50% of rural villages in Alaska are equipped with jacket liquid heat recovery systems for heating. Among them many are facilitated with combined jacket water and charge air heat recovery systems for heating purposes.

Considering 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. However, the fuel price surge in recent years has led to re-evaluation of exhaust heat recovery for village fuel energy efficiency improvement.

Applications of recovered heat from diesel engines may include desalination, refrigeration, space heating, and power generation [1-12]. The selection of the most desirable application, one with good potential for many years, is based on need, availability, feasibility, and benefit for rural Alaskans.Considering the Alaska's geographical location and the diesel consumption there,a reliable and economical heat recovery application for Alaska villages may be heating. In general, about 50% or more of heat present in exhaust may be recoverable for heating. The same research team at UAF Energy Center has successfully demonstrated the experimental and economic analysis of exhaust heat recovery for heating [1, 8, 9].

Due to the infrastructure, economic impact and needs of the villages, many of village diesel gensets may not be appropriate for applying heat recovery for the purposes other than electrical power generation. In addition, many other villages that do not have enough public buildings to utilize all the waste heat may choose to apply the extra heat energy for electrical power generation. Based on feasibility, reliability, and cost considerations, organic Rankine cycle (ORC) and Ammonia/water vapor absorption power cycle are promising candidates.Following few paragraphs give the literature search for these two thermodynamic cycles and the reason for selecting this particular 50kW ORCfor testing is also explained.

1.2 Diesel engine waste heat to power conversion

Diesel engine waste heat for power conversions using the thermodynamic cycles (Organic Rankine Cycle or ammonia-water vapor absorption power cycle (e.g. Kalina Cycle)) is a promising possibility for increasing the efficiency of Alaska village power plants. ORC has been implemented at industry level for waste heat recovery, and 8% to 18% recovery of waste heat has been achieved (table-2). This is a considerable amount. The thermodynamic efficiency of the ammonia-water absorption cycle for waste heat recovery was reported to be 11% to 26% (table-4).Table-1 to table-4lists the literature review (both analytical and physical system) for Organic Rankine cycle and ammonia-water vapor absorption power cycle [13-36]. The selection of the 50kW ORC was based on the suitable heat energy requirement, availability of physical system, and cost consideration. According to literature search of the authors, there was no ammonia/water commercial unit available for low grade heat recovery. Research data for more complex thermal power cycles have also been searched [49-52]. Similar to the reasons as for the ammonia/water system and the potential high cost, literature search for complex power cycles were not further conducted.

This paper discusses the design of experimental setup for testing a 50kW ORC power unit for data collection so that the data could be used for performance prediction and economic impact of installing this system on individual rural Alaska diesel power plant. This paper includes reliability and system performance test results conducted on ORC power unit. The experimental setup includes the heat source loop, heat sink loop, electrical system and instrumentation. The experimental setup also includes the component selection process.Finally we present the experimental results from reliability test and performance test for screw expander andsystem operating power outputs, screw expander and system operating efficiencies, effect of parasitic power consumption, diesel fuel saved, emissions reductions, and payback period estimation.

Table-1 Literature review for analytically analyzed ORC power systems

Literature Citation	Heat source	Heat source Temperature	Heat Sink Temperature	Working fluid	Power output	Efficiency
[13]	Engine exhaust and jacket water from turbocharged, Mercedes-Benz OM422A, 243 kW, direct injection,	Jacket water at 88°C and Exhaust gas at 470°C	Ambient temperature at 30°C for R-12 and Water cooled condenser for "Water"	R-12 and water	38.5kW (R-12); 37.2kW (water)	16% (R-12) and 15.3 (water) boost in engine efficiency

	water-cooled 6-cylinder, Vee engine		as working fluid			
[14]	Biomass, industrial waste and solar thermal	90°C to 110°C (at inlet to expander)	25°C	R134a, Water, R227ea, R245fa, Toluene, Iso-Butane, Acetone, Iso-Pentane, n-pentane, Dimethylether	9.9W (Toluene) to 282W (Dimethyl ether)	2.04% (Water) to 2.86% (Iso-butane)
[15]	275kW turbo-charged heavy duty diesel engine	95°C (Jacket water); 184°C (Charge air cooler); 688°C (EGR)	20°C	A dry fluid with critical pressure <65bar (No specific fluid)	55kW	18% thermal efficiency
[16]	Engine waste heat	260°C	9°C (air cooled)	Isobutane and Isopentane	1650kW to 1750kW	7% to 9% thermal efficiency
[17]	Geothermal sources	90°C	30°C (Water)	Ammonia, R123, n-pentane, PF5050	10MW	5.7% to 7.4% thermal efficiency
[18]	2 X 8.9MW diesel engine exhaust	346°C exhaust gases	25°C Water	Hexamethyldisiloxane	1603kW	17.3%

Table-2 Literature review for installed or working prototype ORC power systems

Literature Citation	**Heat source**	**Heat source Temperature**	**Heat Sink Temperature**	**Working fluid**	**Power output**	**Efficiency**
[19]	20 kW electric heaters (2 X 10 kW)	45°C to 70°C (HCFC-123) and 115°C to 125°C (water) (measurement at inlet to turbine)	Not available	HCFC-123 and water	20W to 150W (HCFC-123) and 60W to 150W (water)	0.2% to 1.25% (HCFC-123) and 0.36% to 0.75% (water) (heat conversion efficiency)
[20]	Hot air source	172.6°C to 182.3°C	13.2°C to 15°C	HCFC-123	0.67kW to 1.03kW	4% to 6.1% cycle efficiency

[21]	Hot water heater	100°C	26.67°F (Water)	HFE-7000	937W	3.1% thermal efficiency
[22]	Hot water from a geothermal well	73.33°C	4.44°C	R-134a	210 kW	8.2% heat conversion efficiency
[23]	Exhaust from a 27MW gas turbine driving a natural gas pipeline compressor	480°C	–18°C to 35°C (Air cooled condenser)	Pentane	5.5MW	15% heat conversion efficiency
[24]	Hot water from a geothermal well	106°C	10°C	Fluorocarbons	500kW	8.13% heat conversion efficiency
[25]	Solar energy	300°C	Not available	n-pentane	1.0MW	12.1% design point efficiency
[26]	Hot water from a geothermal well	98°C	25°C	Isopentane	120kW	6.0% heat conversion efficiency
[27]	Exhaust gas from cryogenic gas turbine	260°C (Hot oil temperature)	21.11°C (Air cooled)	n-pentane	3.05MW	13% thermal efficiency
[28]	Biomass CHP plant	250°C to 300°C (thermal oil temperature)	60°C	Not available	1.1MW	18% thermal efficiency
[29]	Biomass CHP plant	300°C (thermal oil temperature)	60°C (Water)	Silicon oil	1.0MW	18% thermal efficiency
[30]	7MW - 8MW stationary diesel engine	240°C to 310°C (thermal oil temperature)	25°C	Not available	0.6MW	16% to 20% thermal efficiency

Table-3 Literature review for analytically analyzed ammonia/water vapor absorption power systems

Literature Citation	Heat source	Heat source Temperature	Heat Sink Temperature	Power output	Efficiency
[31]	Low temperature heat source	90°C	30°C	Not available	2.3% to 10%
[32]	Hybrid system for power and refrigeration	Dry Saturated steam at 0.69MPa (164.37°C)	32°C	115kW	58% Coefficient of performance
[18]	2 X 8.9MW diesel engine exhaust	346°C exhaust gases	25°C Water	1615kW	17.5%

Table-4 Literature review for installed or working prototype ammonia/water vapor absorption power systems

Literature Citation	Heat source	Heat source Temperature	Heat Sink Temperature	Power output	Efficiency
[33]	Solar Centaur 50 natural gas turbine	475°C to 520°C	11°C Water	1.6MW to 2.7MW	22.5% to 26.5%
[34]	Hot water from geothermal well	125°C	4°C	2MW	11.8%
[35]	Hot water from geothermal well	120°C	Not available	3.4MW	11% thermal efficiency (Calculated)
[36]	Hot water from geothermal well	100°C	Not available	50kW	Not available

2. Purpose of System design and Experimental setup

In Alaska the diesel power plantsaverage load ranges from 100kW to 4.5MW [Alaska Statistics]. The available engine waste heat range from 52.5kW to 2.4MW in jacket liquid and 79kW to 3.5MW in exhaust. Considering 80% heat recovery from jacket liquid, 42kW to 1.9MW of heat can be recovered.Considering 50% heat recovery from exhaust, 40kW to 1.8MW of heat energy can be recovered.

Considering the characteristics of this 50kW ORC power unit, table-5 gives the limitations on heat source, heat sink, power generation etc., provided by the power unit manufacturer. With estimated heat-to-power conversion efficiency of 4% (for heat source such as jacket liquid) to 8% (for heat sources such as jacket liquid combined with exhaust), a minimum of 250kW heat source is needed to operate the power unit. From the Alaska electric power statistics report [45] about 18 diesel powered utilities can be used for jacket liquid heat recovery and 61 utilities could be used for engine jacket water combined with exhaust heat.

Therefore from the above discussionthe amount of available diesel generator waste heat varies widely from utility to utility due to varying engine sizes, load variation due to seasons, flow rates, and other factors.The purpose of this experiment is to collect data with input to the power unit matches the possible diesel engine waste heat condition. The ORCpower unit was tested under controlled conditions at the University of Alaska Fairbanks power plant, and data was collected as different diesel engine waste heat conditions were simulated. The goal was to estimate the performance and economic benefit of installing this power unit on different village diesel generators.In this regard for the presentdesign of experimental setup should be able to have flow rate and temperature control of both heat source and heat sink side to perform reliability and performance test on the 50kW ORC power unit. Following paragraphs give detailed description of reliability and performance test conditions.

Table-5 Characteristics of the 50kW ORC power unit

Parameter	Limitation
Heat source supply temperature	65.5°C (150°F)–107.2°C (225°F)
Heat source supply flowrate	27.2m^3/hr (120gpm) – 68.1m^3/hr (300gpm)
Heat sink supply temperature	43.3°C (110°F)
Expander power output	10kW – 50kW
Expander rotational speed	1800rpm

2.1 Reliability test

Reliability test is conducted at full load gross power output of the power unit for 600hrs to see the long term performance of the machine and the whole system. As per the power unit manufacturer specification the power unit will generate 50kW screw expander power output at hot water supply condition of 104.4°C (220°F) and 36.4m^3/hr (160gpm) and cold water supply condition of 10°C to 20°C (50°F to 68°F) and 36.4m^3/hr (160gpm).

2.2 Performance test

The power unit was also tested for performance under different hot water and cold water flow rates and temperatures, while performance was monitored both of the power unit as a whole and of the individual components (expander, evaporator, condenser, pump etc.). The different parameters are given in table-6. The total number of performance tests conducted on power unit was around 150.

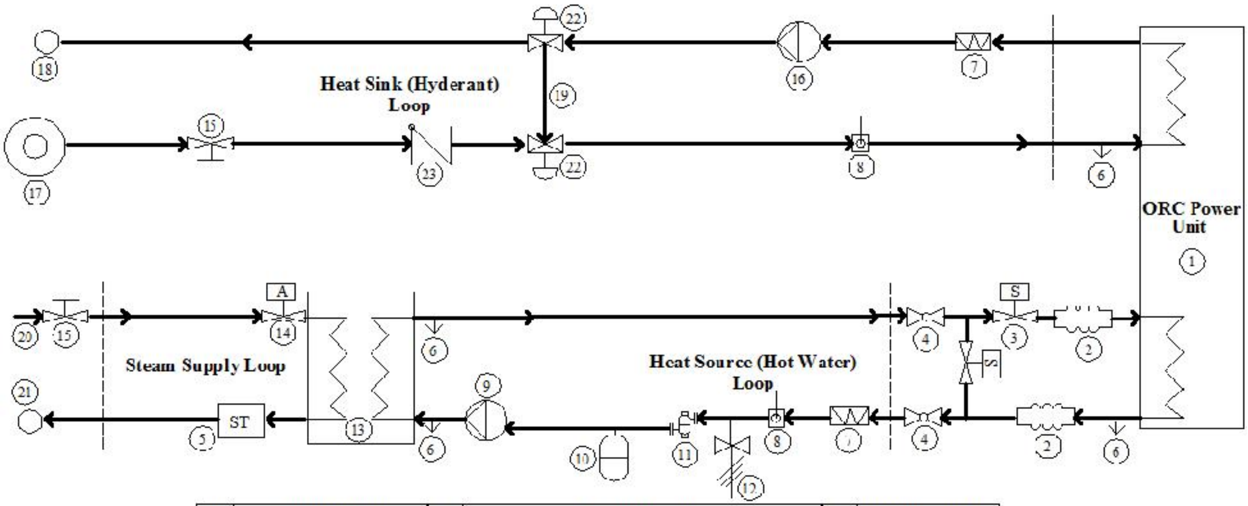
3. System design and Experimental setup

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 water. The organic fluids vaporize at a lower temperature, enabling power extraction from lower-temperature heat sources, such as a geothermal source, or in our cases of interest, waste heat from a generator.The basic components of ORC arepump, evaporator, expander, and condenser.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.

Table-6 Various hot water and cold water flow rates at which power unit will be tested

Hot water temperatures , ºC (ºF)	**Hot water flow rate, m^3/hr (gpm)**	**Cold water temperatures , ºC (ºF)**	**Cold water flow rate, m^3/hr (gpm)**
68.33 (155)	27.25 (120)	10 (50)	27.25 (120)
79.44 (175)	36.34 (160)	20 (68)	36.34 (160)
90.56 (195)	45.4 (200)		45.4 (200)
101.67 (215)	56.8 (250)		
107.2 (225)	68.1 (300)		

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) Electricalsystem (figue-2) and (iv) Instrumentation (not shown in figures).Experimental setup was located in University of Alaska Fairbanks (UAF) CHP plant. Abundant amount of low pressure steam (around 205.7kPa and 121.1ºC) readily available was used as heat source for the power unit and the cold water (around 10ºC) from near-by fire hydrant was used as heat sink. Heat in steam was transferred to water in the hot water loop through a plate heat exchanger. Heated water was controlled to desired temperatures and transferred heat to the refrigerant in the ORC system. Electrical system is the wiring required to upload power to grid and wiring various power consuming components. Instrumentation included various data collection components such as flow meters, thermocouples, electrical meters etc.Details about component designs and selections are given in Appendix I.



#	Component	#	Component	#	Component
1	ORC power unit	12	Pressure relief valve	23	Check valve
2	Expansions joints	13	Steam-to-hot water heat exchanger		
3	Solenoid valve	14	Steam control valve with actuator		
4	Ball valve	15	Manual control valve		
5	Steam trap	16	Pump (Constant flow rate)		
6	Drain	17	Hydrant source (Cooling water source)		
7	Temperature mixer	18	Cooling water from GM (Hydrant sink)		
8	Ultrasonic flow 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	3-way butterfly valve with actuator		

Figure-1 Experimental setup for testing 50kW ORC power unit

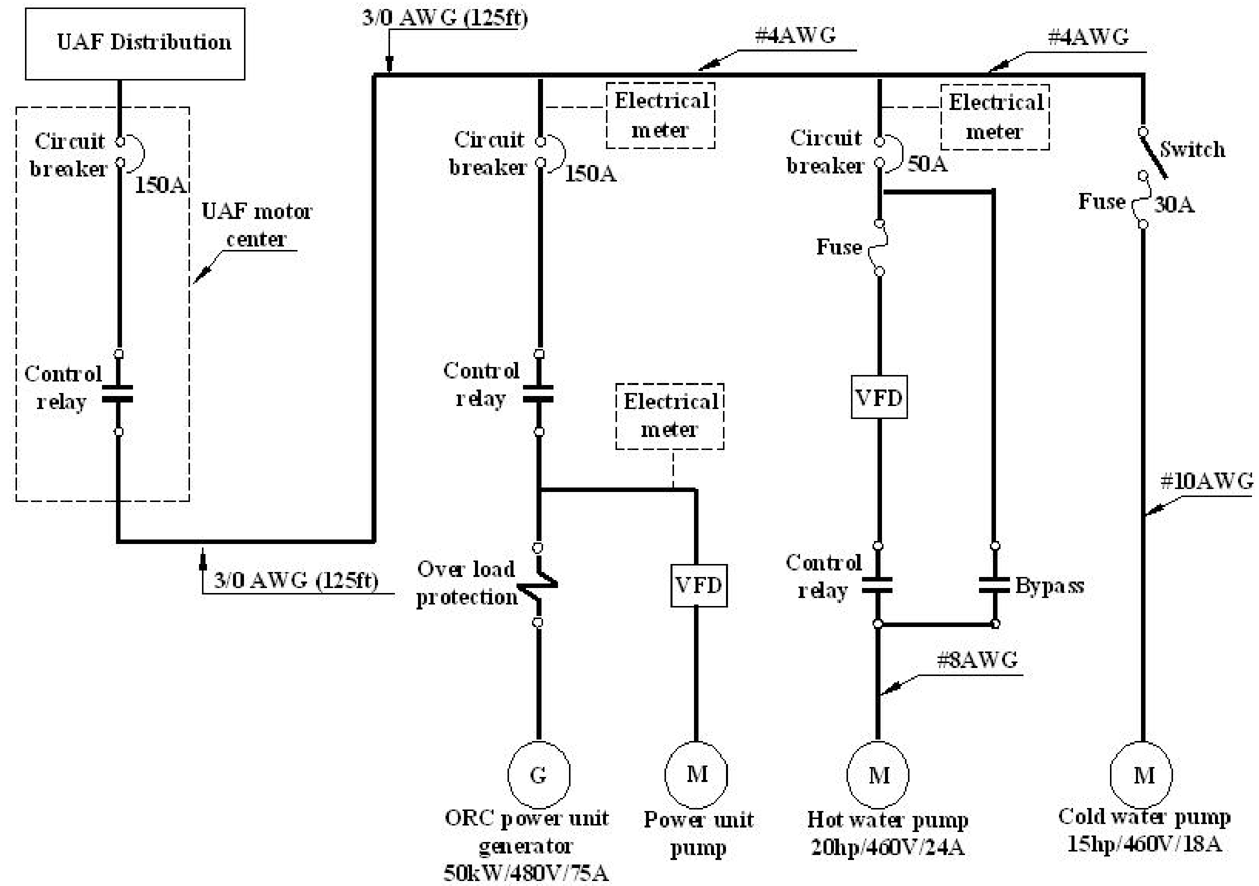


Figure-2 Line diagram for electrical wiring for uploading power to University system and powering hot water and cold water pumps

3.1Parameters 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 reliability and performance test results of 50kW ORC power unit, which include heat input, screw expander and power unit power generated, effect of parasitic power consumed, efficiencies, emissions, fuel savings, and payback period. Parameters measured are the direct measurements taken from installed measuring equipment which will be used in data reduction process for further analysis of the system.

3.1.1 Parameters measured

The various parameters measured during reliability and performance test of 50kW ORC power unit were (i) hot water flow rate, inlet and outlet temperatures to power unit (V_H, $T_{H,i,P}$, $T_{H,o,P}$), (ii)cold water flow rate, inlet and outlet temperatures to power unit (V_C, $T_{C,i,P}$, $T_{C,o,P}$), (iii) electrical power output of power unit (P_N), (iv) electrical power consumed by power unit pump ($P_{P_1,P}$), (v) hot water pump power ($P_{P_1,H}$), and (vi) cold water pump power ($P_{P_1,C}$). Here it should be noted that electrical power output of power unit (P_N) 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.

3.1.2 Instrumentation and data collection

For flow rate, as shown in figure-1, KamstrapUltraflow® 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-353EDM 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 powermeasurement 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 virtualinstrument 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 SCIX-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.

4. Experimental procedure

Experimental procedure is the operation procedure followed to operate the power unit for reliability test and each case of performance test.

4.1 Experimental procedure for reliability test

Reliability test is conducted on ORC power unit at full load (i.e. 50kW power output) for 600hrs to know the long term performance of the machine.As stated in section-2.1,for full load operation the hot water supply condition is 104.4ºC (220ºF) and 36.4m^3/hr (160gpm) and cold water supply condition is 10ºC (50ºF) and 36.4m^3/hr (160gpm).

4.2 Experimental procedure for performance test

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 bellow.

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 50ºF.
2. At this cold water flow rate we than set the ORC hot water supply to the desired temperature 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 27.2m^3/hr (120gpm)and 55Hz to 68.1m^3/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.

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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-6.

During steady state data collection (Step-5 above) for 30minutes, the hot water and cold water temperatures are stored by LabVIEW in excel format at frequency of 1sec. 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 30sec. 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.

5. 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 CO_2 and the economic impact of installing an ORC power unit on village diesel power plant.

Heat supplied ($Q_{H,S}$) by hot water to evaporator of power unit is obtained by,

$$Q_{H,S} = V_H \times \rho_H \times (h_{H,in,P} - h_{H,o,P}) \quad (1)$$

Here density of hot water (ρ_H), inlet enthalpy ($h_{H,in,P}$)and outlet enthalpy ($h_{H,o,P}$) of hot water to power unit were obtained based on evaporator hot water inlet and outlet temperatures and using NIST REFPROP 8.0 [44] program.ρ_H is the average density of hot water obtained at inlet and outlet evaporator hot water temperatures.

Heat rejected ($Q_{C,R}$) to cold water by condenser of power unit is obtained by,

$$Q_{C,R} = V_C \times \rho_C \times (h_{C,o,P} - h_{C,in,P}) \quad (2)$$

Here density of cold water (ρ_C), inlet enthalpy ($h_{C,in,P}$)and outlet enthalpy ($h_{C,o,P}$) of cold water to power unit were obtained based oncondenser cold water inlet and outlet temperatures and using NIST REFPROP 8.0 [44] program. ρ_C is the average density of cold water obtained at inlet and outlet condenser cold water temperatures.

Screw expander power output (P_S), which is given by Eq.3, is the power generated by power unit expander without considering any pump power consumption (i.e. without considering power unit pump, hot water and cold water pumps). Whereassystem operating power output (P_O) is the power generated by power unit which was uploaded to university power system, given by Eq.4,which considers the power unit pump and cold water pump powers. Here in calculating system operating power output (P_O), the ORC power unit pump 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 section-1 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_O will be used in annual diesel fuel saved, emissions reductions and economic outcome calculations discussed in following paragraphs. Here both P_N and $P_{P1,P}$ are measured parameters explained in section-3.4.1 above.

$$P_S = P_N + P_{P1,P} \quad (3)$$

$$P_O = P_N - P_{P1,C} \quad (4)$$

Screw expander efficiency (η_S),power unit efficiency (η_N) andsystem operating efficiency (η_O) are estimated using Eq.5, Eq.6 and Eq.7 respectively,

$$\eta_S = \frac{P_S}{Q_H \quad ,S} \qquad (5)$$

$$\eta_N = \frac{P_N}{Q_H \quad ,S} \qquad (6)$$

$$\eta_O = \frac{P_O}{Q_H \quad ,S} \qquad (7)$$

Liters (or gallons) of diesel fuel saved per year($F_{S/Y}$) was calculated using Eq.(8) which was based onsystem operating power output (P_O), 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 3.7kWh/lit (14kWh/gal) [45, 46] is a reasonable value for rural Alaska village diesel gensets. Dollar amount saved on diesel fuel per year ($F_{\$/Y}$) 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.

$$F_{S/Y} = \frac{P_O \;\times 3 \quad \times 2}{3.7} \qquad (8)$$

5.1 Reductions in emissions and CO_2

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 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-7. Annual CO_2 reductions were based on Liters (or gallons) of diesel fuel saved per year ($F_{S/Y}$). Table-7 gives the TIER-4 interim emissions standards set by EPA for non-road diesel enginegensets [47, 48].

Table-7 TIER-4 interim EPA emissions standards for non-road diesel engines [47, 48]

NO_X g/kWh (lb/kWh)	**Particulate matter (PM) g/kWh (lb/kWh)**	**CO g/kWh (lb/kWh)**	**HC g/kWh (lb/kWh)**	**CO_2 kg/lit (lb/gal)**
3.5 (0.0077161)	0.10 (0.0002204)	3.5 (0.0077161)	0.40 (0.0008818)	2.66 (22.2)

5.2 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 costbased on annual cost savings.Here annual cost savings is the dollar amount saved on diesel fuel per year in operating the ORC power unit on recovered waste heat from a rural Alaska diesel engine power plant.

The total initial investment cost can be further divided into component costs and installation costs. Component costs are the material and instrumentation cost incurred on building the whole heat recovery system and data acquisition system. For the present case thecomponent costs 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), fright charges etc.Table-8 gives the categorized component cost incurred on building the present experimental system and component cost for the present experimental setup was estimatedto be $191,000.

The installation costmay 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 30days 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 the whole system comes to $89,000 and this value is used in the present payback period calculations.

Therefore the total initial investment cost is estimated at $280,000, which is the sum of component costs and installation costs.

Table-8 Total component cost incurred on building the experimental system

Component	**Cost ($)**
50kW ORC Power unit	119,388.00
Steam loop	8,997.22
Hot water loop	16,762.03
Cold water loop	14,613.57
Electrical system	3567.91
Instrumentation	21,246.25
Structural material	5,409.22
Miscellaneous parts and other costs	948.38
Total component cost	**190,932.58**

6. Experimental results and discussion

6.1 Reliability test results

Reliability test was completed on the ORC power unit at full load (i.e. 50kW gross power output) for 600hrs to know the long-term endurance and performance of the unit.Figure-3, figure-4 and table-9 give the results of measured and derived parameters from the test. Figure-3 and figure-4 give plots for some measured parameters results for more than 40hrs of reliability test. As shown in table-9 the average hot water supply temperature was 104.2°C and flow rate of 36.28m^3/hr. Similarly the average cold water conditions were 9.7°C and 37.15m^3/hr. Table-9 also gives the reliability test results for emissions reduction, CO_2 reductions (a greenhouse gas) and payback period for operating the ORC power unit at full load for 355days a year (with 10days for maintenance).

6.1.1 Discussion based on 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 shutdowns etc., during the reliability test.
2. No major problems were observed with the steam loop, heating loop, cooling loop, and instrumentation i.e. we were able to keep stable hot water and cold water flow rates and temperatures to power unit. Occasional hot water temperature changed due to the surge in power plant steam supply conditions i.e. change in steam supply pressure (note that the steam is a saturated steam).

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3. During reliability test the average electrical power output (P_N) by power unit was obtained to be 47.8kW with power unit efficiency (η_N) of 7.8% and system operating efficiency (η_O) of 7.5%. Here in calculating η_O, the cold water pump power consumed was taken to be 1.76kW same as hot water pump power consumption.
4. As noted from table-9, ORC power unit has a maximum heat input limit of around 610kW, if excess heat is available (i.e. more than 650kW) than the extra heat can be used for other purposes such as building heating etc., or a optimization process based on villages diesel power plant economics may be needed. In other words, the data may provide the rural Alaska diesel power plant personnel with information on how to distribute the waste heat to optimize the benefit from waste heat.
5. ORC power unit achieved screw expander efficiency (η_S) of 8.4% at full load operation was well within the manufacturer's claim of 8.5%.
6. Payback period of 2years and 2.3years was obtained with 0% and 10% interest rate on capital respectively.

Table-9 Reliability test results

Parameter	**Value**
Average hot water supply temperature to power unit ($T_{H,in,P}$)	104.2^{o}C (219.7^{o}F)
Average hot water flow rate to power unit (V_H)	36.28m^3/hr (159.8gpm)
Average cold water supply temperature to power unit ($T_{C,in,P}$)	9.7^{o}C (49.4^{o}F)
Average cold water flow rate to power unit (V_C)	37.15m^3/hr (163.6gpm)
Power unit electrical power output (P_N)	47.8kW
Power unit pump power consumption ($P_{P_1,P}$)	3.61kW
Hot water pump power consumption ($P_{P_1,H}$)	1.76kW
Cold water pump power consumption ($P_{P_1,C}$)	1.76kW
System operating power output (P_O)	46.04kW
Heat supply by hot water to power unit evaporator ($Q_{H,S}$)	610.4kW
Screw expander efficiency (η_S)	8.4%
Power unit efficiency (η_N)	7.8%
System operating efficiency (η_O)	7.5%
Diesel fuel saved per year ($F_{S/Y}$)	106,060.2Lit (28,018.12gal)
Dollar amount saved on diesel fuel per year ($F_{\$/Y}$)	$140,090.6/year
Emissions reductions	
Oxides of nitrogen (NO_X)	1372.8kg/year (3026.7lb/year)
Hydrocarbons (HC)	156.9kg/year (345.9lb/year)
Particulate matter (PM)	39.2kg/year (86.5lb/year)
Carbon monoxide (CO)	1372.8kg/year (3026.7lb/year)
Carbon dioxide (CO_2)	282135.5kg/year (311tons/year)
Payback period	
Payback period @ 0% interest on capital	2years
Payback period @ 10% interest on capital	2.3years

6.2 Performance test results at hot water supply temperature of 90.56°C

Table-10 and figure-5 to figure-11 give the measured and derived performance test results for hot water supply temperature of 90.56°C. Table-10 gives the measured average values of hot water and cold water supply temperatures and flow rates and also the power unit electrical power output (P_N), electrical power consumption by power unit pump (P_{P_1} $_{,P}$) and hot water pump

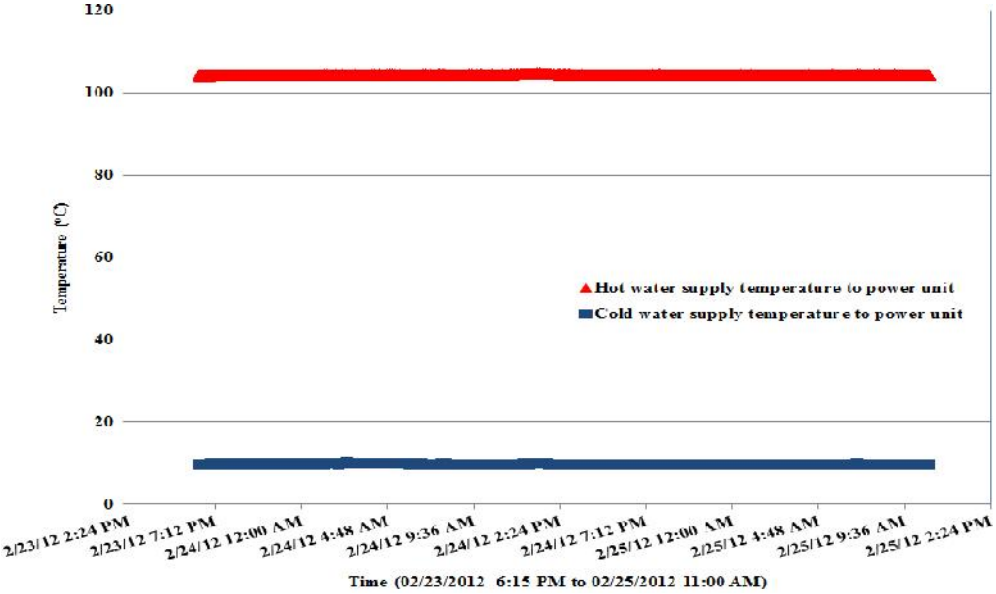

Figure-3 Hot water and cold water supply temperatures to ORC power unit during reliability test

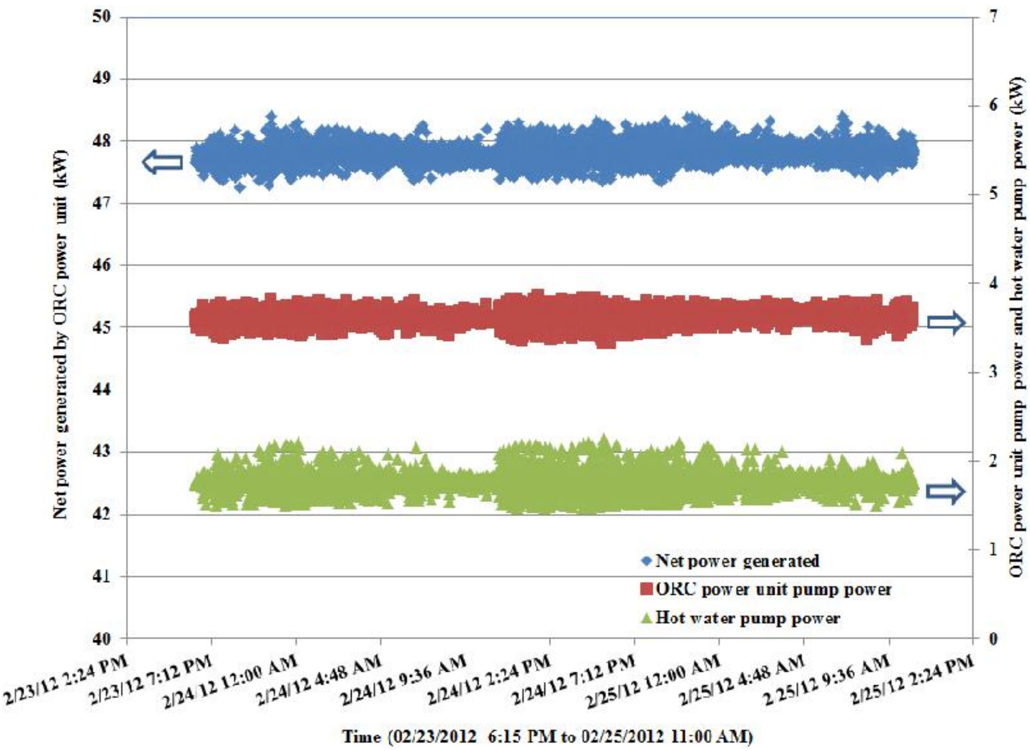

Figure-4 Net power generated, ORC system pump power and hot water pump power

($P_{P_1,H}$). The average values are the average obtained from 30min sampled data after the system reached steady state condition as discussed in section-4.2.Temperatures were sampled at frequency of 1sec, electrical power data was sampled at frequency of 30sec, and flow rate data was noted manually from flow meter display screen.Here it should be noted that the ORC power unit was tested for 5 different hot water flow rates and 3 different cold water flow rates which accounted for 15 cases (table-10) of performance testing, maintaining hot water and cold water supply temperatures of 90.56ºC and 10ºC (approx.) respectively. Figure-5 gives the heat supplied by hot water ($Q_{H,S}$) to power unit evaporator. Figure-6 gives the screw expander (η_S) and power unit (η_N) efficiencies of the power unit. Figure-7 shows the system operating power output (P_O) and system operating efficiency (η_O) for various hot water and cold water flow rates. Figure-8 gives the diesel fuel saved and dollar amount saved on fuel. Figure-9 and figure-10 give the emissions reductions. Figure-11 shows the payback period estimations based on 0% and 10% interest on total initial investment cost.

6.2.1 Discussion on performance test results

Heat supplied by hot water ($Q_{H,S}$) to power unit evaporator increased with the increase of hot water flow rate as well as with the increase of cold water flow rate as shown in figure-5. Heat supplied by hot water increased from 412.84kW at 27.17m^3/hr and 27.11m^3/hr (Case#1 in table-10) of hot and cold water flow rates respectively to 518.55kW at 68.43m^3/hr and 46.76m^3/hr of hot and cold water flow rates respectively. The irregular nature of increase in$Q_{H,S}$ for cold water flow rate of 36.34m^3/hr (orange curve in figure-5) is due tooccasional disturbancein hot water supply temperature from 91.03ºC at 35.88m^3/hr to 90.38ºC at 45.85m^3/hr (Case#7 and Case#8) resulted from the surge in power plant steam supply condition.

Power unit electrical power output(P_N) and power unit pump power consumed ($P_{P_1,P}$)also increased with the increase of hot water flow rate as well as with the increase of cold water flow rate as given in table-10. Net power generated by power unit increased from 29.91kW at 27.17m^3/hr and 27.11m^3/hr (Case#1) of hot and cold water flow rates respectively to 38.11kW at 68.43m^3/hr and 46.76m^3/hr (Case#15) of hot and cold water flow rates respectively. The increase in P_N and $P_{P_1,P}$ with increase of hot water and cold water flow rates can be directly related to increase in heat supplied by hot water and cooling effect provided by cold water.

Power unit showed an increase in screw expander efficiency (η_S) (figure-6) with increase of cold water flow rate for a given hot water flow rate. For example from figure-6, at hot water flow rate of 27.2m^3/hr, η_S increased from 7.71% to 8.07% for cold water flow rate increase from 27.11m^3/hr to 46.35m^3/hr, which can be attributed to the increase in cooling effect provided by cold water.

For a given cold water flow rate, screw expander efficiency (η_S) showed an increasing trend until 36.4m^3/hr of hot water flow then decreasing with increase of hot water flow rate until 68.13m^3/hr (300gpm). For example from figure-6, for cold water flow rate of 45.4m^3/hr, η_S increased from 8.07% to 8.11% for hot water flow rate increase from 27.77m^3/hr to 36.49m^3/hr and then decreased to 7.85% for hot water flow rate increase to 68.43m^3/hr. This behavior of η_S can be attributed to the ORC power unit's performance itself. Take for instance at cold water flow rate of 45.4m^3/hr, as the hot water flow rate to power unit increased from 27.77m^3/hr to 36.49m^3/hr the heat input ($Q_{H,S}$) to the power unit increased from 430.5kW to 461.47kW (an increase of 7.2%) and at the same time P_S increased from 34.73kW to 37.44kW (an increase of 7.8%), roughly a proportional increase in P_S with increase in $Q_{H,S}$. Whereas for the same case of cold water flow rate, as the hot water flow rate to power unit increased from 36.49m^3/hr to 68.43m^3/hr, the $Q_{H,S}$ increased from 461.47kW to 518.55kW (an increase of 12.4%) and at the same time P_S increased from 37.44kW to 40.68kW (an increase of only 8.65%). The same trends are shown in figure-6 for power unit efficiency (η_N) and the same discussion above is valid to η_N as well for both hot water and cold water flow rate changes. The behavior of screw expander efficiency (η_S) for cold water flow rate of 36.4m^3/hr(orange curve in figure-6) is due to hot water supply temperature disturbance resulted from the surge in power plant steam supply condition.

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Table-10 Measured parameters from performance testof hot water supply temperature at 90.56°C (195°F)

Case #	Hot water supply temperature, $T_{H,i,P}$, °C (°F)	Hot water flow rate, V_H, m³/hr (gpm)	Cold water supply temperature, $T_{C,i,P}$, °C (°F)	Cold water flow rate, V_C, m³/hr (gpm)	Power unit power output, P_N, kW	Power unit pump power consumed, $P_{P,P}$, kW	Hot water pump power consumed, $P_{P,H}$, kW
Cold water flow rate of 27.25m³/hr (120gpm)							
1	90.42 (194.75)	27.17 (119.63)	11.49 (52.68)	27.11 (119.37)	29.91	1.93	1.00
2	90.53 (194.96)	36.58 (161.07)	11.29 (52.32)	27.16 (119.60)	32.19	2.13	1.86
3	90.59 (195.06)	45.81 (201.69)	11.33 (52.40)	26.95 (118.66)	33.44	2.27	3.13
4	90.70 (195.26)	57.14 (251.58)	11.37 (52.46)	27.04 (119.07)	34.83	2.42	5.75
5	90.76 (195.36)	68.34 (300.87)	11.37 (52.47)	27.15 (119.53)	35.43	2.49	9.57
Cold water flow rate of 36.34m³/hr (160gpm)							
6	90.95 (195.72)	27.22 (119.83)	11.05 (51.88)	35.89 (158.00)	31.16	1.91	1.01
7	91.03 (195.85)	35.88 (157.97)	11.21 (52.17)	35.91 (158.12)	34.12	2.27	1.78
8	90.38 (194.69)	45.85 (201.88)	10.86 (51.54)	37.02 (162.99)	34.86	2.29	3.16
9	91.26 (196.27)	57.08 (251.33)	11.17 (52.10)	36.23 (159.51)	36.81	2.55	5.73
10	90.72 (195.30)	68.67 (302.37)	10.97 (51.75)	37.29 (164.18)	37.40	2.60	9.70
Cold water flow rate of 45.4m³/hr (200gpm)							
11	90.15 (194.26)	27.77 (122.29)	10.16 (50.29)	46.35 (204.07)	32.69	2.04	1.03
12	90.76 (195.37)	36.49 (160.64)	10.27 (50.49)	46.37 (204.15)	35.18	2.25	1.86
13	90.95 (195.71)	46.22 (203.49)	10.40 (50.72)	46.63 (205.31)	36.69	2.44	3.21
14	91.08 (195.95)	57.07 (251.26)	10.37 (50.67)	46.48 (204.65)	37.86	2.57	5.75
15	90.75 (195.35)	68.43 (301.28)	10.37 (50.67)	46.76 (205.86)	38.11	2.57	9.64

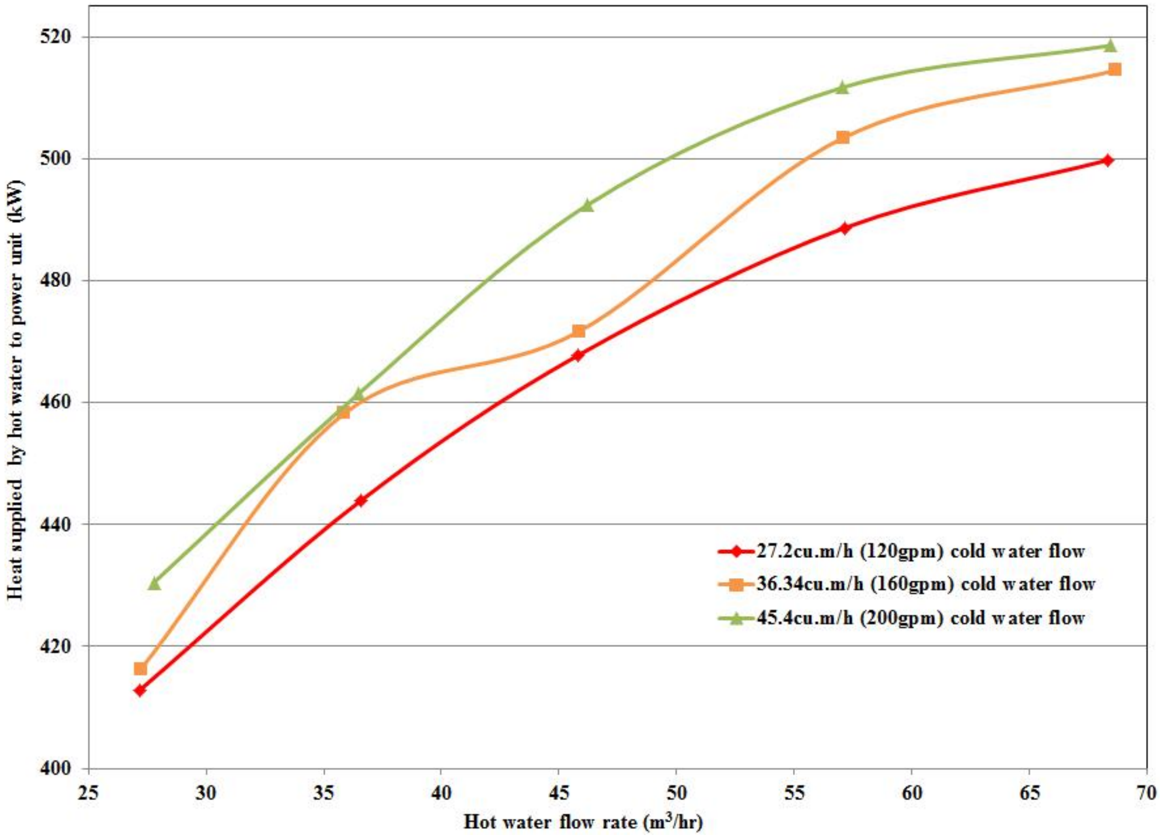

Figure-5 Heat supplied to power unit evaporator at hot water supply temperature of 90.5°C (195°F)for various hot water and cold water flow rates

For a given cold water flow rate,system operating power output (P_O) increased with the increased of hot water flow rate but system operating efficiency (η_O) showed an increasing trend until 36.4m^3/hr of hot water flow then decreasing with increase of hot water flow rate until 68.13m^3/hr (300gpm) as shown in figure-7. This behavior of η_O is similar to screw expander efficiency (η_S) which is due to power unit's performance and the same explanation is valid here.

For system operation the electrical power consumption by cold water pump was taken to be 1.00kW, 1.86kW and 3.21kW for flow rates of 27.25m^3/hr (120gpm), 36.34m^3/hr (160gpm) and 45.4m^3/hr (200gpm) respectively. For a given hot water flow rate, the system operating power output (P_O) and system operating efficiency (η_O) showed an increasing trend with cold water flow rate increase from 27.25m^3/hr (120gpm) to 36.34m^3/hr (160gpm) and then decreasing with increase of cold water flow rate to 45.4m^3/hr (200gpm). This may be explained due to the effect of parasitic power consumption of cold water pump. For example at hot water flow rate of 36.34m^3/hr (figure-7), for cold water flow rate increase from 27.25m^3/hr (120gpm) to 36.34m^3/hr (160gpm) the power unit electrical power output (P_N) increased from 32.19kW to 34.12kW whereas the cold water pump power increased from 1kW to 1.86kW resulting in 3.4% increase in system operating power output (P_O). On the other hand for cold water flow rate increase from 36.34m^3/hr (160gpm) to 45.4m^3/hr (200gpm) the power unit electrical power output (P_N) increased from 34.12kW to 35.18kW whereas the cold water pump power increased from 1.86kW to 3.21kW resulting in 0.9% net decrease in system operating power output (P_O) which is clearly due to the effect of higher cold water pump power consumption at higher flow rate of 45.4m^3/hr.

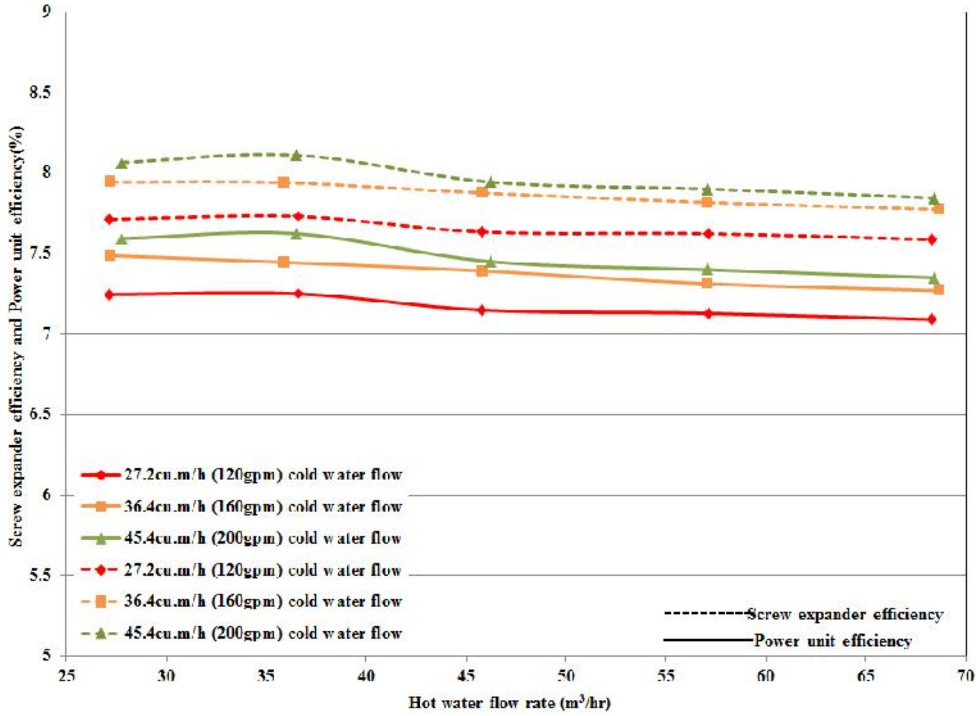

Figure-6 Screw expander efficiency and power unit efficiencyfor various hot water and cold water flow rates

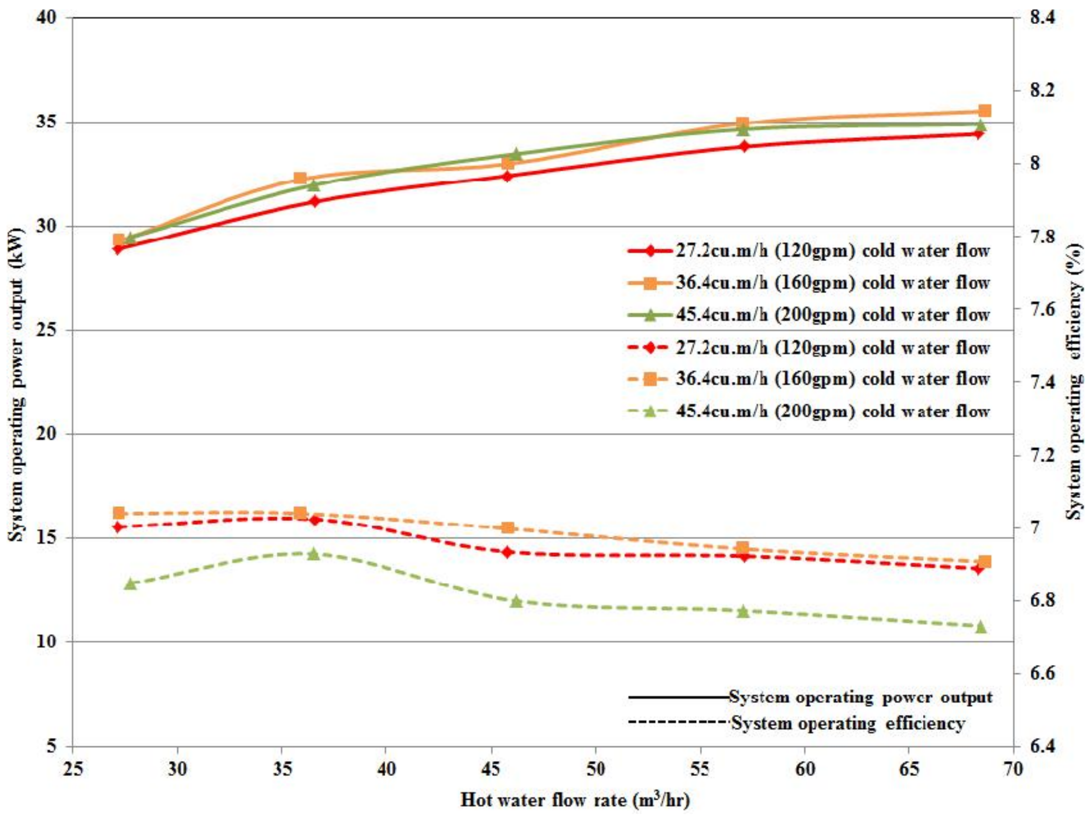

Figure-7 System operating power and efficiencyfor various hot water and cold water flow rates

Gallons of fuel saved per year ($F_{S/Y}$) and dollar amount saved per year on fuel ($F_{\$/Y}$) varied from 17,600gal/year and \$88,000/year at P_O of 28.9kW to 21,628gal/year and \$108,140/year at P_O of 35.5kW as shown in figure-8. Here diesel fuel price was assumed to be \$5/gal which is a reasonable value for rural Alaska diesel power plants. For P_O of 28.9kW to 35.5kW, NO_X and CO emissions reductions ranged between 862.2kg/year to 1,040.8kg/year (both have same TIER-4 emission criteria of 3.5g/kWh each), HC emissions reductions ranged between 98.5kg/year to 121kg/year (figure 9), PM emissions reductions ranged between 24.6kg/year to 30.3kg/year and CO_2 emissions reductions ranged between 195.3short-tons/year to 240 short-ton/year (figure 10). Here it is to be noted that CO_2 is a greenhouse gas. The calculations were based on eq.8 and table-7 given in section-5.

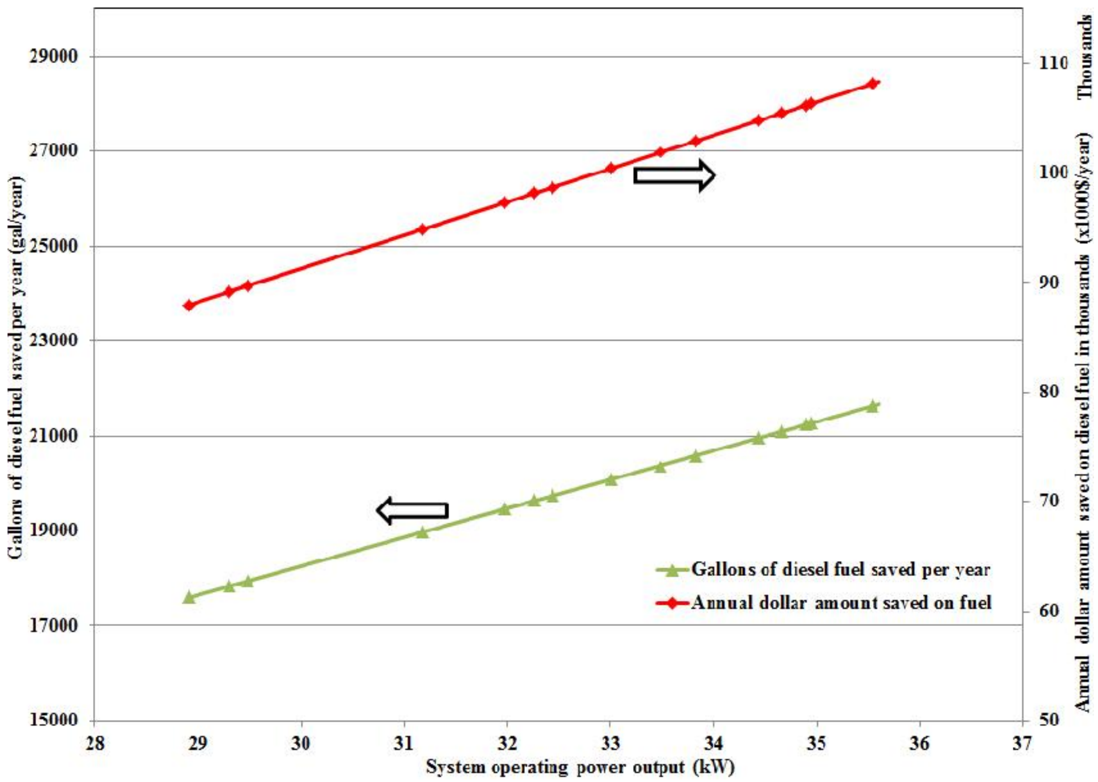

Figure-8 Gallons of diesel fuel saved per year and annual dollar amount saved on diesel fuel vs. system operating power output (P_O)

For 0% interest rate on total initial capital (\$280,000.00) the payback period ranged between 3.2years at P_O of 28.9kW to 2.6years at P_O of 35.5kW. For 10% interest rate on total initial capital the payback period ranged between 4years at P_O of 28.9kW to 3.14years at P_O of 35.5kW as shown in figure-11.

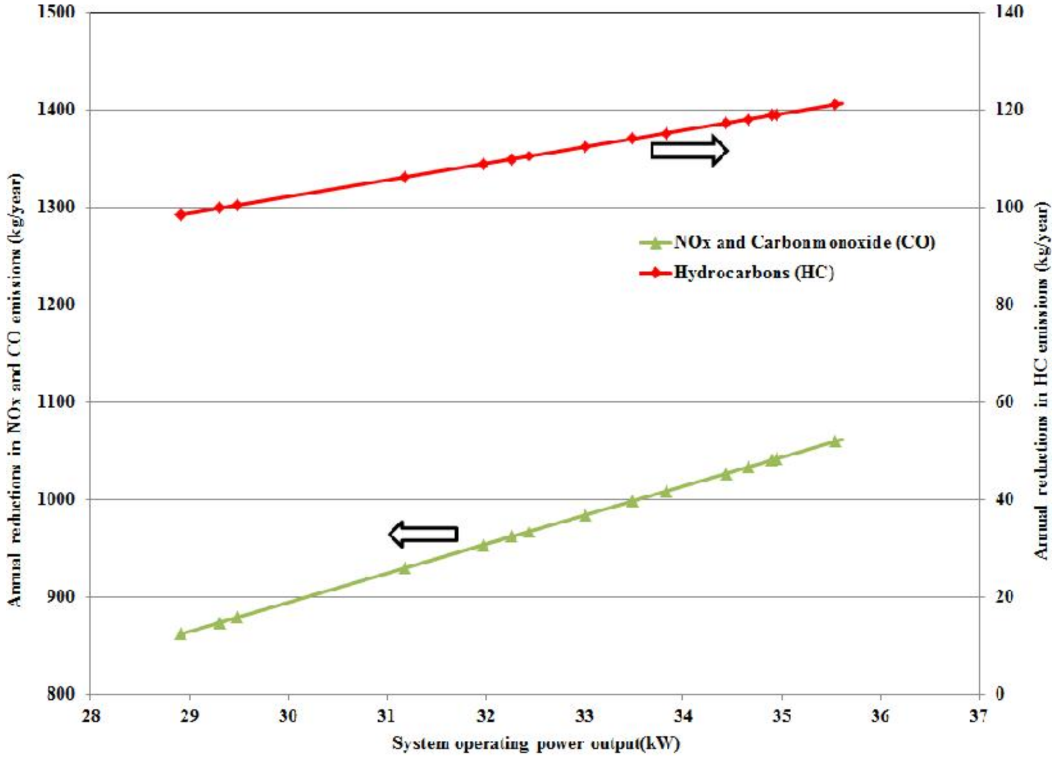

Figure-9 Annual reductions in NO_x, CO and HC vs. system operating power output (P_O) (NO_x and CO have same TIER-4 emission criteria of 3.5g/kWh each)

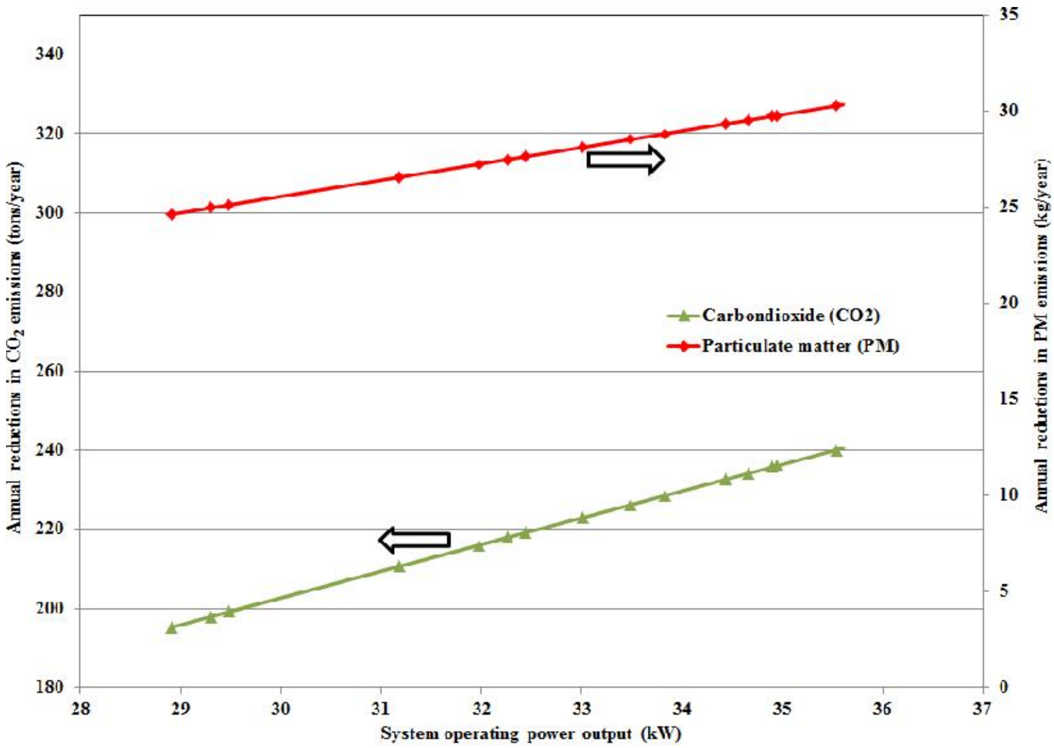

Figure-10 Annual reductions in CO_2 and PM vs. system operating power output (P_O)

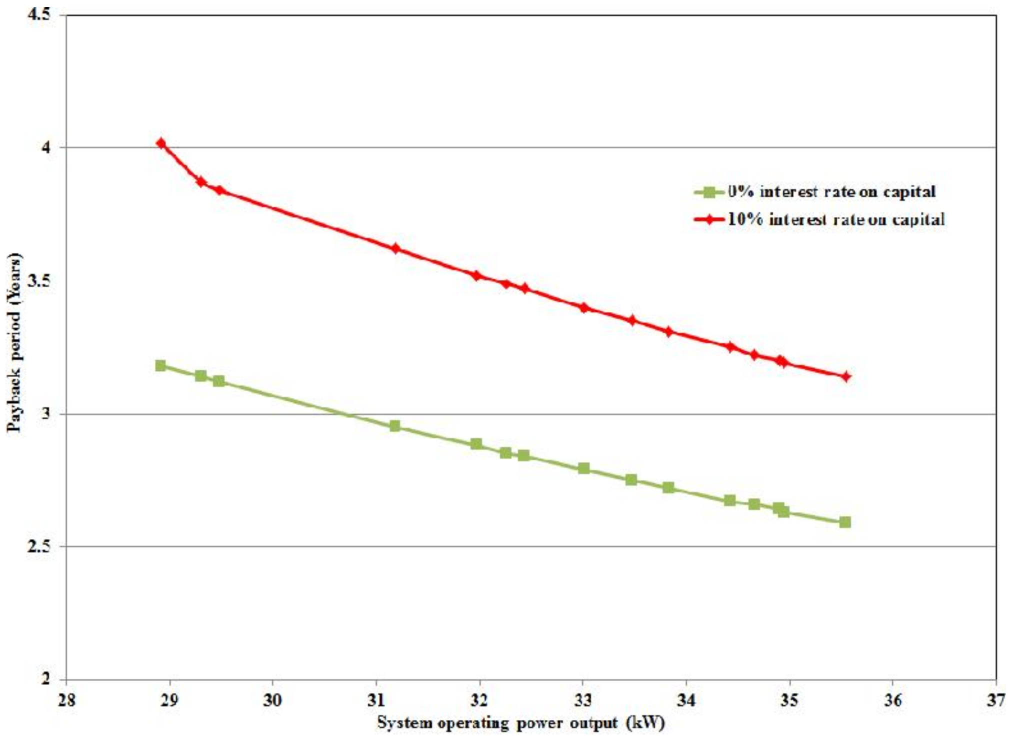

Figure-11 Payback period vs. system operating power output (P_O)

7. 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 reliability and performance according to the experimental procedure discussed. Based on the experimental test results following conclusions could be drawn:

1. The entire selected components for heat source loop, heat sink loop, electrical system and instrumentation performed as desired during reliability test and performance test i.e. no major design concerns were noticed while the system was in operation with the selected components.
2. 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.
3. From our observation of reliability test, it is possible to generate 50kW gross power output using this power unit working on waste heat from stationary diesel engines if the waste heat is from both jacket water and exhaust heat exchanger.
4. For the diesel jacket water heat recovery, case#7 (table-9) would be a better option from the performance test results based on system operating power output (32.3kW), hot water and cold water pump power consumptions (1.78kW and 1.86kW respectively), power unit efficiency (7.45%) and system operating efficiency (7.04%). Case#7 has a payback period of 2.85years and 3.5years at 0% and 10% interest rate on capital respectively.
5. On overall observation, this ORC power unit could be continuously operated on diesel engine jacket waste heat with an average system operating power output of 32.5kW and system operating efficiency of 7%. At jacket waste heat conditions the power unit has payback period less than 3.5years which is a reasonable value for rural Alaska village diesel gensets.

6. The data may 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.
7. Considerable amount of annual emissions and CO_2 (GHG) reductions could be obtained if the ORC power unit was operated year round on waste heat from diesel engines. Emissions reductions obtained from reliability and performance test results were 1372.8kg/year and 975kg/year in NO_X, 157kg/year and 112kg/year in HC, 1372.8kg/year and 975kg/year in CO, 39.2kg/year and 28kg/year in PM, and 311short-tons/year and 220short-tons/year in CO_2 respectively.
8. Due to large variation in power output from power unit for reliability test and performance test, it shows that the power unit performance is greatly dependent on heating and cooling source supply conditions. As part of a future work, it is desirable to test the power unit for more heating and cooling source supply conditions for more experimental data that will allow us to create a performance map; the map may be used to estimate the benefit that the ORC system can possibly bring to each of the rural diesel power plants in Alaska. It is also desirable to apply the experimental data to develop parametric models of the screw expander, evaporator, and condenser, which may serve as tools for thermal system design and analysis.

Acknowledgements

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Appendix I Details of Experiment Component Designs and Selections

A.1 Heat source loop

Figure-1 shows the heat source loop which is further divided into two loops viz. steam supply loop and hot water loop. 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.

The major components used to build the heat source loop are shown in Figure-1, whichon 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 Gruvlokfittings, hot water VFD pump, expansion tank, air separator, pressure relief valve. Following few paragraphs explain this component selection process.

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 waste heat 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 ranges given in Table-6, which would not be possible by using a field engine waste heat. For flow rate control of hot water in the loop a variable frequency drive (VFD) pump is used and for temperature control of the hot water a steam flow control valve with electronic actuator was used.

A.1.1 Steam-to-hot water heat exchanger

Power plant low pressure saturated steam is available at 205.7kPa (i.e. 121.11ºC).Based on the maximum heat rate requirement of 1200kW and maximum desired hot water condition of 107.22ºC (225ºF) and 68.14m^3/hr (300gpm)), a plate heat exchanger manufacturer was contacted for estimating the required size of exchanger. According to the heat exchanger manufacturer a 9.6m^2 (103.3ft^2) plate heat exchanger was sufficient for present testing and steam flow rate of 2041.2kg/hr (4500lb_m/hr) was required. A readily availableplate heat exchanger of 10.4m^2 (112ft^2) with university facility services was used for this project.

A.1.2 Steam flow control valve with actuator

Steam flow control valve with electronic actuator was used to control the hot water supply temperature to the power unit by varying the amount of steam flow rate through heat exchanger.Electronic actuator was used to remotely control the steam valve opening through LabVIEW program. Normally closed (NC) valve was used for safety concerns.

Valve coefficient (C_V) is used for sizing and selection of a valve based on the expression given in handbook [38]. Based on the obtained C_V of 130, a Siemens 4” normally closed steam control valve with SKC62U electronic actuator was selected for following reasons. SKC62U electronic actuator was used to control steam valve opening with 0-10Vdc signal from national instruments (NI) data acquisition system (DAQ) and LabVIEW programming.

A.1.3 Steam trap

Steam trap selection is based on the maximum operating steam pressure [40]. As the maximum steam available pressure is at 205.7kPa (29.84PSIA), a SpiraxSarco FT-30 steam trap was used in the current installation.

A.1.4 Variable frequency drive (VFD) pump

VFD pump was selected for the current project because of the need to test the power unit for different hot water flow rates during performance testing. Bell & Gossett series-80, 11” impeller diameter pump mounted to a 20hp (14.9kW) motor which is rated at 1750rpm and rated for VFD operation was selected. Design flow rate of the pump was 56.8m^3/hr (250gpm). S-Flex Altivar-21 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.Pump selection was based on the estimated 344.7kPa (50psi) pressure drop in the hot water piping, power unit evaporator, and steam-to-hot water heat exchanger.

A.1.6 Expansion tank, Air separator, and Pressure relief (PR) valve

A 0.076m^3 (20gal) Extrol SX-40V expansion tank which has a maximum temperature limit of 115.5ºC (240ºF) was selected and installed. As the water in a closed loop system is heated or cooled it expands or contracts, expansion tank provides a space into which the non-compressible liquid can expand or contract as the liquid undergoes volumetric changes with change in temperature.Expression given in ASHRAE handbook [41] was used to calculate the minimum required expansion tank volume to be 0.045m^3 (12gal).

As the hot water loop is a closed loop circuit,it requires removing air from the loop while filling the loop and while in operation (if any air infiltrates into the loop). Bell & Gossett Rolairtrol R-4F air separator which has 4” flanged inlet and outlet was selected for the present project. The selection of the air separator is based on maximum water flow rate in the loop of 68.1m^3/hr (300gpm) [42].

A PR is installed to prevent the system from exceeding the safe limits of the system components which may occur from over pressurization of system during filling or pressure increase caused by thermal expansion or surges caused by water hammer. A 310.3kPaG (45psig) Bell & Gossett pressure relief valve was used in the current installation. It was installed in the hot water loop at the pump inlet point (Figure-1).

A.2 Heat sink loop

In a typical Alaska village installation, the cooling source may be fluid from cooling tower, radiator, large water body (such as near-by river, lake), or underground well water. As the main aim of the project is performance and reliability testing of the power unit so that the results could be used to predict the performance of the unit for real village conditions. Considering the geographical location of Alaska the available cooling source temperature is usually less than 20ºC (68ºF) year round. As per the power unit specification for 50kW ORC power unitthe heat sink requirement was 530kW (approx. 1.8MBTU/hr) of heat will be rejected by the refrigerant in condenser to cooling fluid.Considering availability of cooling source for the present testing, the cold water from a near-by fire hydrant which was located just outside the power plant building was used.Temperature of water from fire hydrant is always around 10ºC (50ºF).

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. 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 ditched into a nearby open space.

For 600hrs of reliability test, during which the power unit was tested for full load, the cold water was supplied at 10ºC (50ºF) and 36.3m^3/hr (160gpm). For this the manual flow control valve was adjusted to a cold water flow rate of 36.3m^3/hr through the loop, and the pump, 3-way butterfly valves and bypass line were shutoff (i.e. not used). The following paragraphs explain few component selection process and reason for selection.

A.2.1 3-way butterfly valves, Bypass connection, and Cold water pump

During performance test the power unit was tested for 10ºC (50ºF) and 20ºC (68ºF) (Table-6). The bypass line is used to test the power unit for cold water temperature of 20ºC (68ºF). 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 20ºC (68ºF) cold water temperature into the power unit is reached.

Triad 3-way butterfly valves were installed in the present experimental setup. Radius EW-880 actuators were used to control the valve position. Two 3-way butterfly valves were used, one in supply line of piping and another in return line of piping, for better control of cold water supply temperature to power unit. Valves position were controlled manually by using a 0-10Vdc signal input to the actuators from national instruments (NI) data acquisition (DAQ) system using LabVIEW programming.

Cold water pump was a 15hp (11.2kW) Scott pump which is rated at 3500rpm. This is not a VFD pump. For 10ºC cold water supply temperature to power unit, water pressure from fire hydrant was sufficient to move water in the loop and the pump was shutoff. The pump, bypass line and 3-way butterfly valves were used only in the case of performance test case when the cold water supply temperature was 20ºC.

A.2.2 Manual valve for flow control

As the power unit was tested at three different cold water flow rates, viz. 27.2m^3/h (120gpm), 36.3m^3/h (160gpm) and 45.4m^3/h (200gpm), a POK butterfly valve was mounted directly on the fire hydrant outlet to manually control cold water flow rate to the power unit.

A.2.3 Check valve

Walworth check valve was selected for present installation. The check valve was used to prevent back flow of water into the fire hydrant when the pump is in operation (for performance test case), a local city code need to be followed for fire hydrant to prevent contamination of the water source.

A.3 Electrical system

As the main purpose of the present experimental setup is waste heat recovery from Alaska diesel power plants, in a typical village installation the generated electrical power from the ORC power unit would be uploaded to the diesel power plant grid. In the present case too as the experimental setup was located within the UAF power plant, the generated power 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 #4AWG cable was used to connect VFD CB box of hot water pump and safety switch of cold water pump as shown in Figure-2. #8AWG and #10 AWG cables was used for powering the motor of hot water pump from VFD and the motor of cold water pump from safety switch respectively.

A.3.1 Cable size selection process

Cable size selection depends on (i) voltage drop in cable due to distance between the source and load (e.g. from power unit to motor center CB), (ii) allowable amperage for cable, (iii) local codes, and (iv) readily availability of selected cable size. According to industry practices less than 3% voltage drop at the load with respect to line voltage is allowable. Following few paragraphs explain the various cable size selection processes.

According to the ORC power unit specifications, generator power output had a line voltage of 480V, full load current of 75A, 3-phase and 60Hz. A 38.1m (125ft) cable was required for wiring from power unit circuit breaker (CB) to motor center CB. By using the expression given in electrical reference handbook [43] for 75A capacity and MC cable, we have to use a minimum conductor size of #1AWG or higher. Due to the geological location of Fairbanks and also due to project timeline, #1AWG cable had a long lead time. Only readily available cable which was used in the present installation was 3/0AWG MC 3-conductor cable with ground, which has an amperage rating of 175A and voltage drop of 0.26% for 38.1m.

Byusing the expression given in handbook [43], as shown in Figure-2, #4AWG cable was used for wiring from power unit CB box to VFD and safety switch. #4AWG cable had rated amperage of 95A [43] and voltage drop of 0.09% for 3m length of cable.VFD circuit breaker (CB) was rated for 50A and cold water switch for 30A, combined to 80A.

Hot water pump, which is rated for VFD operation, had a 20hp (14.9kW) rated motor size with input of 460V/24A. Cold water pump had 15hp (11.2kW) rated motor size with input of 460V/18A. As shown in Figure-2, #8AWG and #10AWG cables served the purpose of poweringhot water pump and cold water pump motor from respective circuit breaker boxes. #8AWG cable had rated amperage of 55A [43] and voltage drop of 0.5% for 20m length of cable, and #10AWG cable had rated amperage of 40A [43] and voltage drop of 0.08% for 3m length of cable.