Technical Assessment of the Combined Heat and Power Plant at the Oregon Institute of Technology, Klamath Falls, Oregon

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ABSTRACT

The recently installed combined heat-power (CHP) plant at the Oregon Institute of Technology is described and its performance analyzed using thermodynamic First and Second Law principles based on energy and exergy, respectively. Characteristics of the three production and two injection wells are presented. Real-time plant data for the binary cycle and heating system are shown in a screen-shot from the control panel and used to carry out a system analysis. Both the power cycle by itself and the whole CHP system are assessed. The R245fa working-fluid power cycle is shown to have a thermal efficiency of 8.2% and a utilization efficiency of 33.5% relative to the exergy change of the geofluid, and the CHP system has an efficiency of 83.6%, using geofluid pumped to the plant at 196.9 ºF.

Brief History of Geothermal Energy Usage at OIT

For over one hundred years, the people of Oregon have been using geothermal energy to heat buildings, melt snow from sidewalks, grow plants in greenhouses, and more. Situated 25 miles north of the border with California (see Figure 1), the community of Klamath Falls lies atop a particularly abundant supply of geothermal energy. One thousand homes are heated with hot water obtained from nearly 600 wells.

The existence of these geothermal resources was the motivation behind Oregon Institute of Technology (OIT) moving its Klamath Falls campus in 1964 to its present location in the northern part of the city. Specifically, the newly-constructed school was designed to tap hot water from the geothermal reservoir to heat campus buildings. Today that geothermal district heating system serves sixteen buildings totaling roughly 818,200 square feet of floor space at OIT (see Figures 2 and 3, next page).

The institute is the only 100% geothermally-heated campus in North America. Now, with the inauguration of its first combined heat and power plant (CHP), OIT is well on its way to becoming not only geothermally heated but also geothermally electric powered with geothermal resources found on its own property. When this effort is brought to completion, this will set OIT apart from all other institutions of higher education in the world.

Production and Injection Wells

There are three production wells in service to supply the OIT CHP plant: OIT-2, -5 and -6. Two injection wells receive the waste

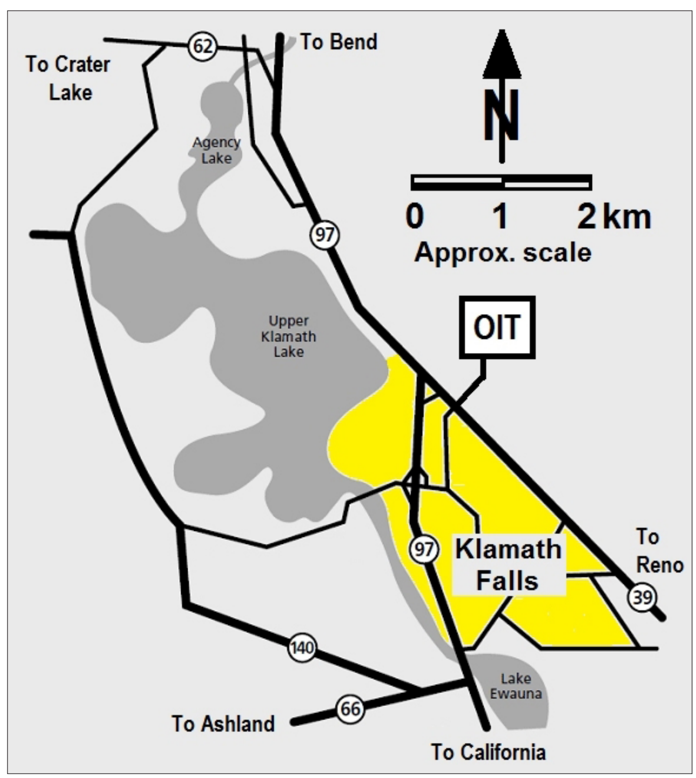

Figure 1. Location map for OIT.

Figure 2. Aerial view of OIT campus and CHP plant [*Google Earth* image, August 8, 2011].

geofluid from the heating system: OITINJ-1 and -2. These are shown in the campus layout map in Figure 3. Selected information of these wells is given in Table 1.

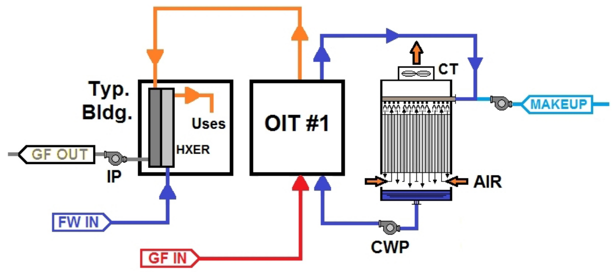

Figure 5. Overall system schematic flow diagram.

Figure 6. ORC power module photo [2].

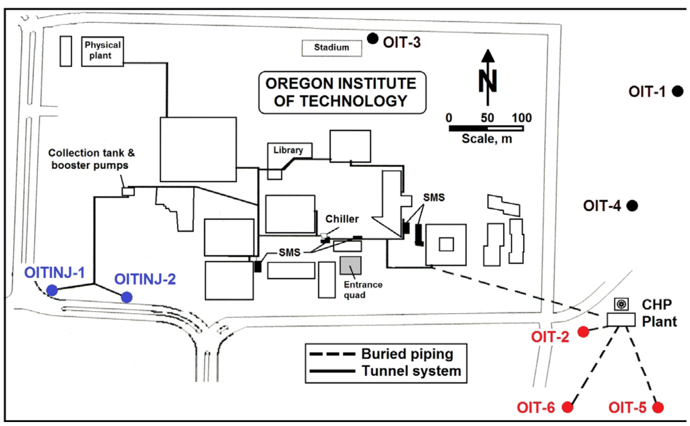

Figure 3. Layout of the campus of OIT and the locations of the production (red) and injection wells (blue): wells OIT-1 and -4 are used for domestic water, irrigation and cooling tower makeup; OIT-3 is not in use; SMS=Snow-Melt System; scale is approximate; modified and updated from [1].

Figure 4. Power house with water cooling tower [2].

Table 1. Selected characteristics of active OIT wells.

	Production Wells			Injection Wells	
Well No.	OIT-2	OIT-5	OIT-6	OITINJ-1	OITINJ-2
Total depth	1,288 ft (393 m)	1,716 ft (523 m)	1,800 (549 m)	2,005 ft (611 m)	1,675 ft (511 m)
Depth to Static Water Level	332 ft (101 m)	358 ft (109m)	359 ft (109 m)	234 ft (71 m)	173 ft (53 ft)
Volumetric Flow Rate	150 GPM (9 L/s)	460 GPM (29 L/s)	350 GPM (22 L/s)	400 GPM (25 L/s)	1,000 GPM (63 L/s)
Pump Mfgr.	Goulds	Goulds	Layne/ Bowler	N.A.	N.A.
Power	50 hp (37.3 kW)	75 hp (55.9 kW)	75 hp (55.9 kW)	N.A.	N.A.
Pump Setting Depth	700 ft (213 m)	440 ft (134 m)	600 ft (183 m)	N.A.	N.A.
Wellhead Temperature	192 °F (89 °C)	195 °F (91°C)	197 °F (92 °C)	98 °F(1) (37 °C)	80 °F(1) (27 °C)

(1) Original produced fluid.

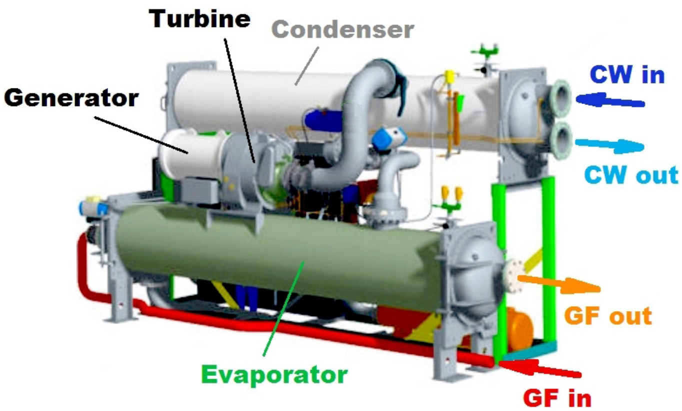

Figure 7. Power module schematic; feedpump and motor are located at ground level behind evaporator; control panel is at left rear (not visible).

Table 2. Selected characteristics of PureCycle® unit.

Item, units	Value
Working Fluid	R245fa[(1)]
Maximum Rated Gross Power, kW	280[(2)]
Maximum Rated Net Power, kW	260[(3)]
Turbine Type	Radial inflow
Generator Type	Induction
Power Factor (Lagging)	>0.95
Noise (at 33 ft), dBA	78
Dimensions (L x W x H), ft	19.9 x 7.5 x 11.25
Operating Weight, lbm	33,300
Inlet Fluid Temperature Range, °F	195-300

[(1)] 1,1,1,3,3-pentafluoropropane
[(2)] At 480 V/3-phase/60 Hz
[(3)] At 60 Hz.

Power Plant Design

The OIT combined-heat-power plant is comprised of one modular organic Rankine cycle (ORC), a water cooling tower, and individual heat exchangers in various campus buildings. Three wells are available to send hot geofluid to the plant, although only two wells were in operation on the day the data were taken on which this paper is based. The power house and cooling tower are shown in Figure 4 and in simplified form, in the flow diagram depicted in Figure 5.

The ORC was manufactured and supplied by Pratt & Whitney Power Systems and is called a Model 280 PureCycle® [3]. Figure 6 is a site photo and Figure 7 is 3-D schematic rendering. Some characteristics of the unit are given in Table 2.

Table 3. State-point properties for geofluid and cooling water; see Figures 8 and 10.

State	Temperature °F	Pressure psia	Volume flow GPM	Specific volume ft^3/lbm	Enthalpy Btu/lbm	Entropy Btu/ lbm.R
			Geofluid			
1	196.9	26.72	624.3	0.01661222	165.16	0.28953
2[(1)]	NA	NA	624.3	---	TBD	TBD
3	163.0	15.56	624.3	---	131.11	0.23635
0	37.2 (wb)	12.34	---	---	5.2575	0.010563
			Cooling Water			
4	56.3	30	1,309	0.016029	24.477	0.048415
5[(1)]	NA	NA	1,309	---	TBD	TBD
6	69.8	15	1,309	---	37.942	0.074261

[(1)] Pinch-points.

Combined Heat-Power Plant Overall Performance

The performance of the OIT Unit 1 power plant will be analyzed using the data obtained during a snapshot taken on January 20, 2012; see Figure 8. The relevant data for the geofluid and the cooling water are shown in Table 3; specific volume, enthalpy and entropy values were found using REFPROP software [4]. The net power of the ORC is used on site to run the

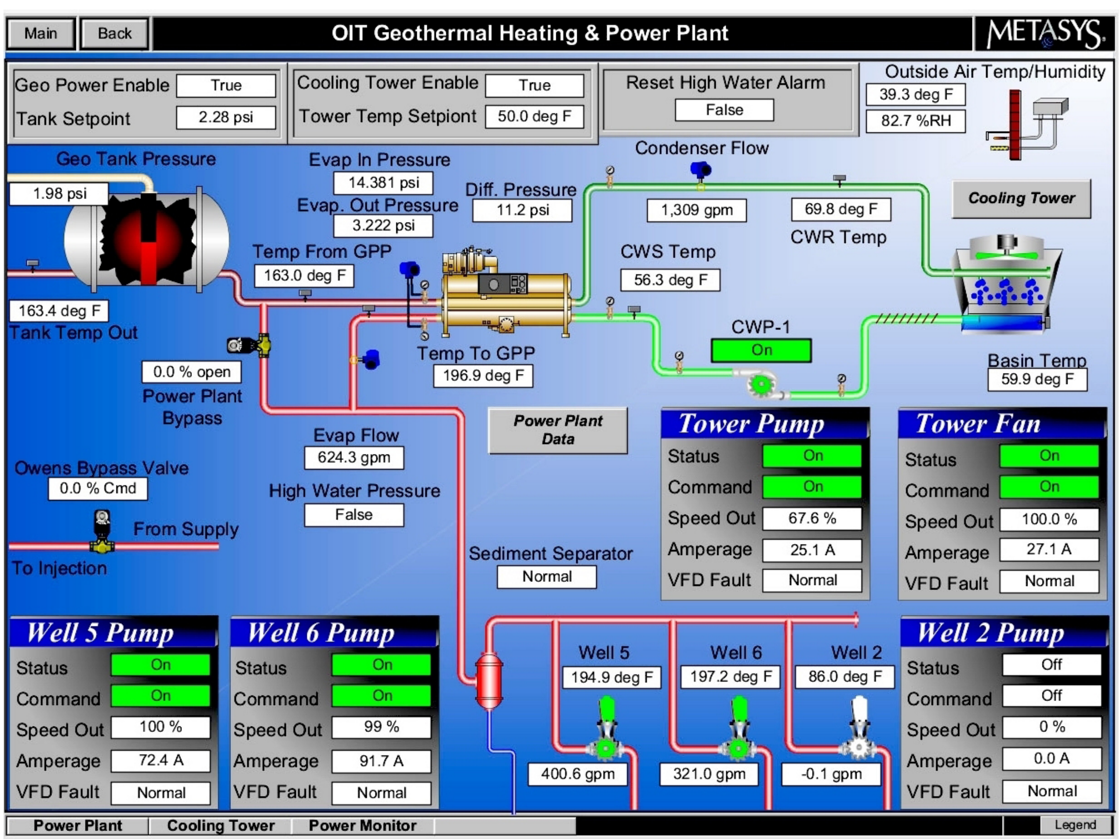

Figure 8. Screen shot of system flow diagram, January 20, 2012.

well pumps, while the rest of the power is delivered to the campus. These data were obtained from another screen of the METASYS monitoring system, and the values are shown in Table 4. Although not shown in the screen shots, the geofluid temperature after leaving the heating system and entering the reinjection wells is 135°F.

Table 4. Power generation and usage.

Item, Units	Value
Net Cycle Power, kW	225.9
Well-Pumping Power, kW	148.0
Power Delivered to OIT, kW	77.9

The goal of this section is to determine the thermal and utilization efficiencies of the plant. Although data for the geofluid and the cooling water are known, nothing is known about the thermodynamic state properties of the R245fa within the ORC since the manufacturer holds this information as proprietary. Thus, the overall performance is easy to calculate, but the detailed performance assessment of the cycle is not straightforward and will require several assumptions.

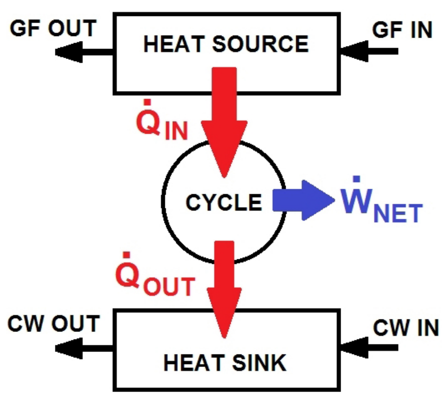

Figure 9. Simple overall system schematic.

Figure 9 is a block diagram for the cyclic power unit (ORC) and its heat source and sink. Figure 10 is a more detailed representation of the plant, albeit still simplified.

The power cycle consists of the usual processes used in binary power plants:

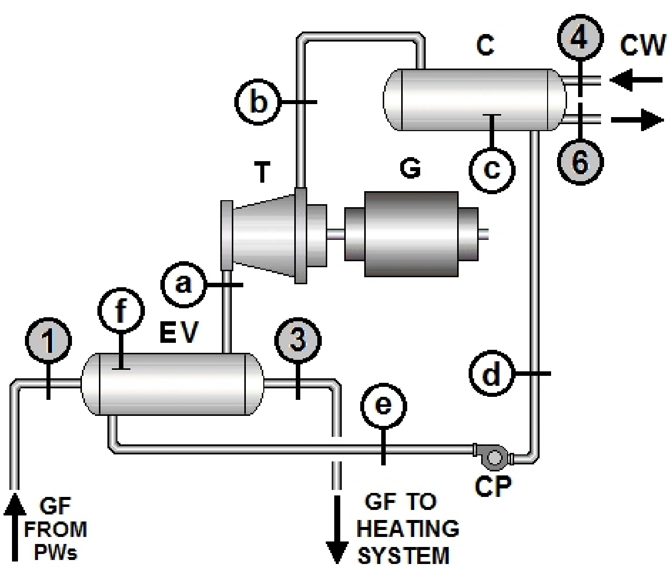

Figure 10. Simplified PureCycle® plant schematic flow diagram for OIT Unit 1.

a-b: turbine expansion (power generation)
a-bs: ideal isentropic turbine expansion (theoretical process)
b-c: desuperheat removed in condenser
c-d: Heat of condensation removed in condenser
d-e: pressurization of liquid in feed pump
d-es: ideal isentropic pressurization (theoretical process)
e-f: sensible heat received in evaporator (preheating)
f-a: latent heat received in evaporator (boiling).

The preheating (sensible heat) and the boiling (evaporation) both take place within a single shell-and-tube heat exchanger that is called the “evaporator”, EV. The geofluid enters the evaporator at one end and makes three passes, leaving at the opposite end. The R245fa enters at the bottom, flows through a series of baffled spaces within the shell, and leaves as a saturated vapor (assumed) at the top. Similarly, the desuperheating of the R245fa coming from the turbine takes place within a single shell-and-tube heat exchanger (the “condenser”, C) that also does the job of condensing the working fluid. The cooling water from the cooling tower enters and leaves at one end of the condenser shell, making four passes inside. The R245fa enters at the top and leaves at the bottom as a saturated liquid (assumed).

If the operation were ideal in the sense that all the heat removed from the geofluid (heat source) was actually transferred to the cycle working fluid, R245fa, and all the heat rejected by the R245fa actually ended up in the cooling water (heat sink), as shown in Figure 9, then the plant performance could be easily determined from the data given for the geofluid and the cooling water. With reference to Figure 9, using basic thermodynamics,

$$\dot{Q}_{IN} - \dot{Q}_{OUT} = \dot{W}_{NET}\,. \tag{1}$$

Using the state-point notation in Figure 10,

$$\dot{Q}_{IN} = \dot{m}_{GF}(h_{IN} - h_{OUT})_{GF} = \frac{\dot{V}_{GF}}{v_{GF,1}}(h_1 - h_3) \tag{2}$$

and

$$\dot{Q}_{OUT} = \dot{m}_{CW}(h_{OUT} - h_{IN})_{CW} = \frac{\dot{V}_{CW}}{v_{CW,4}}(h_6 - h_4)\,. \tag{3}$$

Note that in the flow diagram Figure 10, we have reserved the state points 2 and 5 for the respective pinch-points of the geofluid and cooling water with the R245fa.

The mass flow rate of geofluid is found from the inlet conditions:

$$\dot{m}_{GF} = \frac{\dot{V}_{GF}}{v_{GF,1}} = \frac{624.3 \times 0.13366 \times 60}{0.01661222} = 301{,}382.8 \text{ lbm/h.} \tag{4}$$

The mass flow rate of cooling water is found similarly:

$$\dot{m}_{CW} = \frac{\dot{V}_{CW}}{v_{GF,4}} = \frac{1{,}309 \times 0.13366 \times 60}{0.01603019} = 654{,}867.8 \text{ lbm/h.} \tag{5}$$

Thus, the heat removed from the geofluid and the heat absorbed by the cooling water are, respectively:

$$\dot{Q}_{IN} = 301{,}382.8 \times (165.16 - 131.11)/3412 = 3{,}007.17 \text{ kW and} \tag{6}$$

$$\dot{Q}_{OUT} = 654{,}867.8 \times (37.942 - 24.477)/3412 = 2{,}534.17 \text{ kW.} \tag{7}$$

Thus, without any heat losses, the expected net cycle power would be:

$$\dot{W}_{NET} = 3{,}007.17 - 2{,}534.17 = 473.0 \text{ kW.} \quad (8)$$

However, the actual net power registered by the control system is only 225.9 kW and so, unsurprisingly, the system is non-ideal. Thus, eq. (1) cannot be used to gauge the system performance when the heat values are found from the geofluid and cooling water data. The basic equation, however, still applies to the R245fa cycle:

$$\dot{Q}_{IN,WF} - \dot{Q}_{OUT,WF} = \dot{W}_{NET} = 225.9 \text{ kW.} \quad (9)$$

Clearly, $\dot{Q}_{IN,WF} \le 3{,}007.17$ kW and/or $\dot{Q}_{OUT,WF} \ge 2{,}534.17$ kW. In other words, either not all of the heat released from the geofluid ends up in the R245fa in the evaporator, or more heat is released by the R245fa in the condenser than is received by the cooling water, or both. Since the geofluid is the hottest fluid in the system, any imperfections in the insulation of the geofluid piping and evaporator covering would make it more likely that the former is true. Given the lower temperatures involved at the cold end of the plant, it is likely that the heat loss there is less than at the hot end. This will be used later to help understand the performance of the ORC unit.

Regardless of the non-ideality of the system, the overall thermal efficiency of the power plant can nevertheless be calculated:

$$\eta_{TH} = \dot{W}_{NET} / \dot{Q}_{IN,GF} = 225.9 / 3{,}007.17 = 0.0751 \text{ or } 7.51\%. \quad (10)$$

The actual thermal efficiency of the ORC cycle itself will be somewhat higher than this.

The Second Law utilization efficiency can be found relative to the flow of exergy into the plant:

$$\eta_{U1} = \dot{W}_{NET} / \dot{E}_{GF,1} \quad (11)$$

where the incoming exergy is given by:

$$\dot{E}_{GF,1} = \dot{m}_{GF,1}\left[h_1 - h_0 - T_0(s_1 - s_0)\right] \quad (12)$$

$$= 301{,}382.8\left[165.16 - 5.2575 - (37.2 + 459.67)(0.28953 - 0.010563)\right] /3412 = 1{,}880.7 \text{ kW.} \quad (13)$$

The thermodynamic dead state has been taken at the wet-bulb temperature (37.2ºF) for the ambient conditions at the plant site and at the standard atmospheric pressure (12.34 psia) for the elevation of the plant (4,429 ft asl).

Thus,

$$\eta_{U1} = 225.9 / 1{,}880.7 = 0.120 \text{ or } 12.0\% \quad (14)$$

We may also calculate a utilization efficiency relative to the change in exergy of the geofluid as it passes through the unit:

$$\eta_{U2} = \dot{W}_{NET} / \Delta\dot{E}_{GF} \quad (15)$$

$$\Delta\dot{E}_{GF} = \dot{m}_{GF,1}\left[h_1 - h_3 - T_0(s_1 - s_3)\right] \quad (16)$$

$$= 301{,}382.8\left[165.16 - 131.11 - (37.2 + 459.67)(0.28953 - 0.23635)\right] /3412 = 673.65 \text{ kW} \quad (17)$$

$$\eta_{U2} = 225.9 / 673.65 = 0.3353 \text{ or } 33.53\%. \quad (18)$$

The heating applications supplied by the geofluid after leaving the power plant may be lumped together and added to the useful output of the ORC to assess the full performance of the combined heat and power plant. Knowing the temperatures in and out of the heating system and the geofluid flow rate, the thermal power delivered from the geofluid may be calculated from Eq. (2) written between 163.4 and 135ºF where a heat transfer efficiency of 90% is assumed between the geofluid and the secondary water in the building heat exchangers:

$$\dot{Q}_{HTG} = 0.9 \times \dot{m}_{GF}(h_{HTG,IN} - h_{HTG,OUT})_{GF} = 0.9 \times \frac{\dot{V}_{GF}}{v_{GF,IN}}(h_{HTG,IN} - h_{HTG,OUT}) \quad (19)$$

$$\dot{Q}_{HTG} = \frac{0.9 \times 624.3 \times 0.13366 \times 60}{0.016412}(131.51 - 103.08) / 3412 = 2{,}287.7 \text{ kWt.} \quad (20)$$

Thus, 2,287.7 kWt of direct heating can be attributed to the CHP plant. Thus, the total energetic benefit of the plant is 225.9 + 2,287.7 = 2,513.6 kW. The overall thermal efficiency becomes:

$$\eta_{TH,CHP} = (\dot{W}_{NET} + \dot{Q}_{HTG}) / \dot{Q}_{IN,GF} = 2{,}513.6 / 3{,}007.17 = 0.836 \text{ or } 83.6\%. \quad (21)$$

ORC Performance

Returning now to the problem of determining the performance of the ORC unit, an attempt will be made to thermodynamically fit the ORC between the geofluid cooling curve and the cooling water warming curve. This cannot be done precisely (or uniquely) because no data is available from the ORC manufacturer except the working fluid, R245fa. However, by assuming reasonable values for a set of parameters, it will be possible to arrive at a plausible ORC cycle.

Figures 11 and 12 show the temperature-heat transfer diagrams for the "evaporator" and "condenser", respectively, in schematic form. Note that the "evaporator" incorporates both preheating and evaporation, and the "condenser" incorporates both desuperheating and condensation. In Figure 11, by postulating the R245fa evaporating pressure and the pinch-point temperature difference, $\Delta T_{PP,EV}$, and knowing the geofluid temperatures and flow rate, the First Law energy balance may be applied to the evaporator to determine the R245fa mass flow rate. A similar exercise on the condenser, using its pinch-point temperature difference, $\Delta T_{PP,C}$, will also yield the R245fa mass flow rate. It is not expected that the two values will be equal owing to the heat losses mentioned earlier and some means must be found to account for this situation.

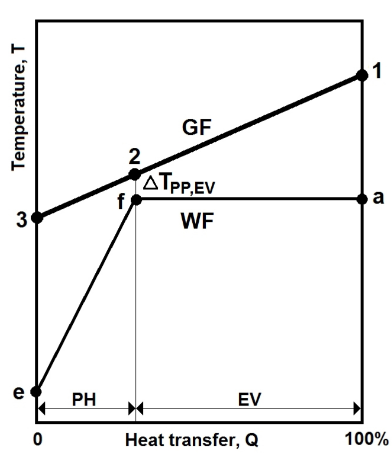

Figure 11. Temperature-heat transfer diagram for "evaporator".

Simultaneously, the ORC turbine power, pump power and generator output can be calculated with the aid of chosen isentropic efficiencies for the turbine and pump and a generator mechanical-to-electrical conversion efficiency. Thus, a multi-variable search must be

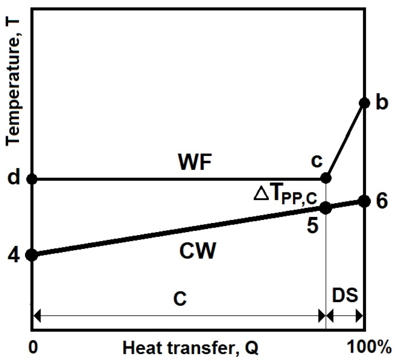

Figure 12. Temperature-heat transfer diagram for "condenser."

carried out until the net ORC power agrees with (or compares very well) with the measured value. The following parameters need to be adjusted while searching for a reasonable answer:

R245fa evaporator pressure ($P_e = P_f = P_a$)
R245fa condenser pressure ($P_b = P_c = P_d$)
Evaporator pinch-point temperature difference, $\Delta T_{PP,EV}$
Condenser pinch-point temperature difference, $\Delta T_{PP,C}$
R245fa turbine isentropic efficiency, η_T
R245fa pump isentropic efficiency, η_P.

The generator efficiency, η_G, was set at 0.95 (95%) and kept constant.

An Excel spreadsheet was written to perform the calculations and REFPROP was embedded in it to obtain all thermodynamic properties for the geofluid (assumed pure water), the cooling water, and the R245fa.

The method of solution is as follows. The pinch-points were taken at the bubble point in the evaporator and at the dew point in the condenser. The locations of these points along the GF and CW curves were assumed as a first guess; i.e., at a certain percentage of the total heat transfer in each heat exchanger; see Figures 11 and 12. Thus, the temperature was found on the GF and CW lines at the pinch-points. Using assumed values for the ΔT_{PP}-terms, the saturation temperatures for the R245fa in the evaporator and in the condenser were found. The heat transfer terms in the evaporator and condenser were calculated along with the matching R245fa mass flow rates. Then the percentage of heat transfer to each pinch-point was calculated and compared to the earlier assumed values. Adjustments were successively made until agreement was obtained. The power terms were also found at each iteration and compared to the measured value of net ORC power. Eventually, the calculations converged to yield a net power of 225.9 kW, but as expected, the mass flow rates of R245fa calculated for each heat exchanger differed significantly, being about 10% apart, and this solution was deemed unacceptable.

In order to simulate the apparent heat loss between the geofluid and the R245fa, the heuristic assumption was made that only 92% of the heat removed from the geofluid was effectively delivered to the R245fa; i.e., there is an 8% heat loss. Additionally, no loss was ascribed to the heat transfer at the condenser end of the plant. Closure was achieved on the iterative solution using the following values for system parameters:

R245fa evaporator pressure, $P_e = P_f = P_a = 85.24$ psia

R245fa condenser pressure, $P_b = P_c = P_d = 24.86$ psia

Evaporator pinch-point temperature difference,

$$\Delta T_{PP,EV} = 16\ °\text{F}$$

Condenser pinch-point temperature difference,

$$\Delta T_{PP,C} = 15\ °\text{F}$$

R245fa turbine isentropic efficiency, $\eta_T = 0.85$

R245fa pump isentropic efficiency, $\eta_P = 0.75$.

The final results for the state-point properties of the R245fa in the ORC are shown in Table 5. The mass flow rates now differ by only ±0.2%, an acceptable amount given the level of uncertainty inherent in the analysis. The R245fa mass flow rate through the evaporator was calculated from:

$$\dot{m}_{R,EV} = \dot{m}_{GF} \frac{0.92(h_1 - h_3)}{h_a - h_e} \tag{22}$$

and through the condenser from:

$$\dot{m}_{R,C} = \dot{m}_{CW} \frac{(h_6 - h_4)}{h_d - h_d}. \tag{23}$$

The turbine and pump power were calculated using the average of these two mass flow rates. The net ORC power under these conditions is 226.0 kW, only 0.1 kW higher than the measured power, or 0.04% error which is probably less than the accuracy of the instrumentation. However, it must be stressed that this solution is not unique as there may be other combinations of the system parameters that might give equivalent results. The cycle processes are shown to scale in Figure 13, a temperature-entropy diagram.

Table 5. State-point properties for ORC working fluid R245fa.

State	Temperature °F	Pressure psia	Enthalpy Btu/lbm	Entropy Btu/lbm.R	Mass Flow lbm/h
Steam & Condensate State-Points					
a	155.54	85.24	195.34	0.42371	101,630(1)
bs	---	24.86	185.73	0.42371	---
b	101.65	24.86	187.17	0.42630	101,850(2)
c	84.16	24.86	183.13	0.41899	101,850(2)
d	84.16	24.86	102.28	0.27031	101,850(2)
es	---	85.24	102.41	0.27031	---
e	84.57	85.24	102.46	0.27039	101,630(1)
f	155.54	85.24	125.89	0.31082	101,630(1)

(1) Obtained from Eq. (19)   (2) Obtained from Eq. (20).

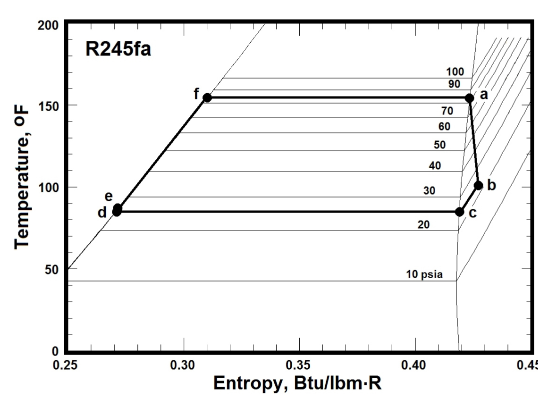

Figure 13. R245fa processes for OIT unit in temperature-entropy coordinates.

The heat and work transfer terms were found from the standard thermodynamic equations. Cycle and plant efficiencies were computed using the First and Second Laws of thermodynamics. Table 6 shows the results for the ORC cycle.

Table 6. Calculated ORC cycle results for state-points given in Table 5.

Item, Units	Value
Evaporator Heat Duty, kWt	2,766.6
Specific Turbine Power, Btu/lbm	8.167
Gross Turbine Power, kW	243.53
Generator Gross Output, kW	231.35
Condenser Heat Duty, kWt	2,534.2
Specific Pump Power, Btu/lbm	0.1798
Pump Power, kW	5.361
Generator Net Output, kW	226.0
Thermal Efficiency, %	8.2

Conclusion

The OIT CHP plant serves both as an educational opportunity for students and as an economic, green means of providing heat and electricity to the campus. In light of the relatively low temperature of the geofluid entering the plant, the efficiencies based on energy and exergy are quite reasonable. Accounting for both heat delivered to the campus buildings and electricity generated, the CHP plant is about 84% efficient in terms of the heat delivered by the incoming geofluid. Since the plant allows OIT to avoid buying electricity from the regional supplier, Pacific Power, this means an avoidance of carbon dioxide emissions in proportion to the generation mix by Pacific Power that includes 79% from fossil fuels, coal and natural gas combined. The plant requires no human supervision and basically runs itself. One operating problem involved the cooling tower freezing up on the external surface but that has been taken care of by the facilities personnel.

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