ME 489 Thesis

Modeling Night-time Heat Rejection of Supercritical Carbon Dioxide Power Cycle

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Introduction

This report summarizes the work done during the Fall 2019 and Spring 2020 semesters for the course ME 489. The project models the performance of a supercritical carbon dioxide power cycle. There were four main stages in this project. The first was to develop an interpolation scheme for the performance of the cycle. The second was to compare the cycle performance in the hottest and coldest hour of each day over a year. The third stage was to model the supercritical carbon dioxide power cycle so that the residual heat is rejected in at night instead of during the day. The final stage was to use the model to estimate the capital cost of night-time heat rejection. The stages of the project, results, and conclusions are discussed in the report. The future of this project is also discussed.

Interpolation scheme

All the simulations run in this project are based on the off-design supercritical carbon dioxide power cycle analysis scheme developed by Ty Neises. An off-design model is one that takes variables characteristic to the compressors, recuperator, and inlet conditions as the input and predicts the performance of the supercritical carbon dioxide power cycle. The inputs of scheme that was used include design parameters of the components, inlet temperature, inlet pressure, and ambient temperature. The only input to the off-design scheme that was varied was ambient temperature. The interested outputs of the scheme were cycle power generated, cycle efficiency and required heat.

The interpolation scheme is computationally intensive and requires a considerable amount of time to run. Using the off-design scheme to evaluate cycle performance at a particular ambient temperature would be unfeasible. To analyze the annual off-design cycle performance at Daggett, CA, the range of temperatures over a year had to be used as the input to the scheme. An annual weather file from Daggett, CA was used in which the temperature ranged from -3°C to 44°C. The off-design scheme was run once at every 1°C interval over the entire temperature range and the compressor inlet temperature, cycle power, and efficiency corresponding to an ambient temperature was saved at every iteration.

Since the cycle power and efficiency at every 1°C interval of ambient temperature was known, an interpolation scheme was built so that the original off-design scheme would not have to be run. This scheme implemented a simple linear interpolation method that could take either ambient or compressor inlet temperature and output the cycle power, efficiency, and heat power corresponding to that temperature. Heat power is defined as cycle power over efficiency.

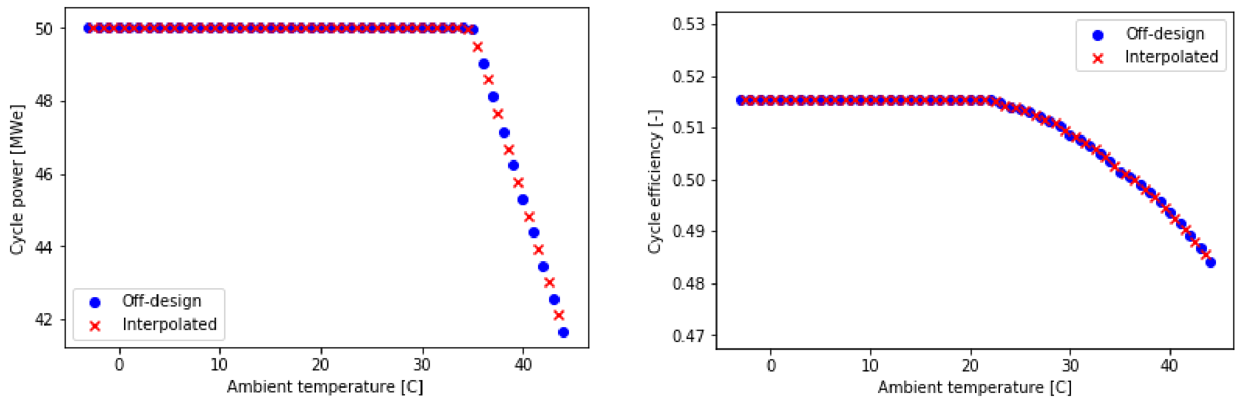

Figure 1: Plots of cycle power and efficiency as a function of ambient temperature for off-design scheme and interpolation scheme

Figure 1 are scatter plots of the cycle power and efficiency for the off-design scheme and interpolation scheme overlaid. The interpolation scheme predicts the performance at temperatures not used in the off-design scheme. Since the interpolation points are between the adjacent off-design points, it shows that the interpolation scheme was correctly implemented.

The performance curves also agree with the expected trends. The cycle power remains constant at about 50 MWe when the ambient temperature is below 35°C. As the ambient temperature rises above 35°C, the cycle power decreases. The cycle efficiency is constant at ambient temperatures of below 21°C, and the efficiency decreases as temperature increases past 21°C.

Monthly Cycle Performance Analysis

Once the interpolation method was built, the performance of the supercritical carbon dioxide power cycle could be analyzed over a year. The first annual analysis was to compare the cycle performance between hot and cold ambient temperatures. The hottest and coldest hour of everyday was extracted from the Daggett annual weather file. The cycle was run at the hottest and coldest hour of the day using the interpolation scheme and the cycle power and heat input were evaluated. This was repeated for every day of the year. The cycle power and heat required were summed for each month of the year. The differences in performance at the different ambient temperatures were normalized for easier comparison.

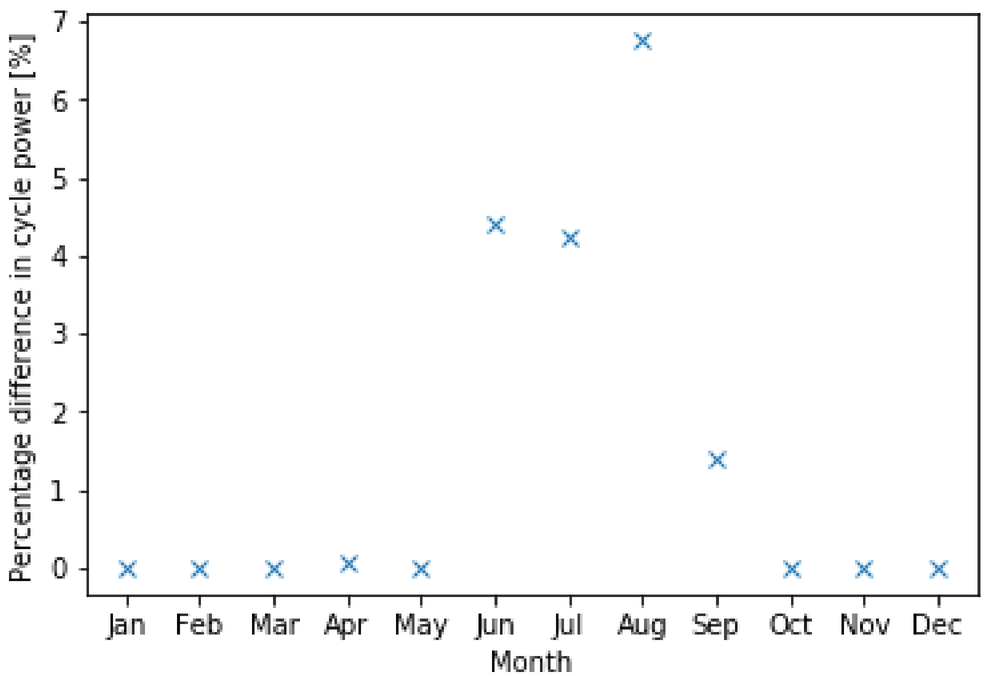

Figure 2: Percentage difference in total monthly cycle power between hottest and coldest hour of the day

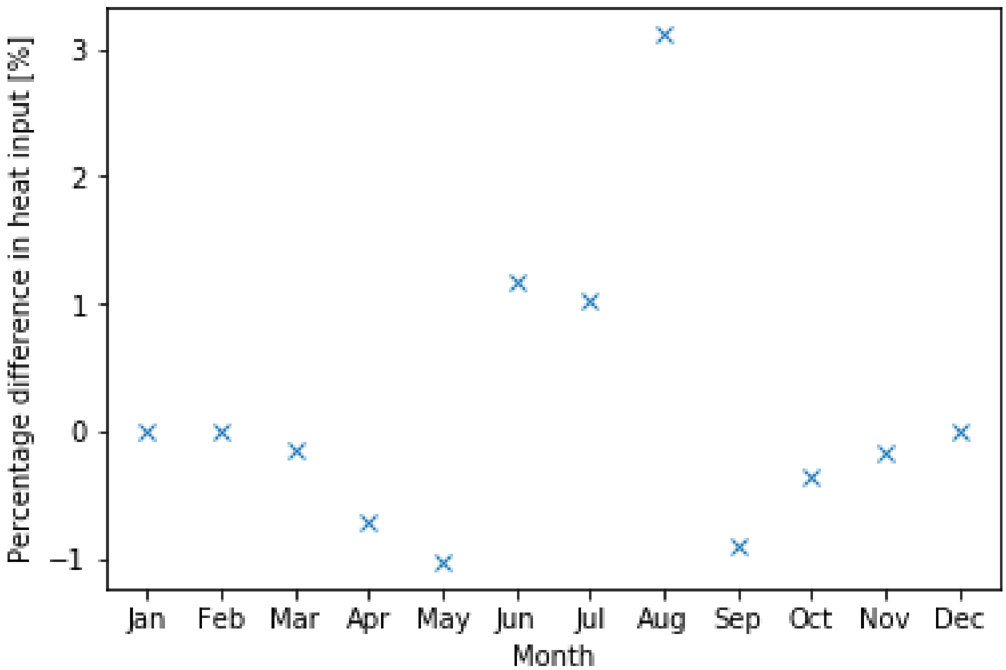

Figure 3: Percentage difference in total monthly required heat between hottest and coldest hour of the day

It can be observed from Figure 2 that the difference in cycle power for the hottest and coldest hour of the day is negligible for the months of January to May and October to December. A similar trend can be observed for heat input from Figure 3. The largest difference in cycle power and heat input exist in the summer months when the ambient temperatures are high.

The performance curves from Figure 1 can be used to explain the trends in Figures 2 and 3. During summer, the highest temperature is above the design point temperature of 35°C and the lowest temperature is below 35°C. So, in the cycle generates significantly more electricity in the coldest ambient temperature than the hottest ambient temperature. Therefore, the difference is cycle power in summer is higher as shown in Figure 2.

In the months of June, July and August, the heat input in the coldest hour of the day is larger than the heat input in the hottest hour of the day. This is expected because the cycle generates more power in the coldest hour of the day than the hottest hour of the day. On the other hand, the heat input in the coldest hour is lower than that in the hottest hour in the months of April, May, September, and November, as shown in Figure 3 with negative normalized differences. This can be explained with the change in efficiency of the cycle with ambient temperature. The hottest hour should never be more efficient than the coldest hour. So, if both cycles are generating the same power output, then the hottest hour will require more heat input. However, when the ambient temperature is hotter than the design ambient temperature, the cycle cannot reach its target power output. In this case, the hottest hour cycle has both a lower efficiency and a lower power output, and the net result can be a lower heat input than the coldest hour cycle.

Comparing Real-time Simulations to Night-time Simulations

The next stage of this project was to compare the performance of the cycle at different times of the day during the months of June, July, and August. The operation of the cycle is modeled by the following stages: use concentrated solar power to heat supercritical carbon dioxide, reject that heat energy at different times of day. The two cases that were analyzed were rejecting the heat energy real time and rejecting the heat starting from midnight. The energy generated by the cycle is of importance because it will aid in doing the cost analysis of each case in the future.

The amount of heat that goes into the supercritical carbon dioxide daily is constant. This heat corresponds to the heat input for the cycle running at it design point ambient temperature of 35°C for 4 hours. This is the amount of energy stored in the supercritical carbon dioxide. The real time simulation starts rejecting the stored heat at the hour before the hottest hour of the day and continues rejecting the heat until no heat energy is available. The nighttime simulation starts rejecting the heat from midnight. From each case, the amount of energy generated by the cycle was calculated. The simulations were repeated for each day in the months of June to August.

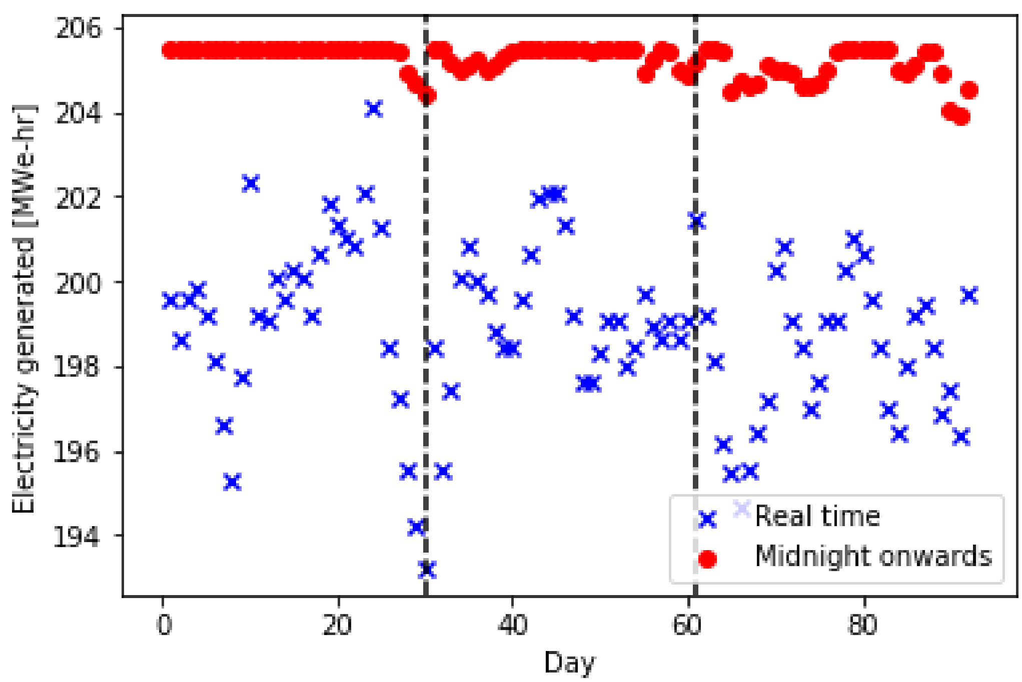

Figure 4: Electricity generated by the supercritical carbon dioxide cycle real time and nighttime.

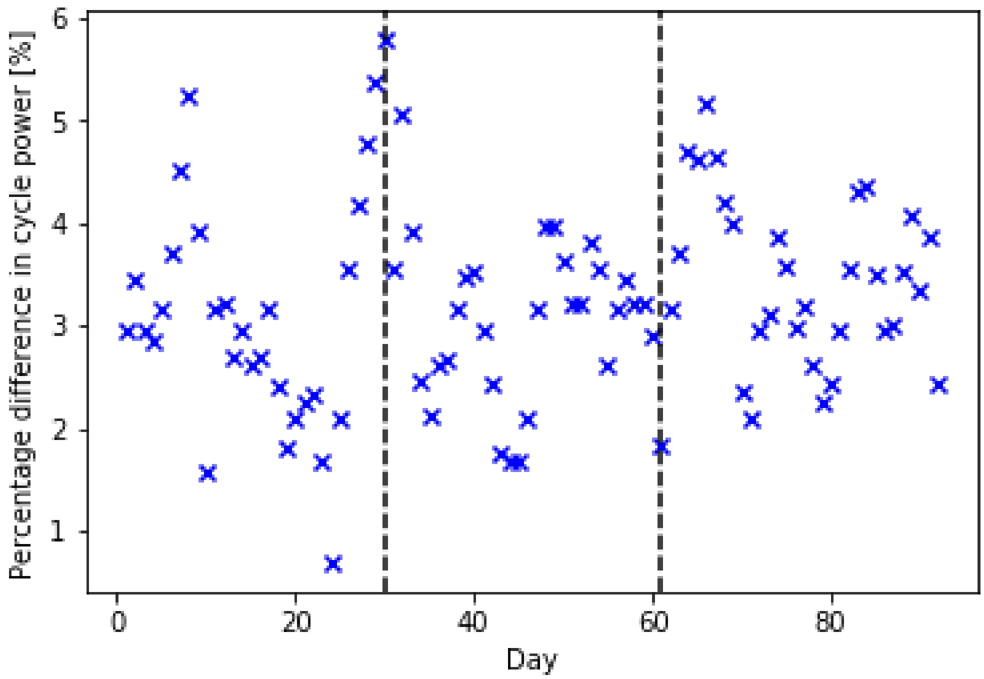

Figure 5: Percentage increase in power generated when rejecting heat starting midnight

Figures 4 and 5 show the results from running the simulations. Based on the simulation results, when the power cycle is run during the day and the heat is rejected at night, on average 3.2% more power is generated. The cycle where heat is rejected at night when ambient temperatures are lower was modeled. This model could be used to investigate the feasibility of night-time heat rejection.

Night-time Heat Rejection Model

The supercritical carbon dioxide power cycle uses concentrated solar power (CSP) as the power source. The CSP heats up molten salt and the heat from the molten salt is transferred to carbon dioxide. Electric power is generated through a Brayton cycle with carbon dioxide as the working fluid. The carbon dioxide that exits the turbine needs to be cooled before being reheated by the molten salt. The basic design of the Brayton cycle rejects heat from carbon dioxide to the atmosphere while the cycle is running. However, based on simulation results, rejecting heat from the carbon dioxide at night when ambient temperatures are lower allows more power to be generated compared to rejecting heat real-time.

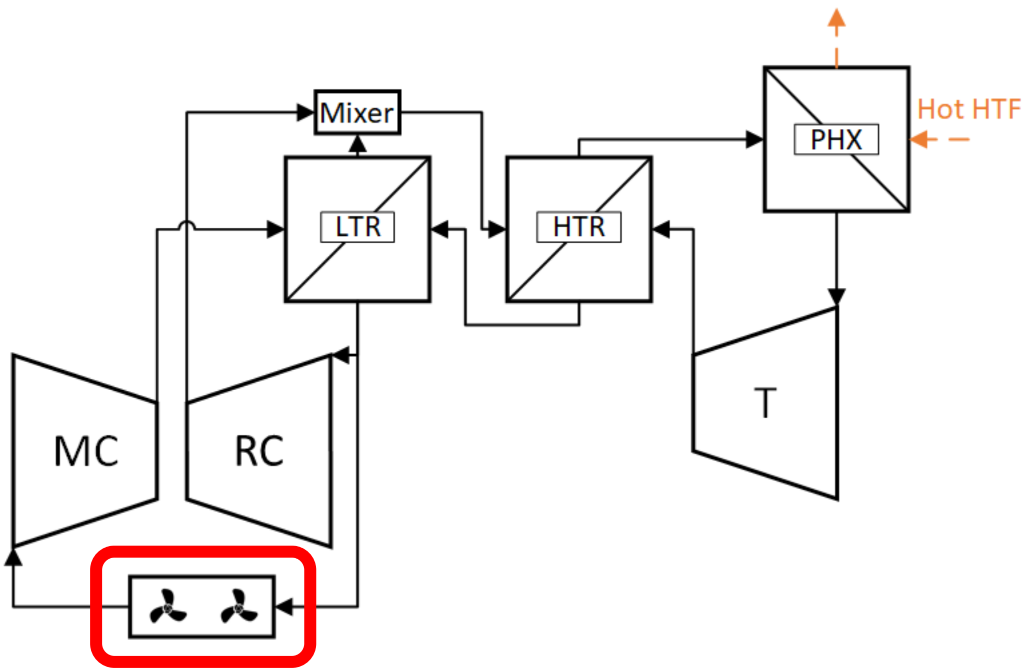

Figure 6: Schematic of power cycle (carbon dioxide to air heat exchanger is in red box)

The schematic of the power cycle is shown in Figure 6. The schematic shows the heat exchangers, turbine, compressors, and recuperators. The heat exchanger through which the residual heat from the carbon dioxide is rejected to the atmosphere is enclosed in the red box. To reject heat in real-time, that heat exchanger is just a crossflow heat exchanger with carbon dioxide and air as the two fluids. To model the night-time heat rejection cycle, that heat exchanger needs to be replaced by two heat exchangers and storage tanks.

To reject heat at night, the heat from the carbon dioxide is transferred to water while the cycle runs during the day. This first heat exchanger is a counterflow heat exchanger. The hot water is stored from the time the cycle runs until it is night. The heat from the stored water is then rejected to the atmosphere at night through a second heat exchanger. This second heat exchanger is a crossflow heat exchanger. A detailed schematic of the process for night-time heat rejection is shown in Figure 7.

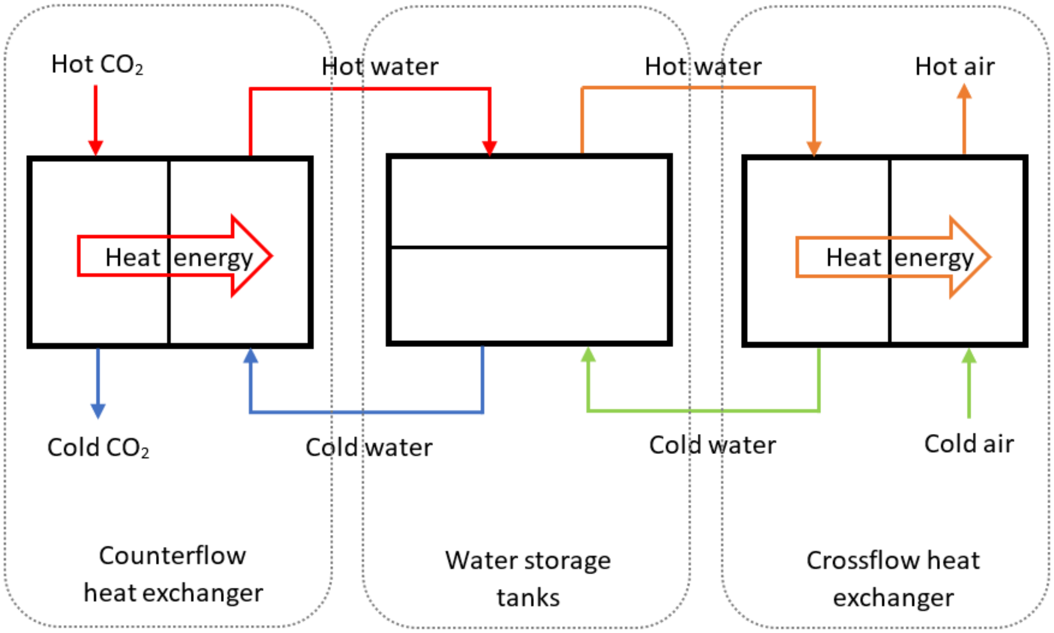

Figure 7: Schematic of the process of night-time heat rejection. Red and blue arrows occur during the day. Orange and green processes occur during the night.

This entire process requires modeling the performance of two heat exchangers. The variables in the analysis of the model are the temperatures of the water exiting the first heat exchanger and air exiting the second heat exchanger. The performance of the heat exchangers is measured in terms of the effectiveness and UA values. The capital cost of night-time heat rejection includes the cost of the two heat exchangers and the water storage tanks. The cost of the heat exchangers is based on their UA values. The objective of the model is to get an estimate of the capital cost of night-time heat rejection and to investigate whether it is economically feasible.

Counterflow Heat Exchanger

The carbon dioxide from the power cycle passes through a counterflow heat exchanger after leaving the turbine and low temperature recuperator. The state and mass flow rate of the carbon dioxide is fixed based on the design conditions of the power cycle. The rate at which heat needs to be rejected is also fixed. Once the carbon dioxide exits this heat exchanger, it enters the main compressor of the power cycle. The only variable in this heat exchanger is the temperature at which the water exits. In this heat exchanger model, there are only four states: the two inlets and two outlets. There are no internal nodes inside the domain of the heat exchanger in this model. Therefore, it is assumed that the pinch point of the heat exchanger is at the inlets or outlets, but not at an internal location in the heat exchanger. The water exit temperature was varied from 74°C to 89°C and the effectiveness, mass flow rate of water, and UA of the heat exchanger were calculated.

After the heat from the carbon dioxide is transferred to the water, the water drains into tanks so that they can be stored until night-time. It was assumed that since the cycle runs for fours hours per day, the amount of water that needs to be stored is the mass of water that flows through this heat exchanger in the four hours. Therefore, the size and number of tanks depends on the mass flow rate of water through the heat exchanger.

Crossflow Heat Exchanger

This heat exchanger runs for four hours at night when ambient temperatures are low. The run duration and total mass of water flowing through both heat exchangers is the same, so the mass flow rate of water in the crossflow heat exchanger is the same as that in the counterflow heat exchanger. It is assumed that the tanks are insulated so that the state of water entering the crossflow heat exchanger is the same as the state of water exiting the counterflow heat exchanger. Conversely, the state of water exiting the crossflow heat exchanger is the same as the state of water entering the counterflow heat exchanger.

The rate of heat transfer through the crossflow heat exchanger is the same as that for the counterflow heat exchanger. The only variable is the temperature of air exiting this heat exchanger. Therefore, the total number of variables is two: temperature of water exiting counterflow heat exchanger and temperature of air exiting crossflow heat exchanger. The temperature of air exiting cross flow heat exchanger was varied from 25°C to 31°C. The temperature of the air could not be increased beyond 31°C because higher exiting air temperature would lead to undefined NTU values.

In the crossflow heat exchanger, the water is unmixed, and air is mixed. For this specification, the NTU is the Equation (1). To yield NTU values that are defined, the argument of the outer logarithm must be greater than 0, giving the inequality in Equation (2).

$$NTU = \ln\left[1 + \frac{\ln(1-\epsilon C_R)}{C_R}\right] \tag{1}$$

$$1 + \frac{\ln(1-\epsilon C_R)}{C_R} > 0 \implies \boxed{\epsilon < \frac{1-e^{-C_R}}{C_R}} \tag{2}$$

If the temperature of the air exceeded 31°C, the inequality in Equation (2) was violated. As a result, temperature of air exiting cross flow heat exchanger could only be varied from 25°C to 31°C. Like the analysis of the counterflow heat exchanger, the effectiveness and UA of this heat exchanger is calculated as a function of the two variable temperatures.

Results and Discussion

Based on the night-time heat rejection model, the additional equipment required are a carbon dioxide to water counterflow heat exchanger, water storage tanks, and a water to air crossflow heat exchanger. The main objective is to investigate the feasibility of rejecting heat at night compared to real-time. Specifically, determine whether the additional power generated from rejecting heat at night compensates for the capital cost for the equipment required for night-time heat rejection. Below are the results of the heat exchanger calculations.

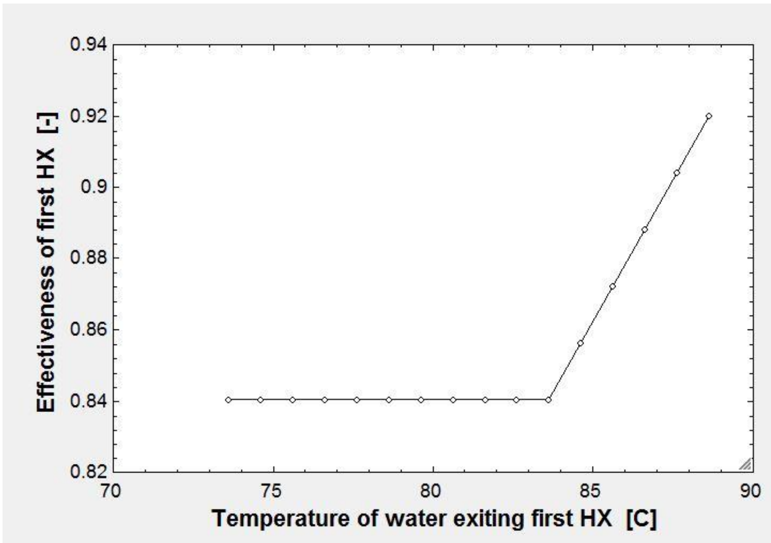

Figure 8: Effectiveness of counterflow heat exchanger as a function of temperature of exiting water

As shown in Figure 8, the effectiveness of counterflow heat exchanger stays constant at 0.84 until the exiting water temperature increases past 84°C. This is explained by the change in value of the minimum capacity rate. When the exiting water temperature is below 84°C, the minimum capacity rate is the capacity rate of the carbon dioxide, which remains constant throughout. When the exiting water temperature is above 84°C, the minimum capacity rate changes to the capacity rate of the water which changes with water temperature. Therefore, the effectiveness of the heat exchanger increases only if the exiting water temperature is higher than 84°C.

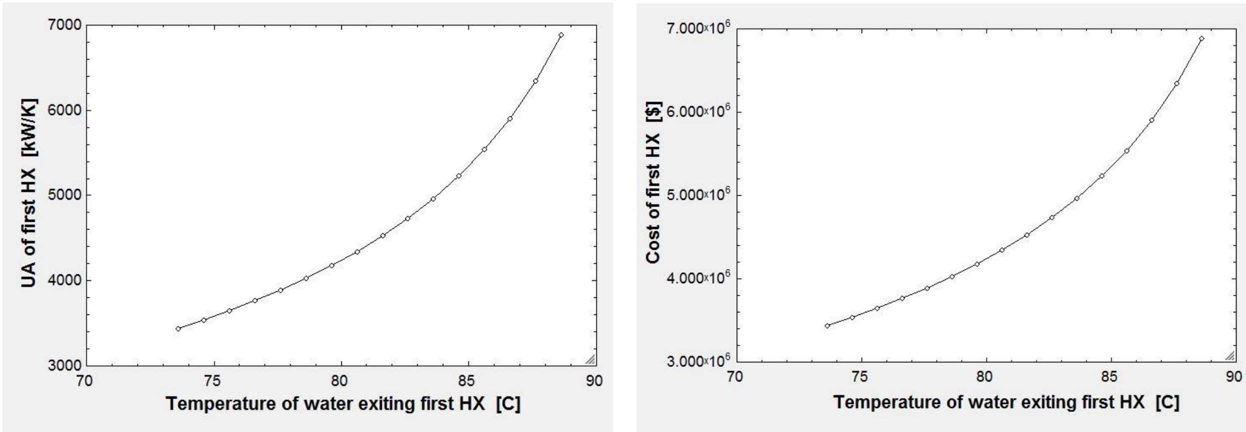

Figure 9: UA and cost of counterflow heat exchanger as a function of temperature of exiting water

The UA of the counterflow heat exchanger was calculated as a function of temperature of exiting water. The UA was then used to calculate the cost of the heat exchanger based on the normalized cost scaling behavior used by Sandia National Labs [1]. To reduce the cost of the counterflow heat exchanger, the temperature of exiting water should be reduced. The volume of water that needs to be stored was also calculated from the mass flow rate of water and is shown in Figure 10.

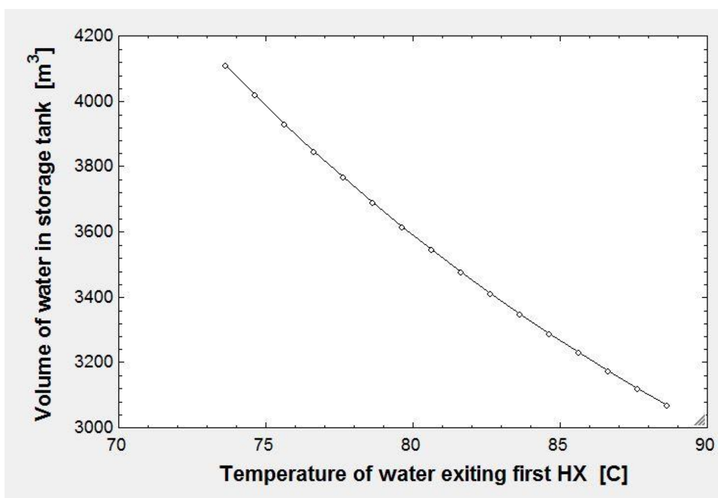

Figure 10: Volume of water that needs to be stored as a function of temperature of exiting water

The results of the crossflow heat exchanger are more complicated than the results of the counterflow heat exchanger because the effectiveness and UA values of the crossflow heat exchanger are functions of both the temperature of water exiting counterflow heat exchanger and the temperature of air exiting cross flow heat exchanger.

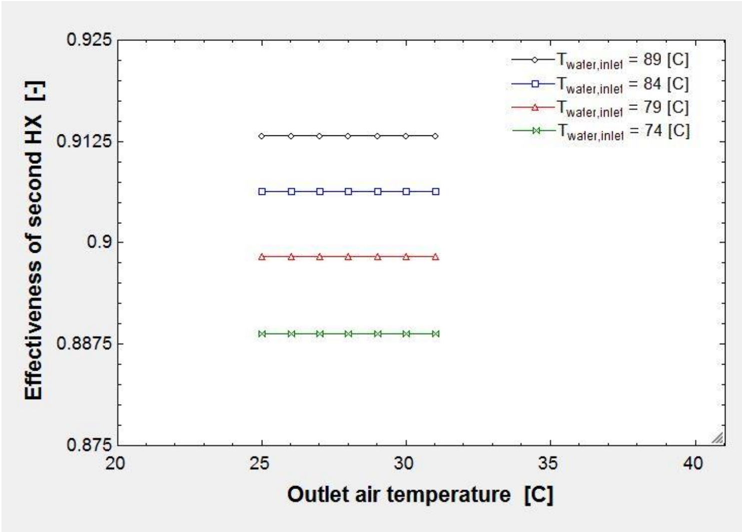

Figure 11: Effectiveness of crossflow heat exchanger as a function of temperature of water and air

It can be observed from Figure 11 that the effectiveness of crossflow heat exchanger changes with the temperature of water, but not the temperature of the air. This indicates that the minimum capacity rate is the capacity rate of water. Figure 12 shows that to reduce the cost of the crossflow heat exchanger, the temperature of the water and air should be reduced.

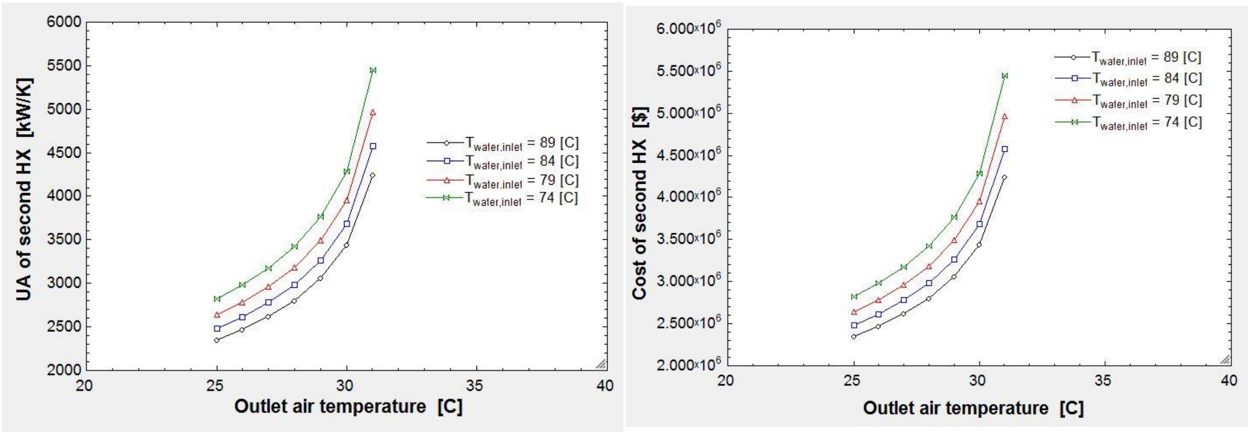

Figure 12: UA and cost of crossflow heat exchanger as a function of temperature of water and air

Through this study, keeping the temperature of water exiting the counterflow heat exchanger at 75°C and the temperature of the air exiting the crossflow heat exchanger at 25°C gave the lowest capital cost. The capital cost of night-time heat rejection, i.e. cost of the two heat exchangers and water storage tanks, is estimated to be about $6.3 million.

Conclusion

The capital cost of night-time heat rejection is estimated to be about $6.3 million. The simulations of the supercritical carbon dioxide power cycle show that night-time heat rejection is only effective from June to August because the daytime and night-time ambient temperatures are significantly different in summer. If it is assumed that night-time heat rejection is implemented in those three months annually, the cycle is run for 4 hours each day, and the electricity generated is sold at the national average od 12 cents/kW-hr, the annual additional revenue from night-time heat rejection is about $70,000. Based on these initial estimates, it would take more than eight decades just to break even with the capital cost. Therefore, from an economic point of view, night-time heat rejection cannot be recommended.

Future Work

The night-time heat rejection model discussed in this report makes some assumptions which might lead to inaccurate conclusions. These assumptions could be tackled in the future of the project. These assumptions occur in modeling the heat exchanger, the economic analysis, and running the power cycle.

One major assumption was that the pinch point of the counterflow heat exchanger was assumed to be either at the inlets or outlets. This assumption could lead to inaccurate calculations for the performance of the heat exchanger because the specific heat capacity of carbon dioxide varies significantly with temperature and pressure. This problem could have been solved if the counterflow heat exchanger was discretized with several nodes in the domain of the heat exchanger. Instead of imposing one overall conservation of energy, imposing the energy balance at every node would incorporate the changing specific heat capacity of carbon dioxide and give more accurate performance metrics.

The cost of the heat exchangers was based on the normalized cost scaling behavior in Carlson’s paper [1]. This paper studied the cost of carbon dioxide to air heat exchangers. The cost of the water to air heat exchanger is expected to be less than the cost of a carbon dioxide to air heat exchanger because the pressure of the carbon dioxide is expected to be significantly larger than the pressure of water. Therefore, a more accurate cost scaling behavior for the crossflow heat exchanger should be investigated.

Some of the minor assumptions are based on how the supercritical carbon dioxide power cycle is run. In this study, it was assumed that the power cycle runs for four hours in a day and the heat is rejected in 4 hours. However, the duration for running the power cycle will depend on the ambient temperature. A more realistic performance profile could be developed so that the power cycle runs in an optimal way not just in summer, but throughout the year.

References

[1] M. D. Carlson, B. M. Middleton, and C. K. Ho, “Techno-Economic Comparison of Solar-Driven SCO2 Brayton Cycles Using Component Cost Models Baselined With Vendor Data and Estimates,” ASME 2017 11th International Conference on Energy Sustainability, 2017.