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# Methodology of Modeling and Comparing the Use of Direct Air-Cooling for a Supercritical Carbon Dioxide Brayton Cycle and a Steam Rankine Cycle

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## Abstract

Dry air-cooling (i.e. cooling with ambient air rather than with water or cooling towers) has become increasingly important for power generation cycles, especially those located in arid regions with restricted water resources. This paper presents a comparison of the use of air-cooling in a supercritical carbon dioxide Brayton cycle to the use of air-cooling in a steam Rankine cycle. These cycles, inherently different in terms of cycle components and operating conditions, are modeled to have equivalent net power output provided by the same external heating and cooling conditions in order to create an analysis that allows proper comparison.

The supercritical carbon dioxide cycle that is considered here is a simple closed-loop Brayton cycle with recuperation and the steam cycle is a Rankine cycle with a single open-feedwater heater. These simple cycle models were chosen over more complex cycle models in order to compare each cycle on the basis of the heat rejection unit without confounding the analysis with details that are related to the rest of the cycle. A recuperator is used in the supercritical carbon dioxide Brayton cycle because the cycle requires regeneration to achieve competitive efficiency. While the steam Rankine cycle does not require any form of regeneration to significantly boost its efficiency, a form of regeneration is added to the cycle by the use of a single open-feedwater heater for consistency.

The comparison is conducted using a life cycle earnings analysis that includes the effect of heat rejection conditions on the cycle efficiency and the air-cooled heat exchanger

capital cost. The air-cooled heat exchanger is modeled using a cross-flow configuration of staggered finned tubes. The results of this comparison show that for the same sized air-cooled heat exchanger, the supercritical carbon dioxide Brayton cycle can achieve lower heat rejection temperatures than the steam Rankine cycle, which corresponds to higher cycle efficiencies. The supercritical carbon dioxide Brayton cycle shows a greater potential to be dry air-cooled than the steam Rankine cycle due to its ability to achieve higher cycle efficiencies while using a smaller, and therefore less expensive, air-cooled heat exchanger.

## Introduction

This paper presents an economic analysis developed to compare two power generation cycles on the basis of their respective heat rejection unit using dry air-cooling. The two cycles that are considered in this analysis are a simple closed-loop Brayton cycle using supercritical carbon dioxide with recuperation and a Rankine cycle using water with one open feedwater heater. In order to create a proper comparison based on each cycle's heat rejection unit, i.e. the supercritical carbon dioxide precooler and the steam condenser, a physical model is needed for both the cycle as well as the heat exchanger itself. This approach allows for the design of the cooling heat exchanger to be defined based on a specified cycle performance, providing a fair comparison of the heat exchanger designs by accounting for the inherent difference between the properties of supercritical carbon dioxide in the precooler and steam in the condenser.

One key difference between the two cycles is the temperature variation of the working fluid that exists in the heat rejection heat exchangers. In the Rankine cycle, the steam in the condenser is going through a phase change and is therefore at a nearly constant temperature for most of the heat rejection process. In the Brayton cycle, the supercritical carbon dioxide enters the precooler at a higher temperature and is sensibly cooled to the cold side temperature without phase change.

The information in this paper is organized into four sections. The first section discusses the supercritical carbon dioxide Brayton cycle and the steam Rankine cycle chosen for this analysis. The second section describes the heat transfer correlations used in modeling the precooler and the condenser. The third section details the cost analysis that is used for determining the life cycle earnings based on the cycle efficiency and the estimated heat rejection unit cost. The final section presents the inputs used in the models as well as the conclusions that are drawn from the life cycle earnings comparison between the two cycles.

## 1. Power Generation Cycles

The power generation cycles modeled in this analysis are discussed in this section. Both cycles were chosen to be of the simplest form that allowed for a single source of regeneration to be included within the cycle. It is not expected that the general conclusions would change if more sophisticated cycles (e.g., the recompression supercritical carbon dioxide cycle or a steam cycle with additional feedwater heaters) were used. The calculations that define each of these cycles are described in more detail by Hruska (2016).

The simple Brayton cycle is shown in Figure 1 and includes a recuperator that allows for the supercritical carbon dioxide to be preheated by the turbine exit flow before entering the primary heat exchanger. The Rankine cycle is shown in Figure 2 and includes an open feedwater heater that allows for the saturated liquid exiting the low pressure pump to be preheated by the high temperature steam exiting the high pressure turbine before being pumped to the boiler. The only pressure loss included in either cycle analysis is related to the cooling unit.

A simple Brayton cycle with recuperation is used to determine the operating conditions associated with the precooler for a given overall cycle performance. Modeling methodology for the closed loop Brayton cycle with recuperation is presented by Dyreby (2014). Each state in the Brayton cycle can be calculated using the model inputs described in Table 1.

A steam Rankine cycle with a single open-feedwater heater is used to determine the inlet and outlet states for the steam air-cooled condenser that provides the same net power production. Figure 2 shows the Rankine cycle with a single open-feedwater heater. Each state in the Rankine cycle is calculated using the model inputs described in Table 2.

With the cycle models defined, the cycle performance is determined. The next step in performing an economic comparison based on the heat rejection unit of these two cycles is to model the heat rejection units themselves in sufficient detail to allow for an estimate of the physical size required for the specified cooling performance.

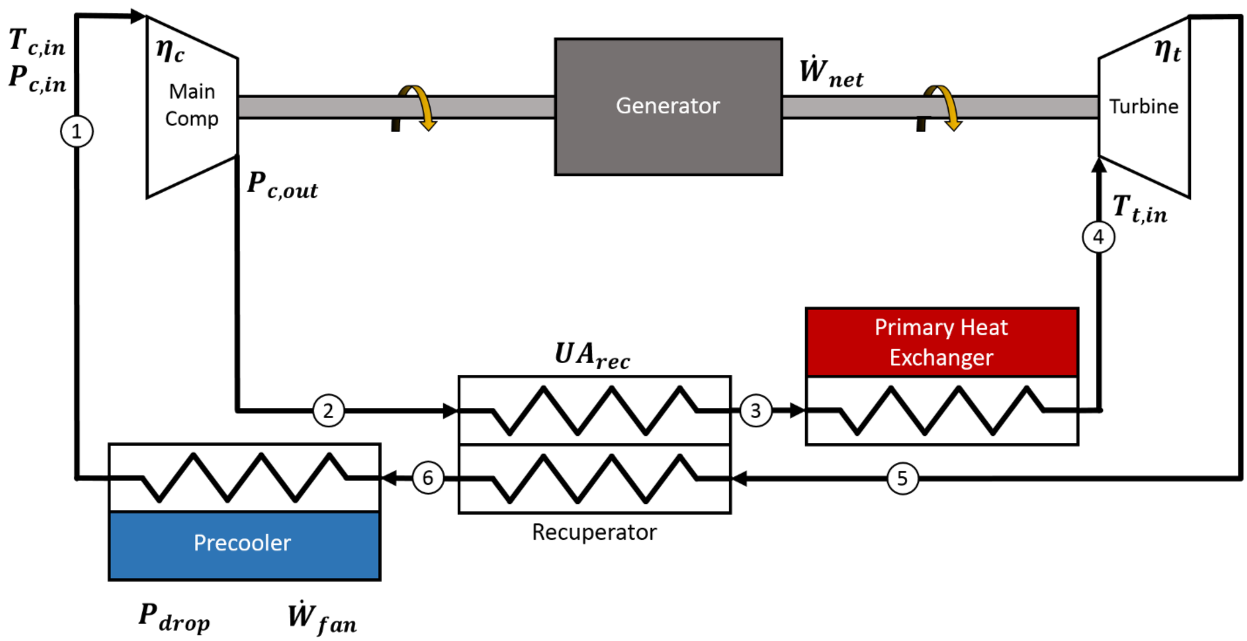


Figure 1. Diagram of the simple Brayton cycle with recuperation. Model inputs shown in bold.

| Model Input Parameters | Description |
|---|---|
| $T_{c,in}$ | Main Compressor Inlet Temperature (Low Side Temperature) |
| $P_{c,in}$ | Main Compressor Inlet Pressure |
| $\eta_c$ | Main Compressor Isentropic Efficiency |
| $P_{c,out}$ | Main Compressor Outlet Pressure |
| $UA_{rec}$ | Recuperator Conductance |
| $T_{t,in}$ | Turbine Inlet Temperature (High Side Temperature) |
| $\eta_t$ | Turbine Isentropic Efficiency |
| $P_{drop}$ | Allowable Precooler Pressure Drop |
| $\dot{W}_{fan}$ | Required Fan Power |
| $\dot{W}_{net}$ | Total Net Power |

Table 1. Brayton cycle model input parameters with descriptions.

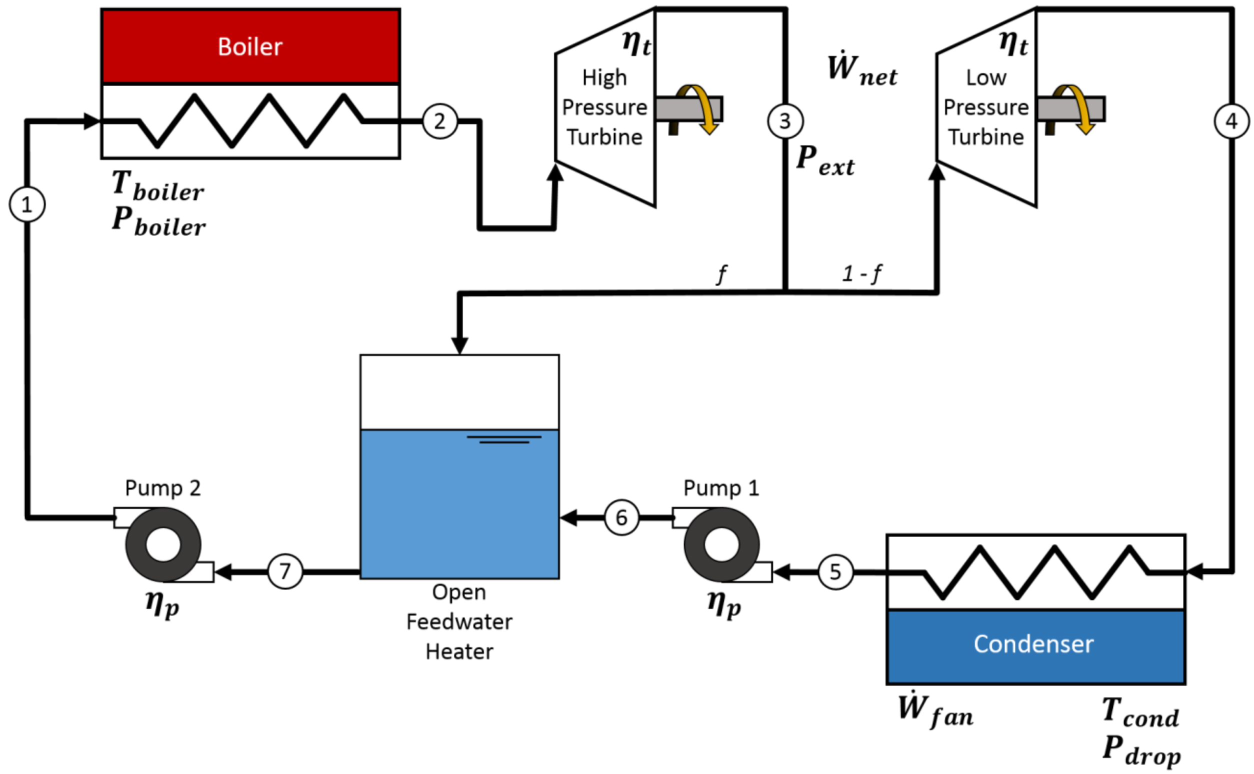


Figure 2. Diagram of the Rankine cycle with open feedwater heater. Model inputs shown in bold.

| Model Input Parameters | Description |
|---|---|
| $T_{boiler}$ | Boiler Temperature<br>(High Side Temperature) |
| $P_{boiler}$ | Boiler Pressure |
| $\eta_t$ | Turbine Isentropic Efficiency |
| $P_{ext}$ | Feedwater Heater Extraction Pressure |
| $T_{cond}$ | Condensing Temperature<br>(Low Side Temperature) |
| $P_{drop}$ | Allowable Condenser Pressure Drop |
| $\eta_p$ | Pump Isentropic Efficiency |
| $\dot{W}_{fan}$ | Required Fan Power |
| $\dot{W}_{net}$ | Total Net Power |

Table 2. Rankine cycle model input parameters with descriptions.

## 2. Air-Cooled Heat Exchanger Model Methodology

The supercritical carbon dioxide precooler and the steam condenser are modeled as a finned tube crossflow heat exchanger using the Engineering Equation Solver (EES) software and a heat exchanger modeling methodology described by Nellis and Klein (2009). This section describes the method used to determine the physical size of an air-cooled heat exchanger from the given performance requirements. This process is used for both the precooler in the Brayton cycle and the condenser in the Rankine cycle. The motivation for determining the heat exchanger size for a specified performance is to ultimately create a single model that is capable of predicting heat exchanger size and pumping costs for a cooling system that can be applied to/designed for either cycle. This approach allows for a self-consistent life cycle analysis to be conducted on both cycles in order to compare overall life cycle earnings. The calculations and equations used to in this analysis are described in more detail by Hruska (2016).

The methodology used for modeling the heat exchangers is based on calculating the total conductance required by the heat exchanger effectiveness associated with the cycle constraints and then using the conductance to determine the required thermal resistance and, therefore, size. This method connects the physical size that is used in the economic analysis with the required heat rejection performance within either cycle. The heat exchanger model is directly integrated with the cycle models, as described in Section 1, in order to investigate the effect of the size of the cooling unit and the required fan power on the efficiency of the cycles and the cost of these units.

The parameters needed in the heat exchanger model are the same for both the precooler model and the condenser model. Table 3 lists the necessary inputs for the heat exchanger model. The term 'working' fluid is used to represent supercritical carbon dioxide for the precooler model and steam for the condenser model.

Both the steam condenser and the supercritical carbon dioxide precooler models begin by determining all of the air side characteristics needed for the heat exchanger analysis. The key parameters on the air side are the air side heat transfer coefficient and the pressure drop. The heat transfer coefficient and the pressure drop are both determined using the compact heat exchanger library in EES that facilitates the use of the correlations based on experimental data presented by Kays and London (1984). These correlations use the geometry of the heat exchanger, the geometry of the finned tubes, as well as the air flow properties and characteristics in order to determine the air side heat transfer coefficient and the pressure drop. The pressure drop is used to calculate total fan power. The fan power is considered in the overall cycle models and the economic analysis.

| Model Input | Description |
|---|---|
| $T_{air,in}$ | Ambient Air Temperature |
| $P_{air}$ | Ambient Pressure |
| $\dot{W}_{fan}$ | Fan Power |
| $\eta_{fan}$ | Fan Efficiency |
| $\Delta P_{air}$ | Air Side Pressure Drop |
| $CHX\$$ | Compact Heat Exchanger Geometry |
| $th_{tubes,min}$ | Minimum Tube Thickness |
| $P_{drop}$ | Allowable Pressure Drop |
| $\dot{m}_c$ | Working Fluid Mass Flow Rate |
| $T_{c,in}$ | Working Fluid Inlet Temperature |
| $\dot{Q}$ | Cooling Capacity |
| $T_{c,out}$ | Working Fluid Outlet Temperature |
| $P_{c,in}$ | Supercritical Carbon Dioxide Inlet Pressure |

Table 3. Air-cooled heat exchanger model inputs with descriptions.

The next step in the models is to determine the internal flow correlations. The steam condenser model uses two EES library procedures that correlate the two phase steam flow through the tubes to provide an average heat transfer coefficient and a total pressure drop on the steam side. The average heat transfer coefficient is calculated using the correlation described by Dobson and Chato (1998) for condensing two phase flow through a horizontal tube. The total pressure drop of the steam is found using the sum of the frictional pressure drop and the momentum pressure drop. The momentum pressure drop is calculated using the drift flux model as described by Ould Didi et al. (2002).

The supercritical carbon dioxide precooler uses a sub-heat exchanger technique, as described by Nellis and Klein (2009), in order to account for the temperature-dependent properties of the supercritical carbon dioxide as it is cooled in the precooler. This technique breaks the precooler into smaller heat exchangers that each assume constant properties. Within each sub-heat exchanger, the average heat transfer coefficient and the pressure drop are calculated. The heat transfer coefficient is calculated using the Nusselt number correlation for fully developed, turbulent flow provided by the Gnielinski correlation (1976). The pressure drop is calculated from the friction factor correlation developed by Zigrang and Sylvester (1982). The total pressure drop is determined as the sum of the pressure drops in each sub-heat exchanger.

The final step in both heat exchanger models is to calculate the total conductance of the heat exchanger using two methods. The first method relates the conductance to the overall performance of the heat exchanger using an effectiveness and number of transfer units methodology. The effectiveness is defined as the cooling capacity normalized by the maximum possible heat transfer rate of the heat exchanger. The number of transfer units is a function of the effectiveness, the heat exchanger flow configuration, and the ratio of the capacitance rates of the two fluids. The conductance is defined as the product of the number of transfer units and the minimum capacitance rate through the heat exchanger. The second method relates the conductance to the physical size of the heat exchanger. This is done by using a resistance network between the air flow and the working fluid. The resistances considered in this analysis are the resistance to convection from the air to the finned tubes using the air side heat transfer coefficient and a fin efficiency based on the geometry of the finned tubes, the resistance to conduction through the tubes based on conduction through an annular tube, and the resistance to convection from the working fluid to the tubes using the internal heat transfer coefficient. The conductance from the resistance network is defined as the inverse of the total resistance. The overall size of the heat exchanger is automatically adjusted until these two methods provide the same conductance value.

## 3. Economic Considerations

Economic analyses of the supercritical carbon dioxide Brayton cycle and the steam Rankine cycle are required in order to perform a life cycle earnings analysis. This economic analysis includes the earnings from generating electricity, the cost of providing the thermal energy to the cycle, and the capital cost of the cooling heat exchanger. This economic breakdown does not include machinery costs for either cycle or any maintenance costs. The purpose of this analysis is to gain an understanding of the possible benefits of using dry air-cooling in a supercritical carbon dioxide Brayton cycle over a steam Rankine cycle and therefore it is important that a common set of economic assumptions be used to examine each cycle.

The life cycle analysis is based on the first year fuel earnings, i.e. the difference between the cost of selling electricity and the cost of buying natural gas as fuel projected over the life of the power plant, as well as the initial cost of the heat rejection unit. Table 4 shows the inputs used in the economic analysis of the cycle performance.

The methodology used in this analysis is the P1-P2 method presented by Duffie and Beckman (2013). The P1-P2 method is used to determine the life cycle earnings of each cycle and heat exchanger combination. The life cycle earnings is defined as the product of the P1 parameter and the first year earnings minus the product of the P2 parameter and the capital cost of the cooling heat exchanger. The P1 parameter is the present worth of the fuel cost over the period of the analysis while accounting for the fuel inflation rate.

The P2 parameter is the ratio of the life cycle costs due to any additional capital investment to the initial investment. For this analysis, the P2 parameter is taken to be unity, i.e. assuming the heat exchanger unit would be completely purchased at the beginning of the analysis with no tax credits. Maintenance costs are neglected.

| Model Input | Description |
| --- | --- |
| $N$ | Number of Years for Analysis |
| $i$ | Fuel Costs Inflation Rate |
| $d$ | Market Discount Rate |
| $\dot{W}_{net}$ | Total Net Power |
| $\eta$ | Cycle Efficiency |
| $C_{electric}$ | Selling Cost of Electricity |
| $C_{thermal}$ | Buying Cost of Thermal Energy |

Table 4. Economic analysis model parameters.

The sole investment taken into account in this analysis is the initial cost of the cooling heat exchanger. Using only the cost of the cooling heat exchanger gives an appropriate estimate for the relative cost of the heat exchanger compared to cycle earnings. Lower cold side temperatures allow the cycle to achieve higher efficiencies. However, the higher efficiency comes at the price of larger cooling heat exchangers in order to reach the lower temperatures. This analysis focuses on the optimization of the cooling heat exchanger to maximize cycle earnings.

A model for the cost of the heat exchanger was determined from predicting the overall cost of the heat exchanger from tubing and fin material costs. The tubing material cost uses a best fit correlation developed based on tubing costs from OnlineMetals for SS304L tubing. The correlation shown in Figure 3 relates the cost per inch of tubing to the cross sectional area of tubing material.

$$C_{tubing} = 4.1163\left(A_{tubing}\right)^{0.6349} \tag{1}$$

where $C_{tubing}$ is the cost ($) per inch of tubing and $A_{tubing}$ is the cross sectional area of tubing material.

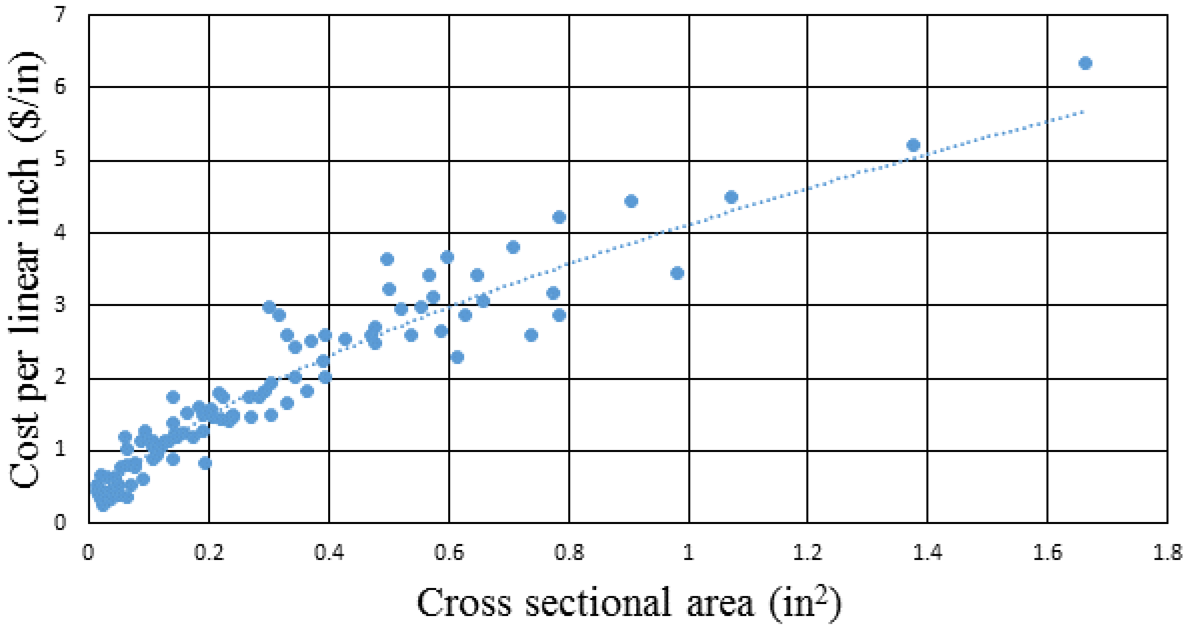


Figure 3. Cost of tubing material per unit length plotted versus cross sectional area.

The fin material cost uses a correlation developed based on sheet metal costs from OnlineMetals for sheet aluminum. The correlation shown in Figure 4 relates the cost per square inch of finned surface area to the fin thickness

$$C_{fins} = 0.5576\, fin_{thk} + 0.0352 \qquad (2)$$

where $C_{fins}$ is the cost per square inch of finned surface area.

The estimated cost of the heat exchanger is calculated using a power law model developed from obtaining two quotes for air-cooled heat exchangers. The power law fit adjusts the material prices based on the quoted heat exchangers in order to obtain an actual cost for the modeled heat exchanger from the predicted cost.

$$C_{HX} = 2.887\left(C_{HX,p}\right)^{0.8297} \qquad (3)$$

where $C_{HX}$ is the actual cost of the air-cooler heat exchanger and $C_{HX,p}$ is the predicted cost from the base materials.

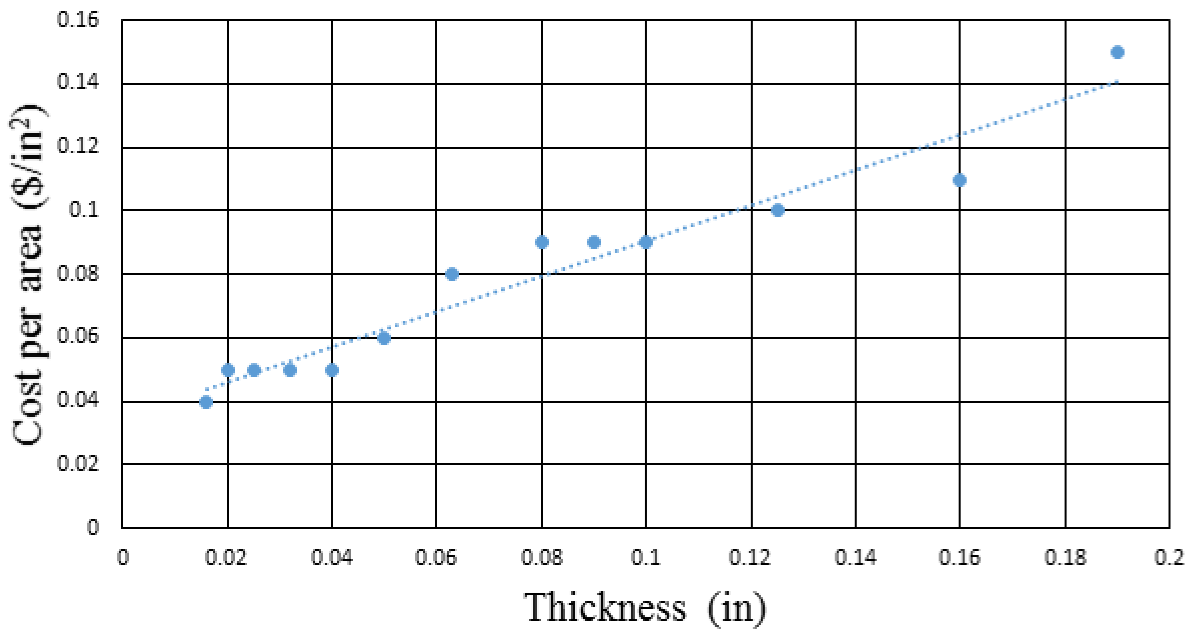


Figure 4. Cost of fin material per unit square area plotted versus sheet thickness.

## 4. **Life Cycle Earnings Optimization**

The life cycle earnings analysis is designed to optimize the life cycle earnings for the given cycle conditions. The life cycle earnings is calculated from the first year earnings and the air-cooled heat exchanger capital cost as explained in section 3. The first year earnings is proportional to the cycle efficiency and the air-cooled heat exchanger capital cost is related to the physical size of the heat rejection unit.

The Brayton and Rankine cycles are inherently different in terms of components as well as where the cycle takes place compared to the fluid vapor dome. In order to create an analysis that properly compares these two cycles, it was assumed that the cycles were provided with the same external heating and cooling temperatures. The cycle model input values that were used in this analysis are shown in Table 5.

Both cycles are modeled to have a net power output of 10 MW using the same 700°C heat source and using 30°C air as the cooling sink. The heat source is assumed to bring the hottest temperature in the cycle, i.e. the turbine inlet temperature or the boiler temperature, completely to the heat source temperature. The difference between the coldest temperature in the cycle, i.e. the supercritical carbon dioxide compressor inlet temperature or the steam condensing temperature, and the ambient temperature (30°C) is varied in order to analyze the effect of the cold temperature on the life cycle analysis. The term 'cold temperature' is used throughout this section in place of the compressor inlet temperature for the Brayton cycle or the condensing temperature for the Rankine cycle. Figure 5 uses a value of 40°C for the cold temperature.

| | Brayton Cycle | Rankine Cycle |
|---|---|---|
| **Working Fluid** | Supercritical Carbon Dioxide | Steam |
| **Compact Heat Exchanger** | Finned circular tubes, surface CF-7.34 | |
| **Cycle Inputs** | | |
| Total Net Power | 10 MW | |
| Hot Temperature | 700° C | |
| High Side Pressure | 25 MPa | 22 MPa |
| Cold Temperature | Variable | |
| Low Side Pressure | 8 MPa | Saturation |
| Turbine Efficiency | 85% | |
| Compressor/Pump Efficiency | 85% | 60% |
| Recuperator Conductance | 1500 kW/K | - |
| Extraction Pressure | - | 1.9 MPa |
| Cooler Pressure Drop | 2% | |
| **Air Side Inputs** | | |
| Ambient Temperature | 30° C | |
| Ambient Pressure | 1 atm | |
| Fan Power | Optimized (LCE) | |
| Fan Efficiency | 50% | |
| Pressure Drop | 200 Pa | |
| **Economic Parameters** | | |
| Number of Years for Analysis | 5 | |
| Fuel Inflation Rate | 2% | |
| Market Discount Rate | 3.25% | |
| Cost of Electricity | 0.05 $/kW-hr | |
| Cost of Thermal Energy | 0.465 $/therm | |

Table 5. Cycle model input values used for the cycle comparison analysis.

The Brayton cycle is limited to having a high side pressure of 25 MPa while the Rankine cycle is limited to a high side pressure of 22 MPa. The Brayton cycle limitation is chosen based on the analysis from Dyreby (2014). The Rankine cycle limitation is chosen to be close to the Brayton cycle limit without exceeding the supercritical pressure of steam (22.12 MPa). The low side pressure for the Brayton cycle, i.e. the compressor inlet pressure, is 8 MPa, unless stated otherwise. The low side pressure for the Rankine cycle is the saturation pressure determined by the condensing temperature of the steam.

The recuperator conductance for the Brayton cycle is chosen to be 1500 kW/K in order to be consistent with the simple Brayton cycle designs of interest discussed by Dyreby (2014). Low values of conductance result in low cycle efficiency and first year earnings. Higher values of conductance have less of an effect on the overall life cycle analysis as the dependence on conductance is asymptotic as discussed by Hruska (2016). The extraction pressure for the open feedwater heater in the Rankine cycle is set to be the saturation pressure at the average of the boiling and condensing saturation temperatures.

The fan power is used as an optimization variable in the analysis. The fan power is calculated to produce an optimal heat exchanger size while being constrained to the 200 Pa air side pressure drop. A larger fan power reduces the required physical size of the air-cooled heat exchanger because of higher air side heat transfer coefficients, but decreases the overall efficiency of the cycle. This tradeoff results in an optimal value of fan power in life cycle earnings, as shown in Figure 5.

The number of years for the life cycle analysis is chosen to be 5 years. Increasing the number of years for the analysis results in the optimal life cycle earnings shifting to a lower optimal cold side temperature (i.e. requiring a larger heat exchanger) as described by Hruska (2016). This shift is due to the longer analysis periods favoring a larger initial capital cost of the cooling heat exchanger, which results in a higher cycle efficiency.

The compressor inlet pressure in the Brayton cycle has a significant effect on the life cycle analysis of the Brayton cycle because the properties of carbon dioxide vary significantly near the supercritical point (7.39 MPa, 304.25 K). The effect of the varying properties on the life cycle earnings analysis is shown in Figure 6.

The life cycle earnings for the 5 year period is similar to the first year cycle earnings. The effect of the capital cost of the air-cooled heat exchanger is not very large because the ratio of the capital cost to the yearly earnings is small. The overall effect of the capital cost of the cooling heat exchanger shifts the peak life cycle earnings to a slightly higher cold temperature than the peak efficiency temperature as shown in Figure 6. Shorter life cycle analysis periods show the heat exchanger capital cost having a larger effect.

The first year earnings is directly proportional to the cycle efficiency. The largest efficiency for the Brayton cycle occurs at the largest pressure ratio. Since the high side pressure is limited to be 25 MPa, the largest efficiency occurs at 7.5 MPa. However, even though the higher efficiency occurs at 7.5 MPa at colder temperatures, the optimal low side pressure begins to increase as the cold temperature increases. This effect is shown in the middle plot of Figure 6, as the cold temperature increases the larger the optimal compressor inlet pressure. The Rankine cycle efficiency continues to increase as the condensing temperature decreases.

The capital cost of the heat exchanger is directly proportional to the physical size of the heat exchanger. Colder temperatures, i.e. a smaller temperature difference relative to the ambient air, requires a larger heat exchanger. This trend can be seen on the bottom plot in Figure 6. The main effects of the low side pressure on the heat exchanger volume in the Brayton cycle are attributed to the variation in the specific heat of the supercritical carbon dioxide as well as the effect of the efficiency on the cycle. Higher efficiencies for the cycle will lead to less cooling being required and therefore a smaller cooling heat exchanger. The Rankine cycle condenser is primarily affected by the condensing temperature.

The peak life cycle earnings for the Brayton cycle is $5,061,000 occurring at 7.5 MPa and 36°C. The peak life cycle earnings for the Rankine cycle is $4,237,000 occurring at 49°C. The peak life cycle earnings for the Brayton cycle is 19% larger than the peak life cycle earnings of the Rankine cycle. This difference is due to the Brayton cycle requiring a smaller cooling heat exchanger that can provide lower temperature operation and therefore higher efficiencies.

The specific finned tube geometry is also considered in this analysis; however changing the air-side heat transfer geometry did not have a strong effect on the general trends that can be seen in Figure 6. It was found, though, that the optimal finned tube geometry was different for the supercritical carbon dioxide precooler than for the steam condenser. The precooler in the supercritical carbon dioxide cycle tended to optimize towards smaller tube diameters while the condenser in the steam cycle tended towards larger diameter tubes. The full effect of the selection of the finned tube geometry is described by Hruska (2016).

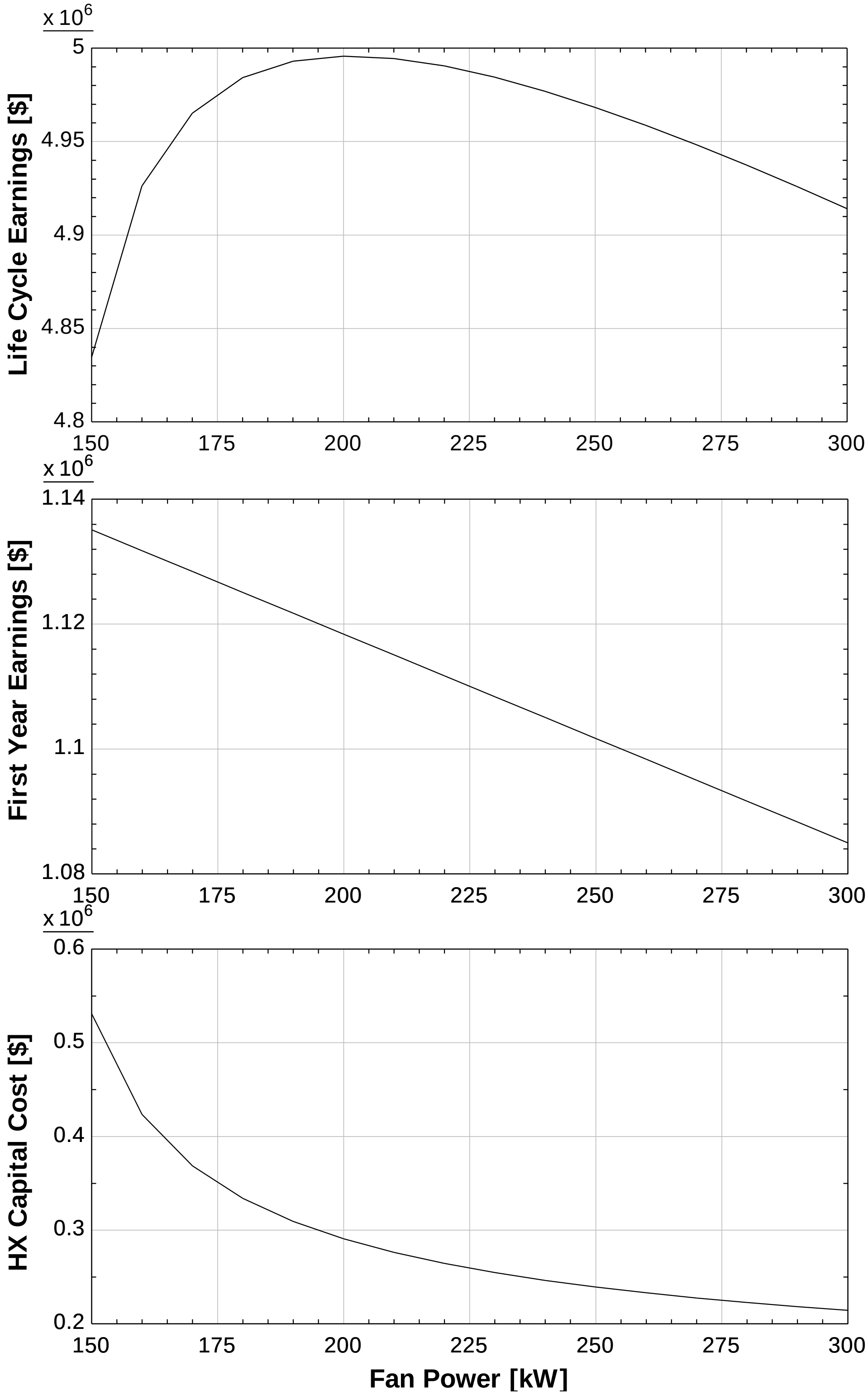


Figure 5. Effect of fan power on the life cycle earnings analysis for the Brayton cycle.

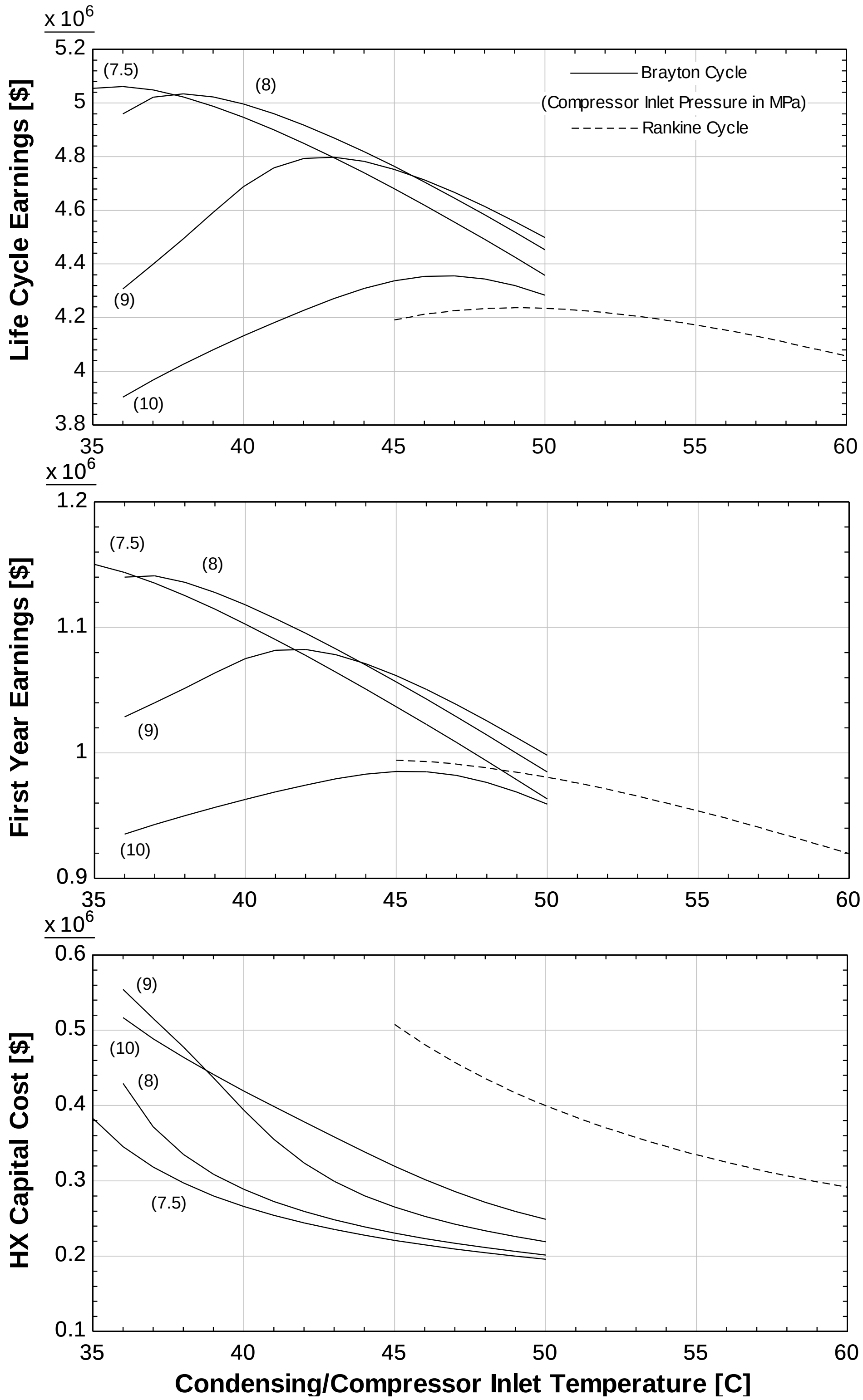


Figure 6. Cost breakdown plotted versus condensing temperature and compressor inlet temperature for different compressor inlet pressures. Compressor inlet pressure in MPa shown in parentheses.

## Summary and Conclusions

A cycle model was created for both a supercritical carbon dioxide Brayton cycle and a steam Rankine cycle. In addition, a detailed cross-flow heat exchanger was modeled to examine the possibility of dry air-cooling both types of cycles. An economic comparison was conducted to examine the impact of dry air-cooling each cycle using a life cycle earnings analysis. The life cycle earnings optimization balanced the overall cycle efficiency against the cooling heat exchanger capital cost. The supercritical carbon dioxide Brayton cycle and steam Rankine cycle that were considered both contained the minimum number of components to achieve a simple, regenerated cycle. The cycles each produced the same electrical output from an identical heat source and cooling source in order to create a foundation for a fair comparison between the power generation cycles.

In conclusion, the supercritical carbon dioxide Brayton cycle is superior to the steam Rankine cycle for use with dry air-cooling in terms of both the cycle efficiency and the required physical size of the cooling heat exchanger. The physical size of the cooling heat exchanger is naturally smaller for the supercritical carbon dioxide precooler than the steam condenser due to the larger temperature difference between the fluid and the ambient air throughout the heat exchanger. This effect allows for lower temperatures to be reached in the supercritical carbon dioxide precooler compared to a physically equal sized steam condenser and also reduces the fan power; lower temperatures leads to higher cycle efficiencies and overall higher life cycle earnings.

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