Executive Summary

Senior design Team 16 is comprised of three mechanical engineers and one chemical engineering: Karl Falk, Isaac Kuiper, Chris Newhuis and Marc Loffert, respectively. We are one of three subteams looking into the feasibility of cleaner coal power generation.

Team 16 is proposing that an economical, and environmental analysis be done on a standard coal power plant and compared against a clean coal power plant. We have compared the standard power plant to two clean technology alternatives; an Integrated Gasification Combined Cycle with carbon sequestration and a Pulverized Coal Oxygen Fed plant with carbon sequestration. In order to keep a fair comparison, each of these plants was designed to produce 1 GW of power.

Towards this purpose, we have identified the key components of each process. Physical components, such as the boiler, pulverizers, turbines, etc. have been analyzed to define key variables around them that have allowed us to make informed estimates of dimensions and operating conditions. From this, we have estimated costs of material, using values taken from current market values of commonly produced machinery or empirical estimates for construction of the system. Scale-up has been used to estimate costs of operating, installation, and other variable costs of construction.

Once we acquired costs, we performed a comparison between the different power plant varieties. It is assumed that the IGCC and the Oxygen Fed plants will be more expensive, but our study will give a value of how much more it will cost the customer for a power plant to switch to an economically safe option.

Communication with Team 14 was necessary to determine the necessary environmental equipment to remove pollutants from the effluent gas produced by the plant.

Our subgroup (16) in particular is focusing on the energy cycles for the power plants: Boiler steam loops, turbines, pre-processing of the coal, etc. Equipment choices are based on efficency as well as cost, with the purpose of being cost effective. We are performing optimizations based on both initial and long term costs. Now that a final design is made, efficiency is determined. The thermodynamic design of these plants included two reheats with regenerations in the Rankine Cycle. This provided thermal efficiencies of 44% for the conventional plant, and 45% for the Oxy-Coal plant. Both of these plants use over 3 million short tons of coal per year.

Recent challenges have arisen in the coal power industry due to recent EPA restrictions. These restrictions have limited emissions from coal generating power plants to 1 lb of CO2 for every kWh of energy produced. This requires any new power plant to utilize carbon capture technology. Our designs are expected to emit 2.8 lb/kWh for the conventional case, and 1.8 lb/kWh for the Oxy-Coal case, before the implementation of carbon capture.

Using our cash flow analysis, we were able to determine the capital cost of the conventional plant was $2.3 Billion, while the oxygen case costed $3.2 Billion. This information, along with the price of the fuel, we were able to determine the cost to produce electricity. This was 0.07 $/kWh for the conventional plant, and 0.09 $/kWh for the Oxy-Coal plant. Comparing this to 0.18 $/kWh for the IGCC plant, we were able to find the final cost to produce cleaner energy using coal as a fuel.
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2012, Calvin College and Karl Falk, Isaac Kuiper, Marc Loffert, Chris Newhuis
1 Introduction

1.1 Problem Definition

Since the industrial revolution, the coal power industry has emitted high levels of pollutants such as mercury, sulfur oxides, carbon dioxide, and particulate matter. These pollutants have had a negative effect on the earth’s climate, human health, and have contributed to occurrences of acid rain. According to the National Oceanic & Atmospheric Administration research lab at Mauna Loa the carbon dioxide levels in the atmosphere has been on the rise for the past fifty years.\(^1\) The current level is at 388 ppm up from 315 ppm in 1960.

There are many contributing factors for this high level, but one of the largest factors is power production by coal fired power plants. According to the Environmental Protection Agency fossil fueled power plants are responsible for 40% of manmade carbon dioxide emissions in the United States.\(^2\) In the past, these emissions were just an afterthought but we are now beginning to understand that they are having a significant negative impact on the local and global ecosystems.

Coal power may contribute greatly to pollution levels, but coal is currently an abundant resource in the US. Coal will be used to generate power as long as it is inexpensive to obtain and use.

In an effort to help reduce carbon dioxide emission by coal power plants, there are new clean coal technologies being developed. Some of these technologies are carbon dioxide scrubbers located within the flue stacks, syngas turbines and coal burned with an oxygen feed. Although these new technologies are effective at reducing emissions, they also have a higher cost due to additional processing in the plant. None the less, these technologies are still being considered when new power plants are being designed. This is because of the possible implementation of a carbon tax on large industries. These carbon taxes could quite possibly be very costly and expensive for any existing coal power plant. In order to help power plants to be as profitable as possible plant designers will have to weigh the costs of implementing technology to reduce emissions.

1.2 Project Statement

The goal of the Clean Coal project is to compare and contrast a Conventional Coal power plant with both an Internal Gasification Combined Cycle (IGCC) power plant and an Oxygen Fed power plant on an economic and environmental basis. These tasks were divided up into three different sub groups, each responsible for a different aspect of the designs. The first group, Team 14, is the environmental group. This group is in charge of capturing and sequestering the carbon dioxide, as well as removing other hazardous compounds. The second group, Team 15, is in charge of simulating the processes of the IGCC

\(^1\) (Tans)

\(^2\) (Environmental Protection Agency, 2007)
plant. Their main focus is to model the gasification process. Finally, our group is in charge of modeling the thermodynamic cycle for both the conventional coal and oxygen fed power plant.

We are also working with two geology students, who are our plant location consultants. These students have researched coal varieties, to determine what type of coal we should use, and what impurities will be inherent in the fuel. They have also informed us of a location in West Olive, Michigan where the plant can easily obtain the coal from nearby railways, deposit carbon dioxide into nearby geological formations, and draw cooling water from Lake Michigan.
2 Project Goals

2.1 Overall Goals

We will be creating a design for three different types of power plants. The first power plant that we will model is a conventional coal power plant, the second will be an oxygen-fed clean coal power plant. The conventional plant will be our base case which we will compare to both an oxygen feed and an IGCC power plant.

2.1.1 Conventional

The purpose of designing the conventional coal fired power plant in this project is to use it as a basis of comparison to the clean coal plant option. For our conventional plant we will be designing two of the major components, the boiler and the cooling system as well as specifying other components such as the turbine and pumps. In order to create the best design of a conventional coal plant we will be analyzing many variations of the simple Rankine Cycle which can be seen in Figure 1. This Rankine Cycle shows the basic energy flow for our conventional plant. The coal is fed into a pulverizer and then burned in a boiler to create steam. This steam is sent to a turbine which turns a generator to create electrical power. The steam then exits the turbine into a condenser and cooling tower from where it is pumped back into the boiler to become steam again. Some variations of this basic cycle we will be considering are the possibility of multiple reheats within the turbine so we can operate at higher pressure and temperature. We will also be considering implementing a regeneration system which preheats the boiler feed water increasing the overall efficiency of the plant.

![Figure 1: Process Flow Diagram of Basic Rankine Cycle](image)

A reheat in the Rankine Cycle is implemented by diverting steam from the turbine at an intermediate pressure. This steam is directed through the boiler once again before it travels back to the turbine. This
addition of heat into the steam causes an overall increase in power output from the turbines, and an increase in overall process efficiency.

A regeneration portion is added into the Rankine Cycle by taking steam from the turbine at intermediate pressures, and using it to preheat the water into the boiler. This can be done by using open feed-water heaters, which directly mix the steam stream with the water. Regenerations can also be implemented by using a closed feed-water heater, which indirectly heats the water while condensing the steam in a heat exchanger. The two streams are joined when they are both liquids at the exit of the heat exchanger.

In order to find the most cost effective design for the Rankine Cycle, we performed multiple iterations of our analysis. The cycle model was modified and reanalyzed in order to determine the most cost effective methods to increase efficiency. With each added stage in the cycle, there was an increase in system efficiency along with an increase in equipment costs. We found the balance between fuel costs and capital costs by deciding on a 3 stage cycle. This included two reheats and two regenerations.

2.1.2 Conventional with Oxygen Feed

One form of clean coal technology stems from modifying the boiler in a conventional coal power plant so that the coal is burned with an oxygen feed, instead of an air feed. Implementing an oxygen stream eliminates the presence of Nitrogen from the area of combustion. The nitrogen absorbs the heat released from the combustion reaction, decreasing overall efficiency of the boiler. The oxy-coal plant also uses carbon capture technology in order to separate CO₂ from the flue gases, where it is then stored in nearby underground geological formations. This separation process is also made easier with the use of pure oxygen.

The process flow of this option is very similar to the base case option. The differences arise in the design of the boiler, production of oxygen, operating temperatures within the boiler, and the additional equipment required to separate oxygen from the air and to separate CO₂ from the flue gases.

This type of plant assumes the same process flow as the conventional option but with the exception of the addition of an oxygen feed. Certain components of the plant such as boiler, and sizing of various components are different but the overall energy flow will be the same as the conventional plant. The design of this power plant also has to consider safety. The oxygen stream is much more reactive, and will produce hotter flames, so the design of the boiler will require measures to prevent failure of the piping or disaster.
Figure 2: CO2 Recycle for Oxygen Fed Boiler

This figure demonstrates the carbon dioxide recycle principle. Flue gas coming out of the boiler will first pass through a filtration device to remove the fly ash. This piece of equipment is specified by Team 14. Approximately 76% of the flue gas is then recycled back to mix with the oxygen feed. This provides preheating, and is also used to absorb heat from the combustion in the boiler.

Some studies suggest that water be removed from the flue gas stream, to recover for other purposes. However, as there is no shortage of water at our location, this is deemed unnecessary. The effort to remove the water from the system would counteract the benefits gained by purifying the flue gas.

2.1.3 Integrated Gasification Combined Cycle (IGCC)

The second clean coal technology considered for comparison is the Integrated Gasification Combined Cycle power plant (IGCC). The basis of this plant lies in the processing of the coal before combustion. The coal is sent into a gasifier with a pure stream of oxygen, where it is turned into Synthesis Gas, a mixture of carbon monoxide and hydrogen. It is then burned in a gas turbine within a combined cycle. This clean option will also have a series of flue gas scrubbers that removes sulfur, carbon dioxide, and other pollutants before they are released into the air. Figure 3 depicts the process flow of the IGCC power plant.
The IGCC option is expected to be much more effective in releasing cleaner emissions and operating efficiently, however, it will have a very high cost. This portion of the case study will be handled completely by Team 15.

2.1.4 Cost

A cost comparison has been made comparing all three of our power plants (conventional and our two cleaner varieties). First we will minimize cost without regard for efficiency or environmental impacts. This will determine whether or not our worst case is economically sustainable. Secondly, we will minimize fuel costs by obtaining high efficient components. Components with high efficiency will result in low operating costs. This will be completed in order to minimize the pollutants associated with our plants. Our final product will be an optimized balance between pollutant levels and total cost. Our plants will be evaluated assuming a financial life of 20 years. All of our analyses will have a 20 year payoff period. The financial lifetime is not to be confused with the operating lifetime of the plant.

The cost comparisons that will be made can be broken into four subcategories: Purchased Equipment Costs including installation (PEC), Fuel Costs (FC), Operations and Maintenance (O&M), and Carrying Charges (CC). The sum of these costs represents the Total Capital Investment (TCI) or Total Revenue Requirement (TRR) needed in order for an economically sustainable power plant.

The most important factor in cost calculations is the cost of the final product; ours being cents per kilowatt-hour. Our final product must match or surpass the levelized total revenue requirement per Giga-Watt (This will be multiplied by the hours of operation in order to be converted to cents per kWh) and be below the average retail cost of electricity in order to be considered economically viable.
The significance of this graph is that we will be aiming for a cost of 7.46 cents at the customer per kWh. While we hope to have a competitive price, our main objective is to compare the costs between our plants. At this time it is unknown how much more expensive a clean coal plant will be to operate, but that is the essence of our proposal.

2.1.4.1 Purchase Equipment Cost
The PEC is comprised of the initial costs for all components in the power plant. The primary components of the system are the: compressor, boiler, turbine, condenser, heat exchangers, generator, and pulverizer.

The PEC also accounts for piping, instrumentation and controls, and electrical equipment and materials. The PEC is highly dependent on both size of the components. Because the capacity is pre-determined by the power output of the plant, only the efficiency of the plant can be varied, which will determine the size of the components. As previously stated our first cost optimization will minimize the PEC. We calculated the Total Capital Investment using a breakdown of Lang Factors given to us by Peters\(^1\). This is comparable to values found on page 352 of Bejon\(^2\), as seen in Table 1: Total Capital Investment Factors.

<table>
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<td>Total Cost</td>
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1 (Peters, 2003)

2 (Bejan, Tsatsaronis, & Moran, 1996)
We decided to use the factors given in Peters to our cost estimation, in order to keep costing consistent between all three clean coal teams.

2.1.4.2 Fuel Cost
In order to accurately design the boiler a specific type of coal must be chosen based on heating value, average price, total moisture content, contaminant content, and availability. The four major coal ranks are Bituminous, Subbituminous, Lignite, and Anthracite. Table 2 shows the typical heating values and average prices for the four coal ranks.

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<thead>
<tr>
<th>Coal Rank</th>
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<tr>
<td>Bituminous</td>
<td>55.44</td>
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<tr>
<td>Subbituminous</td>
<td>13.35</td>
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<tr>
<td>Lignite</td>
<td>17.26</td>
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<tr>
<td>Anthracite</td>
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Illinois #6, a Bituminous coal, was chosen as our coal type. It was selected due to its high availability at our location, its relatively low ash content (approx. 15%)\(^2\), its acceptable heating value (approx. 12226 Btu/lbm, dry)\(^3\), its low moisture content (< 20%), and for the basis of fair comparison between the subgroups.

2.1.4.3 Operations and Maintenance
Operations and Maintenance costs are split into two categories which are fixed costs and variable costs. The fixed costs are salaries, labor, and maintenance materials. The variable costs are costs associated with operation, other than fuel, such as cooling tower make-up water and waste disposal. Typically fuel costs are included in O&M, however because this is an entity we desire to minimize we separated it from operations and maintenance costs. These operations and maintenance costs have a strong correlation to the capital cost of the plant. This was assumed to be 20\(^4\) percent of the total purchased equipment cost of the plant.

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1 (U.S. Energy Information Administration, 2011)
2 (Morgantown Energy Research Center, 1978) from Texaco modeling kinetics //group15/gasifier/research/texaco
3 http://www.et.byu.edu/~larryb/Heating%20Value_1.html
4 (Bejan, Tsatsaronis, & Moran, 1996)
2.1.4.4 Carrying Charges

Carrying Charges are essentially every other costs associated with our plants. This includes land purchase, taxes, finance, setup costs, working capital, site development, building needs, coal storage and transportation. We are assuming that the power plants will have one location therefore there will be no difference in land purchase cost between plants. There is no current nationwide carbon tax in the USA, but some states do enforce a carbon tax. If our plants are located in one of those states then carbon taxing will be a variable to consider. The financing and setup costs are dependent upon the components; we anticipate lower setup and financing costs for the conventional case in comparison with the clean coal alternative.

2.1.5 Environmental Impact

As coal power technology has progressed over the last hundred years, we have only recently become aware of our actions, and their effects on the environment. This is especially true in regards to what has been allowed to leave the smoke stacks of coal power plants. The average coal power plant in operation today generates 600 MW of power and produces around 3,700 kilotons of carbon dioxide, 10,000 tons of sulfur dioxide, and 10,200 tons of nitrogen oxide every year.\(^1\) There are several actions that can be done to reduce the amount of emission in a new power plant.

One way to help reduce the amount of pollutants emitted into the atmosphere is by building a power plant with an oxygen feed. This type of plant has several environmental benefits. By burning coal with an oxygen feed, the plant increases the concentration of carbon dioxide and other pollutants within its flue gas stream. In fact, the lack of nitrogen in the oxygen stream helps to increase the concentration of pollutants by a factor of 3.5.\(^2\) With this increased concentration of pollutants it makes it much easier to compressed and sequestrate them.

The IGCC power plant also helps to dramatically reduce the amount of pollutants emitted into the air, as well as achieving high efficiencies. This relatively new technology features higher than ever efficiency, with some plant designs reaching efficiencies around 45%\(^3\). Generally speaking, if you have a more efficient plant you will emit less pollution. In addition to this, studies have been performed on already existing small scale plants that have shown that these plants are able to capture up to 90-95% of mercury.\(^4\) Similar to the power plant with oxygen stream, IGCC can more easily capture its flue gasses and use various methods to safely dispose of the carbon dioxide in an environmentally friendly way.

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1. (Union of Concerned Scientists, 2009)
2. (D. McDonald and D. DeVault, 2007)
3. (World Coal Association)
4. (clean-energy.us, 2009)
By implementing these clean technologies, we will be able to dramatically reduce the amount of pollutants we put into the atmosphere. The amount of pollutants captured is dependent upon the process we choose, whether it is IGCC or Oxygen-Fed PC power plant.

2.2 Team 16’s Scope

Within the overall scope of the entire clean coal comparison, we will be focused on the energy balances, the design of the components of the power cycle, and cost analyses for the power plants. This includes a design of the Rankine power cycle for the conventional power plant, and the conventional power plant with an oxygen feed. The component design does not include the design of the separations or scrubbing equipment used to process the flue gas.

For a fair comparison, it is important that we analyzed state of the art designs for each of our power plants. This required research on recently constructed power plants to learn about common practices, and industry standards. After sufficient research, we modeled several variations of power cycles in Engineering Equation Solver (EES) to make a decision on the most cost effective Rankine cycle for the purposes of our base case plant. These purposes were to find the optimized plant to have the lowest cost over the financial lifetime of our plant. This cost included the overall capital investment of the plant, and it also considered the cost of fuel.

No matter how efficient or environmentally friendly it may be, we cannot choose a power cycle setup for the conventional plant unless it is economically feasible. Throughout the entire process we will base our decisions on equipment costs as well as efficiency. We will then perform an exergetic analysis of the power cycles, so that we can produce optimizations based on both initial and long term costs.

It is our team’s responsibility to design a boiler in detail for the conventional power plant in the base case option, as well as the oxygen feed option.
3 Thermodynamic System Design

3.1 System Requirements

The thermodynamic cycle chosen for our system should represent an industry standard Rankine Cycle implemented in modern thermal power plants. The cycle should have a large efficiency so it can gather as much as the energy from the coal as possible. However, the cycle should not sacrifice lifetime cost for efficiency. Since the design is to represent a common coal fired power plant, it should be motivated by monetary gain, just as power industries are motivated.

It is beneficial to produce the most cost effective power plant possible. This implies that the efficiency of the plant should be as high as possible while reducing the equipment necessary to achieve the power requirement. Our design must find the balance of this tradeoff.

3.2 Design Process

The design of the system started when we modeled a Rankine Cycle in its simplest form. We then modeled other modifications of the Rankine Cycle in order to find the added efficiency of adding more stages. Figure 4, below, shows that there is a significant decrease in added efficiency for every additional stage, up to 5 additional stages.

![Figure 4: Thermal Efficiency vs. Number of Turbine Stages](image)

In order to increase the thermal efficiency of the system is beneficial to implement regenerations and reheats into the Rankine cycle. However, after a certain number of stages it is not cost effective to
implement any additional stages due to a high cost compared to a small improvement in the system’s
efficiency. After making a thermodynamic analysis of all of these options, we narrowed our decision to
three design options. These options were 2, 3, and 4 stages of reheat with regeneration. These three
options were chosen because of the large difference in efficiencies between a 1 and 2 stage systems, and
due to the diminished return of efficiency from 4 to 5 stage systems. In each of these design scenarios
there are an equal number of reheat and regenerations. This is because we found that the most cost
effective layouts included both a regeneration and reheat in each stage of the cycle. These systems would
be carried to the next stage of our design process, where we found the lifetime cost of the system. This
can be found in Section 10.10: Start-up Costs

These are costs associated with: Labor, materials, equipment, etc. that is only used during the startup of
the plant, as well as a loss of income due to the plant not running at full capacity. As was shown in
Section 2.1.4.1 Purchase Equipment Cost, the start-up costs have been capitalized, to be determined as a
fraction of the purchase cost of the components. Generally, these costs are around 25% of the purchase
cost. In the case of the Peters factors, this is included in the working capital.
Figure 5: PFD of 3 Stage Cycle
### 3.3 Pressure Optimization

The pressure of the steam through the Boiler was set to 15 MPa. This limit was set within our design so that the steam remained below the supercritical limit of 22 MPa. The chosen pressure was set 7 MPa below the supercritical limit, because the additional pump work and pump costs required made the system more expensive, and therefore less profitable. Conversely, the pressure of the steam exiting the final turbine is brought down as low as possible. The condenser pressure was set to a low, 20 kilopascals. These pressures have a large effect on the efficiency of the system, since the pressure difference is the driving force that spins the turbines, generating electricity. Figure 6, below, shows a Temperature-Entropy diagram (T-S Diagram) of the 3 stage cycle shown above in Figure 5. This chart is tracks the temperature and entropy of the stream throughout the entire power cycle. This is useful because it visualizes where entropy generation is prevalent in the cycle, which is a major cause of inefficiency.

![Figure 6: T-S Diagram for Steam Cycle](image)

The T-S diagram in Figure 6, above, shows the State Points of the water flow throughout the cycle. The term ‘State Point’ refers to the label on the specific, thermodynamic properties (i.e. Temperature, Pressure, and enthalpy). This is not to be confused with titles given to the stream names, which includes the mass flows of the streams. This process begins at State Point 1, which represents the condition of the water exiting the condenser. It is then pumped up to State Point 2. This pump affects the pressure but, as shown in the T-S Diagram, the temperature and entropy of the water encounters small change from the pump. The water then travels through the first closed feed-water heater, where it exits at the State point labeled as number 3. At the exit of the heat exchanger, it then joins the condensed regeneration stream. The state of the water exiting the mixing is labeled State Point 4. The water travels through the second
closed feed-water heater where it exits as state point 5. It is mixed with the second condensed regeneration stream and make-up at state point 6. This water then reaches the economizer section of the boiler, where it reaches state point 7, representing a saturated liquid. Passing through the radiation section of the boiler, the steam reaches state point 8, which represents the point in the boiler where the steam is completely saturated. It then travels through both of the super heating sections of the boiler, where it contains the label of State Point 9. This stream is then sent through the turbine, where State Point 10 is the exiting conditions of the first turbine stage. The steam at state point 10 is then split into two streams. Most of the stream is sent through the reheat section of the boiler and brought up to the conditions of State Point 11, while the remaining regeneration stream is sent through the second closed feed-water heater and condensed, where it is brought up to State Point 17. This is then fed into a pump where it is combined with the feed-water before it enters the boiler. Meanwhile, the exit stream at State Point 11 is directed back into the second stage of the turbine. The steam at the exit of the second stage of the turbine is defined as State Point 12. The stream is split where most of the stream continues through the second reheat section of the boiler, to become State Point 13, while the remaining portion of the steam at State Point 12 is sent through the first feed-water heater where it heats the water stream, condenses, and emerges at State Point 16. This water is then pumped up to the correct pressure, where it rejoins the feed-water. The steam stream that is exiting the second reheat section of the boiler enters the turbine, where it is brought to State Point 14 upon exiting the turbine. This steam then enters the condenser, where it is brought back to State Point 1.

Since the thermodynamic processes do not differ between the two cases, Figure 5 and Figure 6 represent the thermodynamic cycle for both the conventional case and the Oxy-Coal case. The differences arise in the design of the boiler.

In each power cycle case the intermediate pressures, or the pressure of the steam in between the turbine stages, was varied in order to maximize the power output of the cycle. These pressures affect the temperature of the steam exiting each of the turbines. This also affects the heating duty of the reheat sections through the boiler, the fraction of the steam stream that is diverted into the feed-water heaters for regeneration, and the work requirements for the pumps.

The computer program, Engineering Equation Solver (EES), was used to optimize the intermediate pressures of the system. EES is a program that uses parallel processing to simultaneously solve complex sets of equations, as well as optimize systems with more than 1 or more degrees of freedom. This program is also equipped with fluid property libraries in order to quickly find thermodynamic properties of our fluids. Using these capabilities, we adjusted each intermediate pressure within our model in order to maximize the efficiency of the system. We performed this pressure optimization on every process layout that we analyzed. This was done so that we were able to compare cycles that were using optimized pressures rather than arbitrary guess values for pressures, which provided for a more accurate comparison between cycles.

3.4 Temperature Selection

One major design decision is to select a temperature of the steam exiting the boiler and reheat. When the temperature is higher in the inlet of the turbines, the power output of the turbines rises, and therefore the efficiency of the system increases. Due to heating constraints on the material of the piping, and efficiency
limits of the boiler, the temperature of the steam was limited to 593 °C. A higher temperatures reduces the lifetime of the piping, which is subject to fatigue due to exposure to high pressures and temperatures.

3.5 Economic Analysis

3.5.1 Economic Optimization

With the thermodynamic properties of the fluids within each of our designs and the costing functions of each component within each design, it was then possible to adjust the component sizing and system parameters in order to minimize the total lifetime cost of the system as opposed to minimizing the efficiency.

3.5.2 Total Cost of Electricity Production

The optimization was performed in EES. The costing functions were tied into the original thermodynamic models and a cost was found for the three chosen design options. First, the initial costs of the system were determined, which included all of the equipment costs, building costs, and start-up costs. Next, the operating costs were determined, which included the fuel cost, operations and maintenance costs, and carrying charges of the power plant. These costs were lumped into a levelized annual payment system for the 22 year economic lifetime of the plant, which was used to calculate the cost to produce electricity for each design option.

<table>
<thead>
<tr>
<th>Design Option</th>
<th>Cost of Electricity (cents/kW-hr)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3 Stages</td>
<td>7.2</td>
</tr>
<tr>
<td>4 Stages</td>
<td>8.3</td>
</tr>
<tr>
<td>5 Stages</td>
<td>8.9</td>
</tr>
</tbody>
</table>

The 3 stage system was chosen because it yielded the cheapest cost of electricity. These costs can be seen in Table 3, above. This is not to be confused with the market price of electricity, which refers to the price at which utility companies sell electricity to consumers, which is expected to be higher. These results are reasonable because they reflect the industry cost for electricity generation.

3.6 Final Design Results

The thermodynamic system stage of our design let us narrow our possible designs to 3 key options, Rankine Cycle with 2, 3, and 4 stages of Reheat with Regeneration. Initially, we evaluated systems with closed feed-water heaters, open feed-water heaters, or a combination of both. The final, most cost effective options all contained at least one closed feed-water heater, and some incorporated the use of open feed-water heaters. A closed feed-water heater consists of a heat exchanger which condenses a

1 (Babcock & Wilcox a McDermott company)
stream that is separated from the turbine before it is pumped up to the proper pressure and mixed back in with the cool stream. An open feed-water heater takes a diverted stream from the turbine and directly mixes the streams in a deaerator. These feed-water heating methods serve the purpose of preheating the stream before it enters the boiler, in order to relieve the boiler of some of its heating duty.

In order to confirm the validity of our EES system we then developed a UNISIM model of the design options. The results shown by the UNISIM model were very similar to the results of the EES model. This confirms that the models are valid representations of the chosen systems, as shown in Table 4, below.

Table 4: Comparison of EES Results with UniSim Results

<table>
<thead>
<tr>
<th>System Condition</th>
<th>EES Results</th>
<th>UniSim Results</th>
<th>Difference</th>
<th>Percent Difference</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power Output [MW]</td>
<td>1020.1</td>
<td>1020.1</td>
<td>0</td>
<td>1</td>
</tr>
<tr>
<td>System Efficiency [%]</td>
<td>44.13</td>
<td>44.86</td>
<td>0.73</td>
<td>1.6</td>
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<tr>
<td>Mass Flow Rates:</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>m₁ [kg/s]</td>
<td>610.6</td>
<td>622.2</td>
<td>11.6</td>
<td>1.9</td>
</tr>
<tr>
<td>m₂ [kg/s]</td>
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<td>606.2</td>
<td>1.7</td>
<td>0.3</td>
</tr>
<tr>
<td>m₃ [kg/s]</td>
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<td>474.0</td>
<td>0.6</td>
<td>0.1</td>
</tr>
<tr>
<td>m₄ [kg/s]</td>
<td>6.2</td>
<td>16.1</td>
<td>9.9</td>
<td>61.5</td>
</tr>
<tr>
<td>m₅ [kg/s]</td>
<td>129.9</td>
<td>132.2</td>
<td>2.3</td>
<td>1.7</td>
</tr>
</tbody>
</table>

In the Table above, mass flow rates 1, 2, and 3, represent the mass of steam flowing through the first, second, and third turbine stage respectively. Mass flow rates 4, and 5 represent the mass flow rates of steam into the second, and first feed-water heater, respectively.

A summary of the thermodynamic properties of each state point can be seen in Table 5, below. The exergy of a state point is the potential work that can be gathered from a fluid which is brought to the ‘Dead State’. The dead state is a theoretical value which represents the lowest possible temperature and pressure of a fluid, purely for the purpose of calculating exergy. This exergetic analysis involved finding costs associated with each fluid stream. This method attributes a dollar sign value on each fluid stream. Then a system of equations was used to find a cost of work output. This was performed in a similar fashion to an energy balance, where cost in equals cost out on each component.
<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
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<th></th>
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<td>-</td>
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<td>67.8</td>
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<td>-</td>
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<td>6.1</td>
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<td>6.1</td>
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<td>12.2</td>
<td>8.2</td>
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<td>-</td>
<td>232.6</td>
<td>1.4</td>
<td>7.7</td>
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</tr>
</tbody>
</table>
4  Boiler Design

4.1  Design Overview and System Requirements
The most crucial aspects to boiler design are steam conditions, fuel, and environmental constraints.\(^1\) The boiler must output steam at the right conditions in order to achieve the necessary power output of the system. Once the thermodynamic cycle design was design and optimized we found that the heat requirement of the boiler was 2266 MW. This means that 2266 MW of heat must be absorbed by the water flowing through the boiler. This is achieved in two sections which are the radiation section and the convective pass section. These sections are named thusly due to their primary mode of heat transfer. Note that the following sections describe the process by which the boiler was designed. The designed values can be found in the Results Section.

4.2  Radiation Section
The radiation section is the section of the boiler in which combustion occurs and heat is absorbed by membrane wall tubes. Membrane wall tubes are pipes welded together to form the interior walls of the boiler. The heat requirement of the radiation section was found as part of the thermodynamic cycle design. The amount of heat absorbed by the membrane wall tubes from the hot gas from combustion can calculated using equation 1.

\[
q_{sg} = A(e_g(E_g - \alpha_sE_s))
\]

Where: \(q_{sg}\) represents the net rate of heat flow from the hot gas to the membrane walls, \(A\) represents the enclosure surface area, \(e_g\) represents the emissivity of the gas, \(E_g\) represents the temperature of the gas multiplied by the Stefan-Boltzmann constant, \(\alpha_g\) represents the absorptivity of the gas, and \(E_s\) represents the surface temperature of the wall tubes multiplied by the Stefan-Boltzmann constant. A view factor of 1 was assumed because the furnace walls completely enclose the hot gas. Emissivity of the gas, \(e_g\), was found using equation 2.

\[
e_g = e_{H_2O} + e_{CO_2} - \Delta e\]

Where: \(e_{H_2O}\) represents the emissivity of gaseous \(H_2O\), \(e_{CO_2}\) represents the emissivity of \(CO_2\), and \(\Delta e\) represents an emissivity correction factor that accounts for radiation between gases. It should be noted that other gases such as oxygen and nitrogen emit and absorb insignificant amounts of radiation and therefore were not taken into account for calculations.\(^2\) The emissivity of \(H_2O\) and \(CO_2\) are dependent on their respective partial pressures and mean beam length. Mean beam length can be estimated using equation 3.

\(^1\) (Babcock & Wilcox a McDermott company)

\(^2\) (Babcock & Wilcox a McDermott company)
Where: $L$ is beam length, $V$ is the enclosure volume, and $A$ is the enclosure surface area. The partial pressure multiplied by the mean beam length was used with Figure 7 to find the emissivity of the gas. Note that Figure 7 is an example and only can be used for $CO_2$.

![Figure 7. Emissivity of carbon dioxide](http://books.google.com/books?id=8Au5oOMAdsoC&pg=PT93&lpg=PT93&dq=emissivity+of+water+vapor+partial+pressure+mean+beam+length&source=bl&ots=MCIcTIDCZA&sig=TyH1TbLLPGO_Mw5Mw7RypT5zlio&hl=en&sa=X&ei=_q6pT5y5D8eegwfGy-W-AQ&ved=0CDsQ6AEwAA#v=onepage&q=emissivity%20of%20water%20vapor%20partial%20pressure%20mean%20beam%20length&f=false)

Figure 8 was used with the previously found partial pressures in order to find the gas emissivity correction factor.

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1. [Link to Google Books search result](http://books.google.com/books?id=8Au5oOMAdsoC&pg=PT93&lpg=PT93&dq=emissivity+of+water+vapor+partial+pressure+mean+beam+length&source=bl&ots=MCIcTIDCZA&sig=TyH1TbLLPGO_Mw5Mw7RypT5zlio&hl=en&sa=X&ei=_q6pT5y5D8eegwfGy-W-AQ&ved=0CDsQ6AEwAA#v=onepage&q=emissivity%20of%20water%20vapor%20partial%20pressure%20mean%20beam%20length&f=false)
Figure 8. Radiation heat transfer correction factor for mixtures of water vapor and carbon dioxide.

The absorptivity of the gas, $\alpha_g$, was found using equation 4.

$$\alpha_g = \alpha_{H_2O} + \alpha_{CO_2} - \Delta \alpha$$  \hspace{1cm} (4)

Where: $\alpha_{H_2O}$ is the absorptivity of water, $\alpha_{CO_2}$ is the absorptivity of carbon dioxide, and $\Delta \alpha$ is the absorptivity correction factor. The absorptivity of $H_2O$ and $CO_2$ were found using equations 5 and 6 respectively as shown below. The gas absorptivity correction factor is equal to the gas emissivity correction factor.

$$\alpha_{H_2O} = \varepsilon_{H_2O} \left( \frac{T_g}{T_s} \right)^{0.45}$$  \hspace{1cm} (5)

$$\alpha_{CO_2} = \varepsilon_{CO_2} \left( \frac{T_g}{T_s} \right)^{0.65}$$  \hspace{1cm} (6)

Where: $T_g$ represents the temperature of the products of combustion, and $T_s$ represents the surface temperature of the membrane wall tubes. The temperature of the products of combustion were found by estimating the enthalpy of fuel and air by guessing a final gas temperature, following this the initial and final enthalpies were set to be equivalent. The surface temperature of the membrane walls was assumed to be the saturation temperature of water at our operating pressure of 15,000 kPa. This is assumed because

\[1\] http://books.google.com/books?id=YBaNaLurTD4C&pg=PA866&lpg=PA866&dq=radiation+heat+transfer+correction+factor+associated+with+mixtures+of+water+vapor+and+carbon+dioxide&source=bl&ots=tNPdailJ9x&sig=AOXbpQitQoBA9slexDFR9_zZnKo&hl=en&sa=X&ei=97WpT__7D9LxggeCs6nWAQ&ved=0CDcQ6AEwAA#v=onepage&q=radiation%20heat%20transfer%20correction%20factor%20associated%20with%20mixtures%20of%20water%20vapor%20and%20carbon%20dioxide&f=false
the membrane wall tubes are being cooled by the water/steam passing through them. Once the absorptivity and emissivity of the gas was calculated the only unknown variable from the overall radiation heat transfer equation is the internal surface area.

4.2.1 Sizing
The height of the boiler was determined by multiplying the velocity of the gas in the boiler by the residence time. The residence time was found using experimental data that relates the residence time to the allowable heat release rate per volume. The number of tubes for the radiation section was found by dividing the necessary surface area by the outer diameter of the tubes.

4.3 Convective Section
The convective pass section can be best described as a series of bare tube cross-flow heat exchangers. This section includes the primary and secondary superheaters, both reheaters, the economizer. The heat requirement for each of these was previously calculated as part of the thermodynamic cycle design. The amount of heat absorbed by each component in the convective pass section can be calculated using equation 7.

\[ \dot{Q} = UAF(LMTD) \]  

Where: \( \dot{Q} \) represents the heat requirement, \( U \) represents the overall heat transfer coefficient, \( F \) represents the shape factor, and \( LMTD \) represents the log mean temperature difference. Note that for all components in the convective pass section \( F \) is 1. This is because they are bare tube cross-flow heat exchangers. The log mean temperature difference can be calculated using equation 8.

\[ LMTD = \frac{(T_{g,in} - T_{s,out}) - (T_{g,out} - T_{s,in})}{\ln \left( \frac{T_{g,in} - T_{s,out}}{T_{g,out} - T_{s,in}} \right)} \]  

Where: \( T_{g,in} \) represents the temperature of the flue gas as it enters the heat exchanger, \( T_{s,out} \) represents the temperature of the steam exiting the heat exchanger, \( T_{g,out} \) represents the flue gas temperature as it exits the heat exchanger, \( T_{s,in} \) represents the temperature of the steam entering the heat exchanger. The overall heat transfer coefficient can be calculated using equation 9.

\[ U = \frac{1}{\frac{1}{h_g} + \frac{t_m}{k} + \frac{1}{h_s}} \]  

Where \( h_g \) represents the convection heat transfer coefficient for the flue gas, \( t_m \) represents minimum allowable pipe thickness, \( k \) represents the thermal conductivity of the material, and \( h_s \) represents the convection heat transfer coefficient for the steam. Because the heat requirement of each section was known the heat transfer equation for convection can be used to find the area necessary for heat transfer. The total tube length was found by dividing the necessary surface area by the circumference of the pipes. The length of each pass is the width of the boiler minus length necessary for the pipes to turn into the next pass. The total number of passes is the total tube length divided by the length of each pass. The number of tubes per pass can be calculated using equation 10.
\[ N_{tpp} = \left( \frac{d}{S} \right) - L_{\text{turn}} \]

Where: \( N_{tpp} \) represents the number of tubes per pass, \( d \) is the depth of the boiler, \( S \) represents the surface spacing necessary to allow the flue gas to travel to the subsequent sections without absorbing too much heat, and \( L_{\text{turn}} \) is the amount of length necessary for the pipes to turn. Finally, the total number of tubes can be calculated by dividing the number of passes by the amount of tubes per pass.

The inner diameter of the tubes was calculating using the equations 11 and 12 which were found in the ASME B31.3-2002 Process Piping code book.

\[ t = \frac{PD}{2(SE + PY)} \]

\[ t_m = t + c \]

Where: \( t \) represents the designed pipe thickness, \( P \) is the internal pressure of the pipe, \( D \) is the outside diameter of pipe, \( S \) is the allowable stress at a specific temperature, \( E \) is the quality factor, \( Y \) is a coefficient based on whether the steel is Austenitic or Ferritic, \( t_m \) represents the minimum pipe thickness, and \( c \) represents the sum of the mechanical allowances plus corrosion and erosion allowances. The material is dependent on the internal pressure of the pipe as well as the maximum allowable temperature. The specific materials used in the boiler will be discussed later in the Results Section.

### 4.4 Results

#### 4.4.1 Air Pulverized Coal Combustion

The Results for the Radiation section are shown below in Table 6.

<table>
<thead>
<tr>
<th>Thermodynamics Membrane Wall Tubes</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Requirement (MW)</td>
</tr>
<tr>
<td>Steam Temperature In</td>
</tr>
<tr>
<td>Steam Temperature Out</td>
</tr>
<tr>
<td>Air Temperature In</td>
</tr>
<tr>
<td>Air Temperature Out</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Physical Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Transfer Area (m²)</td>
</tr>
<tr>
<td>Number of Tubes</td>
</tr>
<tr>
<td>Tube OD(m)</td>
</tr>
<tr>
<td>Tube ID(m)</td>
</tr>
<tr>
<td>Tube Length(m)</td>
</tr>
<tr>
<td>Material</td>
</tr>
<tr>
<td>Gas Emissivity</td>
</tr>
<tr>
<td>Gas Absorptivity</td>
</tr>
</tbody>
</table>
The outer diameter of the membrane wall tubes were assumed 2.5 inches on 3 inch centers based on typical outer diameter values.\(^1\) The material selected was chosen from a list of acceptable materials for the membrane walls based from maximum allowable metal temperature. The maximum allowable stress at the surface temperature of 833 K is 71 MPa.\(^2\)

The results for the convective pass section are shown below in Table 7.

**Table 7. Thermodynamic and Physical Specifications for the Convection Section**

<table>
<thead>
<tr>
<th>Convection</th>
<th>Final Superheater</th>
<th>Reheat 1</th>
<th>Reheat 2</th>
<th>Primary Superheater</th>
<th>Economizer</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Requirement (MW)</td>
<td>455.6</td>
<td>334.2</td>
<td>302.5</td>
<td>272.4</td>
<td>296.6</td>
</tr>
<tr>
<td>Steam Temperature In (K)</td>
<td>705.1</td>
<td>628.3</td>
<td>603.4</td>
<td>615.3</td>
<td>474.4</td>
</tr>
<tr>
<td>Steam Temperature Out (K)</td>
<td>873.2</td>
<td>873.2</td>
<td>873.2</td>
<td>705.1</td>
<td>615.3</td>
</tr>
<tr>
<td>Air Temperature In</td>
<td>2273</td>
<td>1720</td>
<td>1511</td>
<td>1336</td>
<td>1141</td>
</tr>
<tr>
<td>Air Temperature Out</td>
<td>1720</td>
<td>1511</td>
<td>1336</td>
<td>1141</td>
<td>827.7</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Physical Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td>Heat Transfer Area (m(^2))</td>
</tr>
<tr>
<td>Tube OD (m)</td>
</tr>
<tr>
<td>Tube ID (m)</td>
</tr>
<tr>
<td># of passes</td>
</tr>
<tr>
<td>Tubes per pass</td>
</tr>
<tr>
<td>Material</td>
</tr>
</tbody>
</table>

Figure 9 shows the size of our boiler as well as the pipe spacing for the convective pass section.

---

\(^1\) (Babcock & Wilcox a McDermott company)

\(^2\) (Babcock & Wilcox a McDermott company)
4.4.2 Oxy-Coal Combustion

A significant difference between an oxygen feed plant and an air feed plant is the boiler. That being said the heat requirement of the boiler is identical for both the Oxy-Coal plant and the conventional pulverized coal plant. This is because the power output must be 1 GW and all other components in the system are the same. A case study showed that an oxygen feed boiler requires only 65%\(^1\) of the surface area and 45%\(^2\) of the volume required for an air feed boiler. This is a result of the higher flame temperatures achieved when combusting a mixture of oxygen and flue gas. In actuality the required surface area for radiative heat transfer of the conventional case is identical to the Oxy-Coal case. This is because the flue

---

\(^1\) (Andrew H. Seltzer)

\(^2\) (Andrew H. Seltzer)
gas recycle stream is used to control the temperature of the combustion zone to be equivalent to the air case. A Unisom model was constructed in order to find the carbon dioxide concentrations necessary for limiting the flame temperature to an allowable level. It was found that 76% of the exhaust stream needed to be recycled. This recycled exhaust stream needs to have the fly ash removed before it can be recycled which was done using an electrostatic precipitator. The flue gas recycle is blown into the boiler by the supplementary vane of the burner. This will be further discussed in the burner specification section. Because the temperature of the flue gas in the Oxy-Coal boiler is the same as the temperature of the flue gas in the conventional case the same materials of construction will be used for their respective sections (e.g. The material for the membrane wall tubes is SA-213 T91 for both cases).

4.5 Preliminary Design Decisions
To design the boiler preliminary decisions had to be made in order to accurately model the boiler thermodynamically. We needed to specify a coal type, boiler layout, burner type, and burner layout.

4.5.1 Fuel
A specific type of coal must be chosen based on heating value, average price, total moisture content, contaminant content, and availability. The four major coal ranks are Bituminous, Subbituminous, Lignite, and Anthracite. Table 8 shows the typical heating values and average prices for the four coal ranks.

<table>
<thead>
<tr>
<th>Coal Rank</th>
<th>Heating Value [Btu / lbm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bituminous</td>
<td>10,500 - 15,500</td>
</tr>
<tr>
<td>Subbituminous</td>
<td>8,300 - 13,000</td>
</tr>
<tr>
<td>Lignite</td>
<td>4,000 - 8,300</td>
</tr>
<tr>
<td>Anthracite</td>
<td>≈ 15,500</td>
</tr>
</tbody>
</table>

Illinois #6, a Bituminous coal, was chosen as our coal type. It was selected due to its high availability at our location, its relatively low ash content (approx. 15%)\(^2\), its acceptable heating value (approx. 12226 Btu/lbm, dry)\(^3\), its low moisture content (< 20%), and for the basis of fair comparison between the subgroups. The cost per energy released by Lignite coal is lower than the cost per energy released by Bituminous coal, however it also typically has a much higher moisture and ash content. Moisture content in Lignite coal typically is around 40%\(^4\). The ash content of Lignite coal varies from 10-68% depending

---

1. (Kentucky Educational Television, 2005)
2. (Morgantown Energy Research Center, 1978)
4. (Babcock & Wilcox a McDermott company)
on where it is mined. Conventional pulverized coal firing systems require extra processing for moisture contents exceeding 40%.

4.5.2 Physical Layout
The layout of the boiler is dependent on: the firing method, the type of coal burned, and the heat requirement. The Universal Pressure coal-fired boiler (UPC) layout was selected. In the UPC subcooled water is pumped sequentially through all heating surfaces until it becomes the desired superheated steam temperature. The basis of a UPC boiler is that the water/steam flow only passes through the boiler once (no recirculation) which eliminates the need for a conventional drum that separates water from steam. Figure 10 is an example of a typical UPC boiler layout.

Figure 10. Example of a Universal Pressure Boiler

4.5.2.1 Process Description of Water Flow through Boiler
Water exits the second mixing tank and flows upward through the economizer to preheat the inlet water stream to a saturated liquid. Next the water flows to the base of the boiler where it is pumped up through the membrane wall tubes and exits as a saturated vapor. After the water has been boiled through the

1 (Babcock & Wilcox a McDermott company)

2 (Babcock & Wilcox a McDermott company)

5 http://www.babcock.com/products/boilers/up_specs.html
radiation section it is pumped up to the primary then secondary superheaters. The superheaters achieve
the desired exit stream temperature.

4.5.3 Environmental Constraints
Information concerning environmental constraints can be found in section 2.1.5.

4.5.4 Burner Specifications
In order to maximize steam production and minimize fuel requirements an efficient burner is needed.
There are several types of burners, and several burner arrangements, that have been considered for both
the conventional coal base case and the oxygen fired plant.1

The first type of burner analyzed was a conventional circular burner. The burner’s main component is a
nozzle with an impeller attached in order to disperse the pulverized coal. Supplementary air is added via
a register, which is controlled with interlinked doors in a circular pattern. Air distribution must be
uniform in order to achieve high combustion efficiency. For that reason the register is typically set in a
position that allows for uniform air distribution (except during start up where less air is used.) All of the
air in a conventional burner is added at the same time as the coal air stream from the pulverizer; because
of this high flame temperatures are reached thereby producing high levels of NO\textsubscript{x}.

By controlling the supplementary air and swirl we arrive at the second type of burner to be analyzed. The
S-type burner is a conventional circular burner in which the secondary air is controlled by a sliding disk
that moves up and down the barrel of the burner. Swirl is independently controlled by spin vanes in the
burner, rather than being dependent on the supplementary air added by a register in a conventional
circular burner. The S-type burner features higher efficiency then the conventional burner, due to better
flame and swirl control, however it also features high NO\textsubscript{x} production rates.

The third type of burner examined is a dual register burner or DRB. This burner was invented in order to
reduce maximize NO\textsubscript{x} reduction while maintaining a high efficiency. Similar to the S-type burner, air is
controlled by a sliding disk. A pitot tube grid is located in the burner barrel in order to measure air flow,
which is necessary for equal flame distribution. The DRB achieves higher NO\textsubscript{x} reduction rates by
introducing the supplementary air on the outer edge of the burner (outside the flame wall). All air, both
main and supplementary, are independently controlled with sliding dampers. This burner also features an
inner spin vane and an outer spin vane in order to maximize swirl control.

The DRB burners feature approximately a 70% reduction in NO\textsubscript{x} emissions from fuel in comparison to a
conventional burner.2 Due to its high NO\textsubscript{x} emission reduction and its high efficiency the DRB was
selected as the burners.

1 (Kitto, 1992)
2 (Babcock & Wilcox a McDermott company)
4.5.4.1 Burner Placement

Burner location in the boiler plays an important role in efficiency and NO\textsubscript{x} production. There are four primary types of boilers that were analyzed: Wall-Fired, Tangentially Fired, Cyclone-Fired, and Stoker-Fired.

A wall fired boiler features burners mounted along one or two opposing walls. This type of boiler layout leads to high peak flame temperatures therefore producing high quantities of NO\textsubscript{x}.

A cyclone fired boiler use crushed coal rather than pulverized; they are generally small which produces high peak flame temperatures, again resulting in high NO\textsubscript{x} emissions. For these reasons the cyclone fired boiler was omitted as an option.

Stoker-Fired boilers are essentially a grill. Coal is dumped onto a shaking grate and burned. Primary combustion air is added underneath the grate, and supplementary air is added at the top of the system (providing 15-20\% of air required for stoichiometric combustion). The “overfire” air leads to relatively low NO\textsubscript{x} emissions.\(^1\) The maximum steam generation rate for a Stoker-Fired boiler is 63 kg/s simply because it is impractical to size the stoker any larger. We require steam production rates around 610 kg/s and for this reason the stoker-fired boiler was omitted.

A tangentially fired boiler features burners located at the four corners of the boiler structure. The burners are individually focused on the tangent of an imaginary circle in the center of the boiler creating a rotating fireball in the center. This allows for stratified fuel-rich and fuel-lean regions (i.e. offers similar NO\textsubscript{x} reduction benefits as a DRB).

Because the efficiency difference between tangentially fired burners and wall fired burner is negligible the tangentially fired boiler was chosen due to high NO\textsubscript{x} reduction rates.

\(^1\) (Utility Boilers)
5 Water Purification System Design

5.1 System Requirements

Water fed into a process system requires extra purity to prevent system failure. This holds even more true for a system that requires the evaporation of water to steam, as any impurities will be forced out of solution once the water has evaporated. Dissolved solids and gasses, such as CO$_2$ and Oxygen, can cause corrosion, scaling, or build-up in the pipelines as the system runs. In addition, new water is needed constantly, as old water within cannot be allowed to build up concentrations of minerals and other impurities.

The most common method of cleaning water requires three steps. First, large particles in the feed must be removed. Ultrafiltration will be utilized for this purpose. Secondly, a deaerator will be used to remove dissolved gasses. Lastly, reverse osmosis will be used to remove dissolved minerals from the water.

Assuming that 2% of the boiler steam loop is to be replaced, another 200 gpm water will be sent into the boiler steam loop as a replacement. Water to the condenser will not be purified, as an alternative method of removing build-up will be employed as described in section 6.3 Condenser Final Design Results. A coarse filter will be employed on the water intake to prevent any large particles from entering the system.

5.2 Final Design Results

As is shown in Figure 11 below, approximately 200 gallons per minute of water is fed through the filter to remove smaller particles from Lake Michigan.

![Figure 11: Boiler Pre feed processing](image)

Small particles will be removed in the ultrafilter, where the primary costs come from replacement of the filters after a period of time due to build-up. F-001, the Deaerator, collects the water in a vessel, as seen in Figure 12: Deaerator cross-section. The water is heated to saturation, causing some of the entrained gasses to be released. A stream of steam then is sent through at high velocity, breaking the water into a mist, causing a release of the remaining gasses. Deaerated water spills over, proceeding towards the boiler, while gasses exit out the top\(^1\). Residence time through this vessel is set to be 8 minutes, and water

\(^1\) (Deaerators, 2002)
level within is held at approximately 80%. Therefore, the deaerator will be modeled as a vessel of size 2000 gallons.

![Deaerator Cross-section](image)

**Figure 12: Deaerator cross-section**

The deaerated water then passes through the RO membrane. A pressure gradient across the membrane forces water to move through against its concentration gradient. In doing so, dissolved ions in the water stream are left behind. A stream of this extra concentrated water is then removed from the stream, in order to dispose of the impurities.

Equation 13.9-2 from Geankoplis\(^2\) gives a method for determining the flow rate through a reverse osmosis system:

\[
N_w = A_w \times (\Delta P - \Delta \pi) \tag{13}
\]

In this equation, \(N_w\) is the flux (kg/s-m\(^2\)) through the membrane. \(A_w\) is the solvent permeability constant, in this case taken to be 0.0005 kg/s-m\(^2\)-atm. \(\Delta P\) is the pressure applied to the system to force the water through the membrane, and \(\Delta \pi\) is the osmotic pressure, a function of the concentration of the solute. In this case, the solute is the dissolved ions in the water.

---

\(^1\) (Deaerotor, 2008)

\(^2\) (Geankoplis, 2003)
An early report states that the dissolved ion concentration in Lake Michigan is approximately 150 ppm. Using information from table 13.9-1 in Geankoplis\(^3\) the osmotic pressure for the system is estimated as 0.4 atm. As it was noted that the study providing the quantity of dissolved minerals is old, additional checks were made for a much more concentrated system.

Flow through the membrane is 12.6 kg/s. Given an operating pressure differential $\Delta P$ of 50 bar, the flux is found to be 0.023 kg/s-m\(^2\). Here we noted that even if the concentration in the water was multiplied by a factor of 10, the osmotic pressure would only increase to around 4-5 atm. This would have a minimal effect on the overall system. A control system will be implemented to increase the pressure leading into the system, if flow rate out of the reverse osmosis is seen to be decreasing.

Given this flux, and the flow rate of 12.6 kg/s, an area for the membrane was found to be approximately 500 m\(^2\). Pressure leading into the system is 200 bar, with pressure leaving at 150 bar, to be the same pressure of the steam loop at the point of entry for the water feed stream.

Costing for the three separations components can be found in section 10.2 Filtration System.

5.3 Alternative Designs

A few small changes were considered to the basic design. It was suggested that an Ion Exchange system could be used in place of the Reverse Osmosis system. Our current process was chosen over this option.

---

\(^1\) (BMET Wiki, 2010)
\(^2\) (Beeton, 1965)
\(^3\) (Geankoplis, 2003)
due to a couple reasons. First, pricing for the two options was roughly similar in terms of variable costs over the range of flow rates that we would consider, about $2.20 per thousand gallons. Our industrial consultant suggested that reverse osmosis performs better for large scale operations like our power plant. In addition, with reverse osmosis being the newer technology, there is a greater potential for a decrease in cost in the future, as the technology is made more efficient.
6 Condenser Design

6.1 Availability of Water

The main factor in selecting a proper condenser design is the amount of water available. Since water is the primary source for cooling systems, if it is not readily available then alternative methods must be designed to provide the necessary energy removal. Three options were considered for this system: an open stream of water pumped in and released out to the lake; a closed loop utilizing a cooling tower; and an air-cooled fin-fan condenser. Our location near Lake Michigan has a convenient source of water nearby, and as such will not be using the aircooled condenser. In addition, the large quantities of water needed for the simple stream would have detrimental effects on the cost of our water cleaning processes. Therefore, it was decided to use a cooling tower to reduce the make-up water fed into the system. The other two designs will still be documented in section 6.3 Alternative Condenser Design for the purpose of future reference, in the event that a new plant is established in a different location.

6.2 System Requirements

Steam comes off turbine T-002, which must be condensed before being sent back to the boiler. This is performed in heat exchanger E-000. This system removes 1.26 GW from the steam flow, condensing about 1700 tons/hr of steam.

![Figure 14: Condenser PFD](image-url)

In the case of a water stream or cooling tower, a shell and tube heat exchanger will be used to perform the actual condensing.
Assuming a counter-current set up for the heat exchanger, we can determine the $\Delta T_{lm}$, the log mean of the temperature differences in and out.

$$
\Delta T_{lm} = \frac{(\Delta T_{in} - \Delta T_{out})}{\ln \left( \frac{\Delta T_{in}}{\Delta T_{out}} \right)}
$$

$U$, the heat transfer coefficient through the exchanger, is calculated from the fluid and steam properties. Heat transferred, $Q$, is known from the UniSim model. $F$ is a factor cooresponding to the effectiveness of the heat exchanger, based on temperature differences, piping, and other factors. From this information, we can then use the equation for heat transfer via conduction:

$$
Q = (U)(A)(\Delta T_{lm})^*F
$$

This will give us a value $A$, the area of heat exchange. Details of these calculations are found in Appendix 4. Knowing $A$, we can then calculate the purchase cost of the heat exchanger.

Specifics on the parameters through the condenser can be found in the final design results below in Table 10: Specifications for Condenser. All numbers on a per exchanger basis

### 6.3 Final Design Results

#### 6.3.1 Simple Flow stream

This system just uses a stream of water pumped from Lake Michigan, sent through the heat exchanger, and pumped back out. This has fewer costs in the heat exchanger/condenser, as there are no additional capital costs needed for construction of cooling towers, fans, etc.

The maximum temperature change of the cooling water through the condenser would be 15 Celsius, to prevent high temperature water being dumped into the Lake. However, it was also noted that raising the temperature of the water too much caused a very small temperature differential through the exchanger, which greatly increased the heat transfer area. A balance was found between the amount of water flowing through the exchanger (which increases the size of the vessel) and the temperature change of the cooling water (which affects the heat transfer area). It was decided to use a flow of 44 $m^3/s$ of water into the condenser

This system will only pass through a coarse filter to remove large particles from the system. It will then bypass the additional water purification systems, and proceed into the condenser. A system of tube cleaning balls will be run through the condenser pipes, which will remove any contaminants that attach to the walls of the tubes. Balls are sized at 1 mm larger than the inner diameter of the tubes, and forced through due to pressure differential through the pipes. These balls are moderately compressible, to allow passage through the tube, while keeping good contact with the walls of the pipe. The friction caused from the oversized balls enables them to thoroughly remove impurities\(^1\). A collection system on the other side

\(^1\) (IN-TA-S Cleaning Balls, 2009)
of the condenser will then recover the balls, and cycle them back to be reused in the feed\textsuperscript{1}. Approximately 12 balls/hour/tube will be cycled through the system.

\begin{figure}[h]
\centering
\includegraphics[width=\textwidth]{figure15.png}
\caption{Demonstration of ball flow through the condenser\textsuperscript{2}}
\end{figure}

Given the flow of 44 m\textsuperscript{3}/s and 6 degree temperature change, an initial estimate of 80,000 m\textsuperscript{2} for a heat transfer area was calculated in our condenser. Given heuristics stated on page 436 of Seider\textsuperscript{3}, it is recommended that larger systems be split into smaller identical heat exchangers. For this reason, the condenser will be designed as 4 smaller condensers.

A heat flow of 285 MW is transferred through each heat exchanger. Steam passes through at 118.5 kg/s, and cooling water passes at 11057 kg/s. A pipe size of 1 ½ inch BWG 9 was selected for the condenser, which gives a pipe inner diameter of 30.48 mm, outer diameter of 38.1 mm. Heuristics\textsuperscript{4} suggest that flow through the tubes should be about 2 m/s, which gives us approximately \( n = 7500 \) tubes per exchanger.

\[ n = \frac{\text{Total volumetric flow}}{2 \frac{ms}{s} \times X \text{Area of Pipe}} \]

\footnotesize
\textsuperscript{1} (ZeroBall Loss - Condenser Cleaning System)
\textsuperscript{2} (Hamza, 2010)
\textsuperscript{3} (Seider, 2009)
\textsuperscript{4} (Shell & Tube Heat Exchanger Design)
In addition, flow of 0.6 m/s was taken for the shell side of the exchanger, giving us a shell diameter of 3.34 m. This is calculated by taking the 0.38 m² area needed, and adding in the additional area already occupied by the tubes.

Unisim models of the steam cycle provided us with the temperatures through the condenser:

Table 9: Temperature through condenser

<table>
<thead>
<tr>
<th>In degrees Celsius</th>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam</td>
<td>47</td>
<td>46.1</td>
</tr>
<tr>
<td>Cooling Water</td>
<td>27</td>
<td>33</td>
</tr>
</tbody>
</table>

From this, $\Delta T_{lm}$ can be calculated to be used for the equations. In addition, a correction factor $F_t$ can be found for the shell-and-tube heat exchanger, which is a measurement of the efficiency of heat transfer. This is determined by reading the data off graphs in Seider¹.

In addition, Unisim provided us with physical properties of the varying streams (viscosity, heat capacity, etc.). All of this data was then input into Appendix 4: Heat Exchanger Calculations. This sheet made calculations to determine the overall heat transfer coefficient, and from that, the heat transfer area. This in turn can be used to determine the length of pipe through the exchanger.

¹ (Seider, 2009)
The final designs are:

Table 10: Specifications for Condenser. All numbers on a per exchanger basis

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td># exchangers</td>
<td>4</td>
</tr>
<tr>
<td>Pipe Specs</td>
<td>1½ in. BWG 9</td>
</tr>
<tr>
<td># Tubes per exchanger</td>
<td>7500</td>
</tr>
<tr>
<td>Shell diameter</td>
<td>3.34 m</td>
</tr>
<tr>
<td>Heat transfer area</td>
<td>19450 m²</td>
</tr>
<tr>
<td>Tube length</td>
<td>24.3 m</td>
</tr>
<tr>
<td>Heat transferred (Q)</td>
<td>285 MW</td>
</tr>
<tr>
<td>$\Delta T_{lm}$</td>
<td>16.4</td>
</tr>
<tr>
<td>Heat Trans Coeff. (U)</td>
<td>929.9 W/m²·K</td>
</tr>
<tr>
<td>Correction factor (Fₜ)</td>
<td>0.96</td>
</tr>
</tbody>
</table>

6.4 Alternative Designs

As stated before, two designs that were not selected for the final project, for use with E-000 were a simple stream of water, and Air-cooled condenser.

6.4.1 Cooling tower

This option reduces how much water we must pump into the system. Because heat is removed by evaporation instead of temperature increase, more energy is transferred per mass. The amount of water needed to replace the evaporated water can be calculated as:

\[
Q_{\text{cond}} = \Delta H_{\text{tower}} = (h_{\text{vap}})(\text{Mass evaporated})
\]

In addition, as water evaporates, its remaining impurities are left behind. This would under normal circumstances cause the water to build up dangerous levels of contaminants in the stream. Therefore, a blowdown is used to remove a portion of the water, which will be more concentrated than the input water. This keeps the level of concentration at steady state. Heuristics\(^1\) indicate that 4 “cycles of concentration” should be used. This implies that the concentration of the blow-down water should be 4 times the

\(^1\) (Energy, 2011)
concentration of the make-up water. This in turn gives us a blow-down equal to 1/3 of the mass evaporated.

Given our condenser’s heat duty of 1.267 GW and a heat of vaporization of 2501.4 kJ/kg, this would imply that 506 kg/s of water is evaporated. Another 169 kg/s water would be removed in the blowdown, to remove impurities. This means that only 10700 gallons per minute of make-up water is needed, as compared to the 310,000 gpm of the standard system.

These benefits are counteracted by the cost of constructing the cooling tower, as well as having to perform extra purification on the water, since the stream passing through the tower will be evaporating. In addition, some power is necessary to run the fans that move the air through the tower.

A flow of 72000 m³/hr (317000 gal/min) will be travelling through the condenser loop, creating a temperature differential of 15 degrees Celsius through the condenser.

The cooling tower has specifications calculated online¹. 30 cooling cells were chosen for the system, which gives us a 15.79 m² cross sectional area for each cell (7917 m² overall). Each fan moving the air through its cell is 34 ft in diameter, consuming 104 kW in power. Overall, 3131 kW of energy is used to power the fans.

6.4.2 Air-Cooled Condenser

This system would not use a shell and tube heat exchanger. Instead, the system would use a system modeled off GEA’s Aircooled Condenser². Steam is split into numerous small tubes, to increase surface area for heat transfer. Air is then blown over the tubes by numerous fans, removing heat from the system.

---

¹ (AG, 2007)

² (AG, 2007)
by means of a temperature differential. Each tube has fins placed around it to ensure turbulent air flow for proper heat transfer.

This system is intended to be a last resort, in the event that there was a minimal amount of water nearby. Costing for this system is not nearly so easy, as it has operating costs as a function of power supplied to the fans. The GEA website stated above provides a total energy use of 194.6 kW per fan, with a total of 102 fans used.

In addition, the heat transfer coefficient for a system involving air and steam is decreased by a factor of 22 (50 W/m$^2$-K)$^1$, in comparison to water and steam (1100 W/m$^2$-K). This means that a much larger area must be used for heat exchange:

$$Q = (U)(A)(\Delta T_{lm})$$  

Given the same $Q$, and a $\Delta T_{lm}$ of about 20 Celsius, this gives an area of about $1.26 \times 10^6$ m$^2$.

\[1 \text{ (The Engineering Toolbox)}\]
7 Component Specification

7.1 Turbines

7.1.1 Steam Turbines

When we determined the type of steam turbine that we are going to use in our power plant, we had several different design specifications that we had to meet in order to achieve our goals.

The first and most prevalent was that it had to meet the power requirement of 1 GW of power output to the grid. This means that it has to be able to exceed this 1 GW threshold by a significant margin in order to also supply power to internal components such as the pumps and, in our clean coal option, the oxygen plant.

The second design specification was that the turbine needed to have the capability of having multiple stages of reheats. This is a very important aspect because it has a very large impact on overall system efficiency. When performing system analysis we were able to determine exactly how reheats directly effects the system efficiency. These results can be seen in Table 11. An added benefit to adding a turbine with reheats is that it will reduce the pollution of the plant due to increased efficiency. Because decreasing total emissions of our plant is one of the major goals of our project this is a very important topic to consider.

When choosing an appropriate turbine for our plant, we had to find certain variables in our thermodynamic model to help specify the best turbine. Because every turbine is unique and built to work under certain operating conditions, we wanted to make sure the turbine that we specified matched closely to our operating conditions. The four variables for consideration for the steam turbine that our team considered were the output range, main steam temperature and pressures, the reheat steam temperature and pressure, and the exhaust properties. These four variables are the most important variables that need to be considered for our project and are the most significant features that Siemens and GE talk about when refereeing to their steam turbines.\(^1\) There are many other technical variables that would be considered when going into greater depth of designing a steam turbine such as the lubricating system, blade lengths, and compression ratios between blade stages but these variables are outside the scope of our project. The values for the four variables that we calculated can be seen below in Table 11.

\(^{1}\) (Siemens)
Table 11: Steam Turbine Specification Variables

<table>
<thead>
<tr>
<th></th>
<th>Temperature (K)</th>
<th>Pressure (kPa)</th>
<th>Power Output (GW)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Main Stream</td>
<td>873.2</td>
<td>15000</td>
<td></td>
</tr>
<tr>
<td>Reheat Stream 1</td>
<td>873.2</td>
<td>2806</td>
<td></td>
</tr>
<tr>
<td>Reheat Stream 2</td>
<td>873.2</td>
<td>380</td>
<td></td>
</tr>
<tr>
<td>Exhaust Steam</td>
<td>452.9</td>
<td>10</td>
<td></td>
</tr>
<tr>
<td>Output Range</td>
<td></td>
<td></td>
<td>1.4</td>
</tr>
</tbody>
</table>

7.1.2 Gas Turbines

When conducting research for steam turbines, we also came across much information about gas turbines that are going to be used in the integrated gasification combined cycle plant. Because this information would be useful to another team in our group we recorded and shared the information that we found. The most important information recorded were the variables for consideration when specifying gas turbines. These variables are: exhaust temperature, exhaust mass flow, net heat rate, pressure ratio and gross power output. Because our team was not focusing on gas turbines we do not have values for these variables.

7.2 Coal Processing

7.2.1 System Requirements

In order to burn the coal received as thoroughly as possible the coal must be pulverized such that at least 70 percent of the particles will pass through a 200-mesh sieve.¹ Pulverization of the coal allows for all the carbon particles to be exposed to oxygen thus burning completely as possible. It also increases the surface area per mass ratio which helps to significantly decrease the time that it takes the coal to completely burn. Three types of coal pulverizers were analyzed in order to determine which pulverizer will work best for our conventional coal fired base case power plant.

7.2.2 Alternative Designs

The first type of pulverizer analyzed was the “Ball Tube Mill” pulverizer. The main component of a ball tube mill pulverizer is a large drum with an extended hollow section at each end. The extended hollow sections, called trunnions, allows for the inflow and outflow of the coal chunks and pulverized coal respectively. The trunnions also provide a bearing surface for which the drum can be rotated on. The drum is then filled with steel balls nearly up to the base of the trunnions. The coal is pulverized when the drum is rotated and the balls are allowed to be continuously lifted and dropped on the coal. The

¹ (EPA,1993)
pulverized coal is extracted by forcing air through the pulverizer which carries the dust like pulverized coal into the boiler. Typically, classifiers are located on each end of the mill which act as size separators in order to ensure the exit stream is completely pulverized. Coal particles that are too big are separated out gravitationally and are added to the inlet stream to be re-processed. It should also be noted that the drum is pressurized. The pressure will be determined based on required inlet boiler pressure. This is in order to minimize the wear on the mill typically found in exhausters on suction mills.\(^1\)

There are several advantages and disadvantages to the ball tube mill pulverizer. The ball tube mill has a high throughput capability which minimizes the total number of mills required per plant. It also features the lowest maintenance cost in comparison to other types of pulverizers. This can be attributed to the lifespan of the mill liners which typically last from 15-20 years. Some other significant advantages are that the balls can be replenished during operation (due to its slow operating speed) and foreign metals in the coal can be ignored. Units also include two feeders and two crusher dryers which allows for full production should one side (one feeder and one dryer) fail. \(^2\)

Disadvantages of using a ball tube mill pulverizer include “high initial cost, high power input, and large floor-space requirements…”\(^3\) There is no information on the amount of energy that it takes to pulverize coal just a plethora of information stating it’s an energy intensive processes.

Another type of pulverizer analyzed was a high speed impact mill. It is the most basic type of coal pulverizer. The coal is crushed by hammers mounted on a shaft and then sent into a grinding chamber. A major advantage of this type of mill is the low initial cost of the units. A disadvantage of this type of mill is its “susceptibility to damage by foreign materials.”\(^4\)

The final type of pulverizer analyzed is a medium speed roller type. Coal is fed at the top dead center of the mill and allowed to fall in the center of a rotating grinding plate. The rotating action of the plate forces the coal to the outside of the plate. Positioned over the grinding plate are series of rollers that crush the coal as they move outward. Dense foreign materials are allowed to fall through a nozzle ring and are continuously scraped into a chamber for disposal.\(^5\)

A main advantage of the roller type pulverizer is its relatively low energy consumption. “The roller can grind material on the pulverizer stones directly, so the consumption of power is only 30%-40% of ball

\(^1\) (Norman K. Trozzi, 1991)
\(^2\) (D. McDonald and D. DeVault, 2007)
\(^3\) (Integrated Publishing)
\(^4\) (Integrated Publishing)
\(^5\) (Schumacher)
Another advantage is the small space requirement of the roller mill. Due to its vertical design the vertical pulverizer uses only 50% of the floor space required for a ball tube mill pulverizer.

A disadvantage of the roller type pulverizer is its high initial cost. Operations and Maintenance costs are also comparatively high due to the uneven wear of grinding parts and the complexities of the system.\(^1\)

Another requirement for the pulverizer is that it can deliver coal to our boiler under pressure. This is due to the fact that the burner that we will be using for our boiler requires it. The way that that this will be achieve is through a pneumatically driven pressures system that will drive pulverized coal straight from the pulverizer to the burner of the boiler. The coal needs to be delivered to the boiler at a pressure between 3 – 7 atm.\(^3\) These types of systems are very easily added onto most type of pulverizers.\(^4\)

An additional requirement is that the pulverizer can deliver the appropriate amount of pulverized coal at a certain mesh and the correctly volumetric flow rate. Different styles of pulverizers can handle larger flow rates than others so we had to make sure that the one we chose could supply enough coal to keep our burners going.

### 7.2.3 Final Design Results

The pulverizer that we decided to use in our power plant is the medium speed roller. Due to the low energy consumption, compact size, high coal output, and ability to crush coal to our desired mesh makes it the best option for our system. When specifying the pulverizer for the power plant we will need to inform the manufacture of two design variables. These variables are the 200 mesh size we need the coal to be and the flow rate out of the pulverizer of 353.7 ton/h. We know that we need to have our pulverizer outputting coal that is 200 mesh because it is the required size for the burner type within our boiler. We also know that we need a coal flow rate of 353.7 ton/h based on the heating value of the Illinois 6 coal that will be burning, the power output of our plant and the overall efficiency of the plant from our model. This flow rate is well within other range of existing coal power plants.

### 7.3 Regenerative Heat Exchangers

Two Heat Exchangers, E-001 and E-002, will be used to condense steam that comes off turbines T-000 and T-001. In addition, these will be providing preheating for the condensed water coming out of the condenser. Costs associated with the heat exchanger are a function of the heat transfer area. This in turn is calculated from the temperatures entering and leaving the exchanger. The method for retrieving the surface area is explained in section 6.2 Condenser System Requirements, and appendix (4) contains the formulas which are used to calculate the heat transfer area.

---

1. (BinQ, 2010)
2. (Chattopadhyay, 2004)
3. (Boiler Reference Book)
4. (Danieli Corus)
Streams S-010B and S-012B are coming off of the first and second turbines (T-000 and T-001) respectively. S-002 is coming from the condenser (E-000).

7.3.1 Closed Feed-water Heater 1

A heat flow of 335 MW is transferred through this heat exchanger. Steam passes through at 132.2 kg/s, and cooling water passes at 474 kg/s. A pipe size of 1 in BWG 12 was selected here, to give more surface area in a smaller length. This gives us an inner diameter of 19.8 mm, outer diameter 22.6 mm. a slower flow rate of 1 m/s was selected for the tube side, in order to let us use more tubes to get a more compacted heat exchange area. This gives us approximately \( n = 1500 \) tubes in the exchanger.

\[
n = \frac{\text{Total volumetric flow}}{1 \frac{m}{s} \times \text{Area of Pipe}}
\]

In addition, flow of 0.6 m/s was taken for the shell side of the exchanger, giving us a shell diameter of 1.03 m. This is calculated by taking the 0.22 m\(^2\) area needed, and adding in the additional area already occupied by the tubes.

Unisim models of the steam cycle provided us with the temperatures through the condenser:
Table 12: Temperature through condenser

<table>
<thead>
<tr>
<th>In degrees Celsius</th>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam</td>
<td>329.6</td>
<td>141.8</td>
</tr>
<tr>
<td>Cooling Water</td>
<td>46.88</td>
<td>205</td>
</tr>
</tbody>
</table>

From this, $\Delta T_{lm}$ can be calculated to be used for the equations. In addition, a correction factor $F_t$ can be found for the shell-and-tube heat exchanger, which is a measurement of the efficiency of heat transfer. This is determined by reading the data off graphs in Seider\textsuperscript{1}. In this situation, $F_t$ was found to be extremely low for the regular 1-2 (one shell pass, two tube passes) exchanger, so this one is modeled as a 2-4 (two shell passes, four tube passes) exchanger, which drastically improves the efficiency.

In addition, Unisim provided us with physical properties of the varying streams (viscosity, heat capacity, etc.). All of this data was then input into Error! Reference source not found. Appendix 4: Heat Exchanger Calculations. This sheet made calculations to determine the overall heat transfer coefficient, and from that, the heat transfer area. This in turn can be used to determine the length of pipe through the exchanger.

Final designs are:

Table 13: Specifications for Exchanger E-001

<table>
<thead>
<tr>
<th>Pipe Specs</th>
<th>1 in. BWG 12</th>
</tr>
</thead>
<tbody>
<tr>
<td># Tubes per exchanger</td>
<td>1500</td>
</tr>
<tr>
<td>Shell diameter</td>
<td>1.03 m</td>
</tr>
<tr>
<td>Heat transfer area</td>
<td>2685 m$^2$</td>
</tr>
<tr>
<td>Tube length</td>
<td>28.8 m</td>
</tr>
<tr>
<td>Heat transferred (Q)</td>
<td>335 MW</td>
</tr>
<tr>
<td>$\Delta T_{lm}$</td>
<td>109.1</td>
</tr>
<tr>
<td>Heat Trans Coeff. (U)</td>
<td>1315 W/m$^2$-K</td>
</tr>
<tr>
<td>Correction factor ($F_t$)</td>
<td>0.87</td>
</tr>
</tbody>
</table>

\textsuperscript{1} (Seider, 2009)
7.3.2  Closed Feed-water Heater 2

A heat flow of 33.6 MW is transferred through this heat exchanger. Steam passes through at 16.05 kg/s, and cooling water passes at 606.2 kg/s. A pipe size of 1 ½ inch BWG 9 was selected for the condenser, which gives a pipe inner diameter of 30.48 mm, outer diameter of 38.1 mm. Heuristics\(^1\) suggest that flow through the tubes should be about 2 m/s, which gives us approximately \(n = 420\) tubes per exchanger.

\[
    n = \frac{\text{Total volumetric flow}}{2 \frac{m}{s} \times \text{Area of Pipe}}
\]

In addition, flow of 0.6 m/s was taken for the shell side of the exchanger, giving us a shell diameter of 0.803 m. This is calculated by taking the .0268 m\(^2\) area needed, and adding in the additional area already occupied by the tubes.

Unisim models of the steam cycle provided us with the temperatures through the condenser:

**Table 14: Temperature through condenser**

<table>
<thead>
<tr>
<th></th>
<th>In degrees Celsius</th>
<th>In</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>Steam</td>
<td>346.7</td>
<td>230.8</td>
<td></td>
</tr>
<tr>
<td>Cooling Water</td>
<td>192.1</td>
<td>203.7</td>
<td></td>
</tr>
</tbody>
</table>

From this, \(\Delta T_{lm}\) can be calculated to be used for the equations. In addition, a correction factor \(F_i\) can be found for the shell-and-tube heat exchanger, which is a measurement of the efficiency of heat transfer. This is determined by reading the data off graphs in Seider\(^2\).

In addition, Unisim provided us with physical properties of the varying streams (viscosity, heat capacity, etc.). All of this data was then input into Appendix 4: Heat Exchanger Calculations. This sheet made calculations to determine the overall heat transfer coefficient, and from that, the heat transfer area. This in turn can be used to determine the length of pipe through the exchanger.

Final designs are:

\(^1\) (Shell & Tube Heat Exchanger Design)

\(^2\) (Seider, 2009)
Table 15: Specifications for Exchanger E-001

<p>| | |</p>
<table>
<thead>
<tr>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Pipe Specs</td>
<td>1 ½ in. BWG 9</td>
</tr>
<tr>
<td># Tubes per exchanger</td>
<td>420</td>
</tr>
<tr>
<td>Shell diameter</td>
<td>1.03 m</td>
</tr>
<tr>
<td>Heat transfer area</td>
<td>473 m²</td>
</tr>
<tr>
<td>Tube length</td>
<td>10.57 m</td>
</tr>
<tr>
<td>Heat transferred (Q)</td>
<td>33.6 MW</td>
</tr>
<tr>
<td>$\Delta T_{lm}$</td>
<td>79.8</td>
</tr>
<tr>
<td>Heat Trans Coeff. (U)</td>
<td>935.3 W/m²-K</td>
</tr>
<tr>
<td>Correction factor (F₁)</td>
<td>.95</td>
</tr>
</tbody>
</table>

7.4 Mixing Section

The mixing sections in our system are going to be used to mix cold water coming from the feed water with warm water existing a heat exchangers supplied by steam from the turbine. The goal of these mixing section is to evenly heat the preheat stream to the boiler. This even mixing will be achieved by using static mixers to combine the two streams.¹ These static mixers use helical patterns within the pipes to encourage axial mixing between the two streams.² Axial mixing is a very effective way of combining two steams in a continuous system because it does not need to enter into a tank to be mixed using blades or other device. To specify a static mixer to a manufacture, we will have to supply to them the flow rates of the two streams we would like to combine along with the temperature and pressures of these streams. The operating conditions for our system can be seen in Table 16: Mixer Operating Conditions.

Table 16: Mixer Operating Conditions

<table>
<thead>
<tr>
<th></th>
<th>Flow Rate Hot [kg/s]</th>
<th>Flow Rate Cold [kg/s]</th>
<th>Temperature Hot [C]</th>
<th>Pressure Hot [kPa]</th>
<th>Temperature Cold [C]</th>
<th>Pressure Cold [kPa]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mixer 1</td>
<td>473.98</td>
<td>475896.67</td>
<td>205.1</td>
<td>15000</td>
<td>143.6</td>
<td>15000</td>
</tr>
<tr>
<td>Mixer 2</td>
<td>606.18</td>
<td>16.05</td>
<td>203.7</td>
<td>15000</td>
<td>234.1</td>
<td>15000</td>
</tr>
</tbody>
</table>

¹ (Static Mixers)

² (Axial Mixing)
7.5 Pumps

There are many different types of pumps that are used in power plants but only two design variables that need to be considered. The first is the flow rate that the pump needs to provide to the component. The second is the operating conditions the pump will be enduring. Due to the different natures of the variety of streams within the power plant we will need to have pumps that can withstand high pressures, high temperatures and high volume depending on where it is located. There were to options of pumps that could provide the appropriate flow rate and operating parameters for our plant. These were the centrifugal and rotary pumps. However, after further research it was found that the centrifugal style pumps are the industry standard within power plants. This is due to the fact that this type of pump can be better modified to stand the high heat and pressures with a number of different types of sealing mechanisms.

The three main pumps and their operating conditions for our system can be seen below in Table 17.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Pump 1</td>
<td>320</td>
<td>20</td>
<td>15000</td>
<td>610.6</td>
<td>8.33</td>
</tr>
<tr>
<td>Pump 2</td>
<td>417</td>
<td>380</td>
<td>15000</td>
<td>604.5</td>
<td>2.38</td>
</tr>
<tr>
<td>Pump 3</td>
<td>506.5</td>
<td>2806</td>
<td>15000</td>
<td>476.6</td>
<td>0.11</td>
</tr>
</tbody>
</table>

7.6 Generator

The generator is another important piece of machinery within the power plant. It is what turns mechanical energy into electrical energy. This process is achieved through the work of a very large scale electric generator. There are several companies that make electric generators for power plants. For our power plant we will have an electric generator that will operate at 60 Hz to match the power grid within the United States. We also will have a generator that is water cooled rather than air cooled because it offers great heat transfer properties. Since our power plant will be located on the edge of Lake Michigan water will not be a problem. Another property that is desired in a generator is that it provides us sufficient apparent power ratings so we can meet our 1 GW power output goal. It also should have a 99% efficiency.
rating along with a 0.85 power factor or greater which have become expected performance values in new generation electric generators.  

### 7.7 Piping and Valves

The importance of piping and valves within a power plant are often over looked and underappreciated even though they are just as important as any other piece in a plant. When looking to specify piping and valves in a plant there are several things to consider. The first is the type of piping that will be used within the plant. The most popular is seamless steel piping. Seamless steel piping offers the best quality because it is more resistant to fatigue, can be treated to handle many different types of materials, and can be easily manufactured. Another design specification to keep in mind is the size and length of pipe sections. Creating large and long pipes becomes expensive very quickly because of the manufacturing and installation costs. In additions to these specifications, you always want to consider what type of insulation that will be used on pipes. Insulation is very important because it helps to protect against heat loss when transferring steam from boiler to turbine. Finally, connection points are a major design feature to think about. There are many different ways to connect joints such as welding, bolting, and chemical sealing. Each option offers a unique advantage in different environments. These design specifications also hold true when determining valves in our plant.

---

1 (Siemens Generators)
8 Control System Considerations

8.1 System Requirements

There are three main types of control system components that we will need to implement within our design to help monitor and help maintain steady operating conditions for our plant. The three measurement devices that will be used are thermocouples in order to measure temperature of streams, pressure gages to measure pressure, and flow rate gages to measure mass flow rate. These key components will be connected to different valves that will be able to control the flow of different streams therefore enabling them to be able to lower the downstream temperature, pressures and flow rates they are associated too.

8.2 Systems Controllers

8.2.1 Boiler Control

Due to the variability of environmental conditions and output of our plant it is important that we can control the temperature of the streams exiting the boiler. In order to do this we will need to control the flow rate of coal and air going into the boiler. This will be achieved by placing thermocouples on the entrance and exist of all the streams involved with the boiler. With the information gathered from these thermocouples, we will be able to tell whether we need to increase the flow rate of coal in order to increase or decrease the operating temperature.

8.2.2 First and Second Stage Turbine Pressure Control

The amount of pressure drop that steam encounters after it exits a stage of a turbine can vary with operating conditions. To compensate for this, the pressure exiting the first stage of the turbine needs to be controlled in order to ensure the proper pressure for the next reheat. This is done by placing a pressure gage at the exit of the turbine stream going to the boiler. This pressure gage will be connected to a controller that operates a correlating pump that will be able to increase pressure depending on the pressure readings received. The pressure stream exiting the second stage of the turbine will be controlled in the same way.

8.2.3 Regeneration Flow Rate Control

Due to the change in temperature to both the steam exiting the turbine and the feed water flow due to environmental and operating conditions the pressure and temperature of streams flowing into the regeneration needs to be controlled. This will be achieved by placing thermocouples at the entrance and exist of both the hot and cold streams of regeneration one and two. In addition the pressure exiting the hot stream will also be taken. These readings will be fed back to a valve that controls the amount of steam that is diverted from the exit stream of the turbine so that the feed water stream can be raised to the sufficient temperature.

8.2.4 Exit of Turbine Pressure Control

The pressure of the steam exiting the final stage of the turbine will be controlled using a pressure gage connected to a pump. This pump will ensure that there is sufficient pressure to flow the steam through the
condenser. It will do this by varying its flow rate depending on the readings it receives from the pressure gage.

8.2.5 Condenser Control
Because the environmental temperatures fluctuate from season to season it is important to be able to measure these because it plays such a large part in the condenser. Depending on the temperature of the cold water flowing into the condenser, this will change the amount of steam that it can cool. In order to determine the correct flow, thermocouples will be placed at the entrance of the condenser in the hot side as well as at the exit of the hot and cold side. By measuring the temperature drop in the hot stream we can determine how much additional water that needs to flow through the cold side. This will be controlled using a valve that will receive all of the data.

8.2.6 Makeup Water Control
Due to the fact that our system will be continually losing water from normal operating conditions it is important to replace this water. This is achieved through a flow meter and a valve. The flow meter is located along the rankine cycle and measures the amount of water that is flowing through the system. If low it opens a valve that allows sufficient water to enter into the makeup water process to that it can bring the system back up to sufficient water flow.

8.2.7 Makeup Water Pressure Control
Because the makeup water has to enter through several different machines that cause large pressure drops within them it is vital to maintain sufficient pressure to keep the makeup water stream flowing. This will be controlled through a set of pumps and pressure gages. The order of operation is that water flows through a valve and enters a ultrafiltration device. This water is then sent through a pump where the pressure is measured at the first pressure gage. It then enters a deaerator followed by a reverse osmosis machine. It then enters a second pump followed by another pressure gage. Finally this completely clean water is added back into the system. These pressure gages ensure that the pumps are pressurizing the stream to sufficient pressures.
Figure 18: PFD of System with Control Systems
9 Environmental Equipment

9.1 Fly Ash Removal

It is important to remove the Fly ash from the flue gas due to major negative public health effects, and damage to equipment. This is done by installing an Electrostatic Precipitator, which attracts the fly ash by inducing a charge on flat plates parallel to the flow of the flue gasses. These plates become covered in the fly ash, and are periodically shaken, so that the ash falls into collectors placed below.

![Figure 19: PFD of Environmental Equipment](image)

9.2 CO₂ and Sulfate Removal

In order to make conventional coal power plants run and operate more clean there are several different types of environmental equipment that can help to capture pollutants before they enter the atmosphere. One type of environmental equipment that is used is CO₂ scrubbers that are placed within the flue stacks of power plants. The two types of carbon dioxide scrubbers that are most widely used are chemical and physical absorption methods. The chemical method uses amine groups to chemically bond to CO₂ where it can be removed in CO₂ streams. The second method used, the physical absorption method, is done by dissolving CO₂ in a solvent. The industry standard for this solvent is Selexol because of its favorable properties. Similarly to the physical absorption to CO₂, hydrogen sulfide is removed using an absorption tower with Selexol.
9.3 Reduction of Nitrogen Oxides

In order to reduce the amount of nitrogen oxides that exit through the plant’s flue gas, some environmental equipment that we can implement to reduce our levels is selective catalytic reduction. This is done by Selective Catalytic Reduction, using ammonia to reduce NOx gases to elemental nitrogen and water over a catalyst.

9.4 Mercury Removal

There are two methods that our plant could implement in order to significantly reduce the amount of mercury that is produced from burning the large amounts of coal to run our plant. The two options are electrostatic precipitator plus flue-gas desulfurization or selective catalytic removal in combination with FGD and ESP.

9.5 Sequestration

A final option for reducing the amount of pollutants our plant produces doesn’t directly involve implementation of clean equipment and technologies but rather is a band aid solution of pumping CO2 deep into the crust of the earth. This method for getting rid of carbon dioxide is called sequestration. There are three options of locations that offer good sequestration locations: Depleted oil and gas fields, terrestrial biological, and ocean sequestration. Each one of these options provides ample room to store thousands of tons of carbon dioxide; however, the drawback to this option is that the power plant has to be relatively close to the location that will sequester too. With this drawback and the fact that there are not very many sequestration locations located within the United States, it doesn’t make it a very realistic option.
10 Costing of Equipment

10.1 Boiler Components

Due to the limitation of information on costing of Pulverized Coal Boilers, we researched scaling functions that we found in Seider’s “Product and Process Design Principles” textbook. Using this formula, we found the cost of the boiler was $150 million.

10.2 Water Filtration System

This section contains costing estimates for the three pieces of section 5 Water Purification Systems. In addition, cost estimates for the alternative case of a cooling tower used for Condenser E-000 as described in section 6.3 Condenser Alternative Designs are provided. This is to provide costing context, as well as a source of future reference, in the event that a cooling tower is desired for additional power plants.

10.2.1 Ultrafiltration system

Approximately 200 gallons per minute (0.288 million gallons per day) will be passing through the ultrafiltration system, as seen in Figure 11. Studies have shown costs as seen in Table 18.

<table>
<thead>
<tr>
<th>Design Flow (mgd)</th>
<th>0.01</th>
<th>0.1</th>
<th>1.0</th>
<th>10</th>
<th>100</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Flow (mgd)</td>
<td>0.005</td>
<td>0.03</td>
<td>0.35</td>
<td>4.4</td>
<td>50</td>
</tr>
<tr>
<td>Capital Cost ($/gal) (^1)</td>
<td>$18.00</td>
<td>$4.30</td>
<td>$1.60</td>
<td>$1.10</td>
<td>$0.85</td>
</tr>
<tr>
<td>Annual O&amp;M Cost ($/kgal) (^2)</td>
<td>$4.25</td>
<td>$1.10</td>
<td>$0.60</td>
<td>$0.30</td>
<td>$0.25</td>
</tr>
</tbody>
</table>

At this flow rate, capital costs are $3.736/gpd. This gives us a capital cost of $1.08 million.

10.2.2 Deaerator

Cost of the deaerator is based off the size of the vessel, as well as the cost of the steam used in the system to strip the oxygen and carbon dioxide from the water. Steam has a cost of $3.00/1000 lb. A flow rate of 22.4 grams steam per 1000 kg deaerator capacity was specified\(^2\). By unit conversions, and a capacity of 200 gpm, this specifies a steam flow of 0.0372 lb/min, or 17,850 lb/year. This provides us with a process

\(^1\) (Microfiltration/Ultrafiltration)

\(^2\) (Sarco, 2012)
cost of $53,544/year. In addition, purchasing cost of the vessel itself are determined to be approximately $20,500\(^1\).

10.2.3 Reverse Osmosis Membrane Separator

Approximately 200 gallons per minute will pass through this system. Operating costs are a function of the cost of replacement of the membrane, as well as the cost of pumping up the water to the required pressure to get the necessary flux through the filter. For capital costs, prices of RO systems were found online, and plotted against their flow rates through the system. This plot is shown in Figure 20.

![Figure 20: Purchase Costs of Reverse Osmosis Systems available used](image)

In accordance with our final design, this would give us a purchase cost of $154,000 for the system.

The Air-Cooled Fin-Fan condenser design would require the same amount of flow through the Reverse Osmosis separator, so cost estimates will be identical.

10.2.4 Overall system costs

Overall, the combined cost of water cleaning is:

| Table 19: Cost of Overall Water Purification System |
|------------------|------------------|
| Purchase Cost    | Variable costs   |
| $1,300,000       | $53,000/year     |

\(^1\) (Matches, 2003)
Only the cost of the steam is included in the variable costs. Additional costs associated with operations for all three parts of the systems are calculated at a latter point as a function of capital costs, to keep them consistent with the other components of the power plant.

### 10.3 Condenser Options

This section contains capital and operating cost estimates for the condenser operations. It does not include the cost of pumping the water and purifying it, as that cost is dealt with in section 9.5 Water Filtration Systems. All three condenser systems: Standard Flow, Cooling Tower, and Aircooled Condenser, will be discussed in this section.

#### 10.3.1 Selected System

![Condenser PFD](image)

**Figure 21: Condenser PFD**

Our selected system, which simply uses a condenser with a constant stream of new water from Lake Michigan, only has a capital cost. As was stated in section 6.3 Condenser Final Design Results, the heat
transfer area of the condenser is 77800 m$^2$. As this system is modeled as four parallel shell and tube heat exchangers, it was costed using the same functions as the heat exchangers.

To find a costing method for the heat exchangers that we will be using in our power plants we pulled costing functions from the book Product & Process Design Principles.$^1$ This book gives equations on how to estimate the cost for shell-and-tube heat exchangers depending on certain variables that were determined for our system. These variables include the surface area for our heat exchangers (A) and the pressure inside the sheel-side (P). It also provided certain variable information depending upon the different types of material that will be used. The following equations were given.

\[
C_p = F_p F_M F_L C_B \\
C_B = \exp\{11.0545 - 0.9228[\ln(A)] + 0.09861[\ln(A)]^2\} \\
F_M = a + \left(\frac{A}{100}\right)^b \\
F_p = 0.9803 + 0.018\left(\frac{P}{100}\right) + 0.0017\left(\frac{P}{100}\right)^2 \\
F_L = 0.79
\]

In the previous equations $C_p$ is the final cost of the heat exchanger, $C_B$ is the type of heat exchanger, $F_M$ is the material factor, $F_p$ is the pressure factor, and $F_L$ is the tube length correction factor. In our heat exchangers we used the material carbon steel/stainless steel which gave us the values $a = 1.75$ and $b = 0.13$.

**Table 20: Variables for Capital cost of one condenser (out of four identical)**

<table>
<thead>
<tr>
<th>Area (ft$^2$)</th>
<th>209,000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure (psig)</td>
<td>1.45</td>
</tr>
<tr>
<td>$C_B$</td>
<td>2,080,000</td>
</tr>
<tr>
<td>$F_M$</td>
<td>4.45</td>
</tr>
<tr>
<td>$F_p$</td>
<td>0.98</td>
</tr>
<tr>
<td>$F_L$</td>
<td>0.79</td>
</tr>
<tr>
<td>$C_p$ ($)</td>
<td>7,190,000</td>
</tr>
</tbody>
</table>

Using this equation, we get a cost of purchase for all the condensers of $28.8 million.

---

$^1$ (Seider, 2009)
10.3.2 Cooling Tower

Capital costs for the cooling tower are taken from Peters\(^1\), where cost for the tower is modeled as a function of flow through the tower, and temperature differential between the in and out streams.

<table>
<thead>
<tr>
<th>Purchase Costs ($)</th>
<th>Temperature Range</th>
</tr>
</thead>
<tbody>
<tr>
<td>( L (\text{M}^3/\text{s}) )</td>
<td>5.5</td>
</tr>
<tr>
<td>0.2</td>
<td>75000</td>
</tr>
<tr>
<td>0.3</td>
<td>90000</td>
</tr>
<tr>
<td>0.4</td>
<td>100000</td>
</tr>
<tr>
<td>0.5</td>
<td>120000</td>
</tr>
<tr>
<td>0.6</td>
<td>140000</td>
</tr>
<tr>
<td>0.7</td>
<td>160000</td>
</tr>
<tr>
<td>0.8</td>
<td>170000</td>
</tr>
<tr>
<td>0.9</td>
<td>190000</td>
</tr>
<tr>
<td>1</td>
<td>200000</td>
</tr>
<tr>
<td>2</td>
<td>340000</td>
</tr>
<tr>
<td>3</td>
<td>500000</td>
</tr>
</tbody>
</table>

Our flow through the cooling tower was found to be 20 m\(^3/\text{s}\), which provides a range of 15 degrees Celsius. Extrapolating from this data gives us a capital cost of $3.51 million. These costs will be added to the overall cost of the condenser, for a total cost using this system of $33.5 million.

Fans must be powered to force the air up against the flow of water. Using the calculator from GEA\(^2\), a fan power of 1655 kW was determined.

10.3.3 Aircooled Condenser

As stated in the Condenser specifications, an estimated heat exchanger area for the aircooled condenser would be approximately \(3.7 \times 10^6 \text{ m}^2\). Using this value, a cost of purchase for the system was estimated using the same method of the shell and tube heat exchanger. It was noted that if the condenser was broken down into a number of smaller condensers, with smaller identical heat transfer areas, the cost could be reduced compared to having just one large exchanger. An optimal minimized cost was found at 1000 small condensers, with heat exchanger area of 1260 m\(^2\).

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\(^1\) (Peters, 2003)

\(^2\) (AG, 2007)
Using the shell and tube heat exchanger area above, a total purchase cost of the system of condensers gave a cost of $206 million. This is notably much higher than a water cooled condenser, which is why we would not use this one unless we had no other choice.

10.4 Turbines

10.4.1 Steam Turbines

Determining the costing for the steam turbine was a very challenging process due to the secrecy of steam turbine operations and the fact that companies would not take calls or return messages when we tried to get cost estimates from them. Company policies were not to deal with students because it was looked upon as a waste of time because we were not potential customers. To make the process even more difficult, we could not find any cost estimating functions that looked legitimate or find costs of brand new steam turbines. However after much searching on the internet we were able to find prices for used steam turbines. These turbines varied in age and amount of hours that they had been running but it was something that we could go on. The data that we were able to record were for steam turbines with zero, one, and two reheats with a power generation range from 0 – 350 MW. Because our power plant is going to require a steam turbine with 4 reheats and generate 1.4 GW of power we had to perform a series of extrapolations in order to achieve a desired data series. We did this by finding the cost difference between the one and two reheats to get to a four reheat turbine. In order to get to a 1.4 GW turbine we continued our data series using the rule of thumb that equipment costs go up by a power of .6 as their sizes get bigger. After this was done we were able to plot our data. This can be seen in Figure 22:Cost of Used Steam Turbine.

![Figure 22: Cost of Used Steam Turbine](image)

This information was very helpful but it was still only the cost for a used turbine. We now had to somehow determine how to make this graph be an accurate representation of new steam turbines. We
conducted some initial research but were not able to come up with any results. After talking to our industrial consultant however, he informed us that used industrial equipment generally sells for about half the price of new equipment. Using this guideline we plotted our data again and used a best fit line to determine a costing function for our steam turbines. This graph can be seen below.

![Figure 23: Cost of New Steam Turbine](image.png)

10.4.2 Gas Turbines

In our search to find costing information for the steam turbines to be used within our conventional coal power plants we also came across sufficient enough information to develop a curve for gas turbines. The type of gas turbines that we found information for can be used for the internal gasification combined cycle power plants which is being research be another team. This information was turned over to them so they can use it to cost the turbines in their plants. A graph of the data we collected for the gas turbines can be seen in Figure 24: Cost for New Gas Turbine.
There are three different types of mills that are used in industry to process coal so that it can be burned in a boiler. As a result we research and found a cost associated with them. When researching them we found that the way they are specified is in ton/hr. The following graph was made. The costing functions that we used for the AG/SAG because it met our 200 mesh requirements along with the coal output.
10.6 Heat Exchangers

To find a costing method for the heat exchangers that we will be using in our power plants we found costing functions from the book Product & Process Design Principles. This book gives equations on how to estimate the cost for shell-and-tube heat exchangers depending on certain variables that were determined for our system. These variables include the surface area for our heat exchangers (A) and the pressure inside the shell-side (P). It also provided certain variable information depending upon the different types of material that will be used. The following equations were given.

\[
C_P = F_P F_M F_L C_B
\]

\[
C_B = \exp\{11.0545 - 0.9228[\ln(A)] + 0.09861[\ln(A)]^2\}
\]

\[
F_M = a + \left(\frac{A}{100}\right)^b
\]

\[
F_P = 0.9803 + 0.018\left(\frac{P}{100}\right) + 0.0017\left(\frac{P}{100}\right)^2
\]

\[F_L = 0.79\]

In the previous equations \(C_P\) is the final cost of the heat exchanger, \(C_B\) is the type of heat exchanger, \(F_M\) is the material factor, \(F_P\) is the pressure factor, and \(F_L\) is the tube length correction factor. In our heat exchangers we used the material carbon steel/stainless steel which gave us the values \(a = 1.75\) and \(b = 0.13\).

10.7 Mixing Section

A costing function was not found for the mixing section of our plant. This was not done because it was taken into account by the cost account and capital cost estimations. The percent delivered equipment cost for fluids processing plant takes into account such things as these static mixers. The specific breakdown of Lang Factors that we used was written by Peters, Timmerhaus and West.

10.8 Pumps

To determine the cost for the pumps we used a cost calculator. This calculator had many different industrial components that are not easily found in other places. This cost estimator would allow us to plug

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1 (Seider, 2009)

2 (Hill)
in the most important variable in a component and would give a price for it. We had two options of pumps to choose from when costing the pumps, centrifugal and rotary but for reason discussed earlier we chose to go with the centrifugal pumps. This can be seen in Figure 26: Cost of Pumps below.

![Figure 26: Cost of Pumps](image)

### 10.9 Piping and Valves

It would have been outside the scope of our project to design every pipe and valve system that would be used within our system. However because it is still a cost that cannot be ignored the cost for piping and valves was taken into account when we calculated the total capital costs for the entire plant. This unknown cost was calculated under the construction and installation costs of the plant.

### 10.10 Start-up Costs

These are costs associated with: Labor, materials, equipment, etc. that is only used during the startup of the plant, as well as a loss of income due to the plant not running at full capacity\(^1\). As was shown in Section 2.1.4.1 Purchase Equipment Cost, the start-up costs have been capitalized, to be determined as a fraction of the purchase cost of the components. Generally, these costs are around 25% of the purchase cost. In the case of the Peters factors, this is included in the working capital.

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\(^1\) (Bejan, Tsatsaronis, & Moran, 1996)
11 Project Management

11.1 Distribution of Responsibility

Our group was comprised of three mechanical engineers and one chemical engineer. This combination proved to be a perfect match for what we were trying to accomplish because we had three large mechanical based problems and one large chemical based problem. The three mechanical problems were thermodynamic analysis, boiler design and component costing equation relating to known operating conditions. The chemical problem was the design of condensers and water purification systems. We divide the projects based upon what each of us was most interested in. Surprisingly enough we did not have any overlapping interests and everyone was able to have their first pick of project. When we had our large projects we each developed ideas, goals, and problems we foresaw in our individual assignments. When we first began our projects we working mainly alone but as time progress we quickly saw the benefits of calling upon each other’s strengths to help solves problems or to analyze things from different points of view. So through a well-balanced combination of individual and group efforts we were able to achieve the goals that we set last fall.

11.2 Team Organization

Our team had the unique experience of having to learn how to interact and communicate with two other senior engineering design teams and a geology team in order to finish our project. In order to achieve the goal set last fall we had to have a well-organized and detailed distribution of work within our own team and that of all the other groups.

Within our own group, we had one member who acted as the team leader and the others who performed the tasks of the team. This leader was in regular communication with the other two team leaders discussing and developing a strategy of how to progress in the most effective and efficient way possible. He made sure that work was evenly distributed out and that no one on his team or any of the other team was doing the repetitive work. The other team members were working alongside the team leader and each other in order to accomplish goals in a timely manner.

In order to hold each other responsible of the task we were working on, we had weekly status update meeting. During these times we would discuss progress, problems, and potential new ideas that we had had during the week. If we had a week where little was accomplished we would discuss the reasons why we had a down week and what we were doing in order to make sure that it would not happen again.
12 Conclusion

The goal of this project was to design a conventional coal power plant so that it may be compared to an oxygen-fed coal power plant, and an IGCC power plant with carbon capture. This comparison was made so that we may know the additional cost of cleaner power generation.

A model for the Rankine Cycle to be used was designed to provide the maximum amount of efficiency, approximately 44% overall, while reducing costs. All key components of this system have been designed and/or specified to the best of our abilities, to fit the model of our steam cycle. All specifications pertaining to the components have been included in this report.

Using our cash flow analysis, we were able to determine the capital cost of the conventional plant was $2.3 Billion, while the oxygen case was priced $3.2 Billion. This information, along with the price of the fuel, we were able to determine the cost to produce electricity. This was 0.07 $/kWh for the conventional plant, and 0.09 $/kWh for the Oxy-Coal plant. Comparing this to 0.18 $/kWh for the IGCC plant, we were able to find the final cost to produce cleaner energy using coal as a fuel.

The cost of electricity in Michigan is about 0.10 $/kWh. Since this price is close to the cost of electricity that we found in our conventional plant, this verifies that our base case produces reasonable numbers. Since our cost estimates for the Oxy-Coal and IGCC cases were performed using the same methods, we can conclude that our comparison was accurate.

Since the EPA carbon restrictions announced in March, 2012, the conventional plant that we designed is no longer a feasible option. However, this design still represents the industry standard for currently operating coal power plants in the US. This economic comparison no longer answers the question “What if we were to design a cleaner option?” Now this study answers the question “How will this law affect the price of electricity for future power plants?”
13 Acknowledgements

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Dr. Matthew Heun – Calvin College Faculty

Calvin College Engineering Department

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15 Appendices

15.1 Appendix 1: EES Code of Final Case Calculations

"2 Reheats & 2 Regenerations"
"State Point 0 - Dead State"

\[ T[0] = \text{converttemp}(C, K, 25) \]
\[ P[0] = 10 \text{ kPa} \]
\[ h[0] = \text{enthalpy}(\text{Water}, T=T[0], P=P[0]) \]
\[ s[0] = \text{entropy}(\text{Water}, T=T[0], P=P[0]) \]

"State Point 1 - Condenser Outlet, Pump 1 Inlet"

\[ T[1] = \text{temperature}(\text{Water}, P=P[1], x=x[1]) \]
\[ P[1] = 10 \text{ kPa} \]
\[ h[1] = \text{enthalpy}(\text{Water}, P=P[1], x=x[1]) \]
\[ s[1] = \text{entropy}(\text{Water}, P=P[1], x=x[1]) \]
\[ x[1] = 0 \]
\[ e[1] = \text{h}[1] - \text{h}[0] - T[0] \times (s[1] - s[0]) \]
\[ E_.[1] = \text{convert}(\text{kW, MW}) \times e[1] \times \text{m}_.3 \]

"State Point 2 - Pump 1 Outlet, Closed FWH 1 inlet"

\[ T[2] = \text{temperature}(\text{Water}, P=P[2], h=h[2]) \]
\[ P[2] = 15000 \text{ kPa} \]
\[ s[2] = \text{entropy}(\text{Water}, P=P[2], h=h[2]) \]
\[ e[2] = h[2] - h[0] - T[0] \times (s[2] - s[0]) \]
\[ E_.[2] = \text{convert}(\text{kW, MW}) \times e[2] \times \text{m}_.3 \]

"State Point 3 - Closed FWH 1 Outlet, Mixer 1 inlet"

\[ T[3] = \text{temperature}(\text{Water}, P=P[3], h=h[3]) \]
\[ s[3] = \text{entropy}(\text{Water}, P=P[3], h=h[3]) \]
\[ e[3] = h[3] - h[0] - T[0] \times (s[3] - s[0]) \]
\[ E_.[3] = \text{convert}(\text{kW, MW}) \times e[3] \times \text{m}_.3 \]

"State Point 4 - Mixer 1 outlet, Closed FWH 2 inlet"

\[ T[4] = \text{temperature}(\text{Water}, P=P[4], h=h[4]) \]
\[ s[4] = \text{entropy}(\text{Water}, P=P[4], h=h[4]) \]
\[ E_.[4] = \text{convert}(\text{kW, MW}) \times e[4] \times \text{m}_.2 \]

"State Point 5 - Closed FWH 2 outlet, Mixer 2 inlet"

\[ T[5] = \text{temperature}(\text{Water}, P=P[5], h=h[5]) \]
\[ s[5] = \text{entropy}(\text{Water}, P=P[5], h=h[5]) \]
\[ e[5] = h[5] - h[0] - T[0] \times (s[5] - s[0]) \]
\[ E_.[5] = \text{convert}(\text{kW, MW}) \times e[5] \times \text{m}_.2 \]

"State Point 6 - Mixer 2 outlet, Boiler inlet"

\[ T[6] = \text{temperature}(\text{Water}, P=P[6], h=h[6]) \]
\[ s[6] = \text{entropy}(\text{Water}, P=P[6], h=h[6]) \]
\[ E_.[6] = \text{convert}(\text{kW, MW}) \times e[6] \times \text{m}_.1 \]
"State Point 7 - Saturated Liquid within boiler"
\[T_7 = \text{temperature}(\text{Water}, P=P_7, x=x_7)\]
\[P_7 = P_2\]
\[h_7 = \text{enthalpy}(\text{Water}, P=P_7, x=x_7)\]
\[s_7 = \text{entropy}(\text{Water}, P=P_7, x=x_7)\]
\[x_7 = 0\]
\[e_7 = h_7 - h_0 - T_0*(s_7 - s_0)\]
\[E_{..7} = \text{convert(kW,MW)}*e_7 * m_.1\]

"State Point 8 - Saturated Vapor within Boiler"
\[T_8 = \text{temperature}(\text{Water}, P=P_8, x=x_8)\]
\[P_8 = P_2\]
\[h_8 = \text{enthalpy}(\text{Water}, P=P_8, x=x_8)\]
\[s_8 = \text{entropy}(\text{Water}, P=P_8, x=x_8)\]
\[x_8 = 1\]
\[e_8 = h_8 - h_0 - T_0*(s_8 - s_0)\]
\[E_{..8} = \text{convert(kW,MW)}*e_8 * m_.1\]

"State Point 9 - Boiler outlet, Turbine 1 inlet"
\[T_9 = \text{converttemp}(C,K,600)\]
\[P_9 = P_2\]
\[h_9 = \text{enthalpy}(\text{Water}, T=T_9, P=P_9)\]
\[s_9 = \text{entropy}(\text{Water}, T=T_9, P=P_9)\]
\[e_9 = h_9 - h_0 - T_0*(s_9 - s_0)\]
\[E_{..9} = \text{convert(kW,MW)}*e_9 * m_.1\]

"State Point 10 - Turbine 1 Outlet, Reheat 1 inlet, Closed FWH 2 inlet"
\[T_{10} = \text{temperature}(\text{Water}, P=P_{10}, h=h_{10})\]
\[P_{10} = 2806 \text{ [kPa]}\]
\[s_{10} = \text{entropy}(\text{Water}, P=P_{10}, h=h_{10})\]
\[e_{10} = h_{10} - h_0 - T_0*(s_{10} - s_0)\]
\[E_{..10} = \text{convert(kW,MW)}*e_{10} * m_.1\]

"State Point 11 - Reheat 1 outlet, Turbine 2 inlet"
\[T_{11} = \text{converttemp}(C,K,600)\]
\[P_{11} = P_{10}\]
\[h_{11} = \text{enthalpy}(\text{Water}, T=T_{11}, P=P_{11})\]
\[s_{11} = \text{entropy}(\text{Water}, T=T_{11}, P=P_{11})\]
\[e_{11} = h_{11} - h_0 - T_0*(s_{11} - s_0)\]
\[E_{..11} = \text{convert(kW,MW)}*e_{11} * m_.2\]

"State Point 12 - Turbine 2 outlet, Reheat 2 inlet, Closed FWH 1 inlet"
\[T_{12} = \text{temperature}(\text{Water}, P=P_{12}, h=h_{12})\]
\[P_{12} = 380 \text{ [kPa]}\]
\[s_{12} = \text{entropy}(\text{Water}, P=P_{12}, h=h_{12})\]
\[e_{12} = h_{12} - h_0 - T_0*(s_{12} - s_0)\]
\[E_{..12} = \text{convert(kW,MW)}*e_{12} * m_.2\]

"State Point 13 - Reheat 2 outlet, Turbine 3 inlet"
\[T_{13} = \text{converttemp}(C,K,600)\]
\[P_{13} = P_{12}\]
\[h_{13} = \text{enthalpy}(\text{Water}, T=T_{13}, P=P_{13})\]
\[s_{13} = \text{entropy}(\text{Water}, T=T_{13}, P=P_{13})\]
\[e_{13} = h_{13} - h_0 - T_0*(s_{13} - s_0)\]
\[E_{..13} = \text{convert(kW,MW)}*e_{13} * m_.3\]
"State Point 14 - Turbine 3 outlet, Condenser inlet"

\[ T_{14} = \text{temperature}(\text{Water}, \ P=P_{14}, \ h=h_{14}) \]
\[ P_{14} = P_{1} \]
\[ s_{14} = \text{entropy}(\text{Water}, \ P=P_{14}, \ h=h_{14}) \]
\[ e_{14} = h_{14} - h[0] - T[0]*(s_{14} - s[0]) \]
\[ E_.{14} = \text{convert}(\text{kW,MW})*e_{14} * m_.3 \]

"State Point 15 - Closed FWH 1 outlet, Pump 2 inlet"

\[ T_{15} = \text{temperature}(\text{Water}, \ P=P_{15}, \ x=x_{15}) \]
\[ P_{15} = P_{12} \]
\[ h_{15} = \text{enthalpy}(\text{Water}, \ P=P_{15}, \ x=x_{15}) \]
\[ s_{15} = \text{entropy}(\text{Water}, \ P=P_{15}, \ x=x_{15}) \]
\[ x_{15} = 0 \]
\[ e_{15} = h_{15} - h[0] - T[0]*(s_{15} - s[0]) \]
\[ E_.{15} = \text{convert}(\text{kW,MW})*e_{15} * m_.5 \]

"State Point 16 - Pump 2 outlet, Mixer 1 inlet"

\[ T_{16} = \text{temperature}(\text{Water}, \ P=P_{16}, \ h=h_{16}) \]
\[ P_{16} = P_{2} \]
\[ s_{16} = \text{entropy}(\text{Water}, \ P=P_{16}, \ h=h_{16}) \]
\[ e_{16} = h_{16} - h[0] - T[0]*(s_{16} - s[0]) \]
\[ E_.{16} = \text{convert}(\text{kW,MW})*e_{16} * m_.5 \]

"State Point 17 - Closed FWH 2 outlet, Pump 3 inlet"

\[ T_{17} = \text{temperature}(\text{Water}, \ P=P_{17}, \ x=x_{17}) \]
\[ P_{17} = P_{10} \]
\[ h_{17} = \text{enthalpy}(\text{Water}, \ P=P_{17}, \ x=x_{17}) \]
\[ s_{17} = \text{entropy}(\text{Water}, \ P=P_{17}, \ x=x_{17}) \]
\[ x_{17} = 0 \]
\[ e_{17} = h_{17} - h[0] - T[0]*(s_{17} - s[0]) \]
\[ E_.{17} = \text{convert}(\text{kW,MW})*e_{17} * m_.4 \]

"State Point 18 - Pump 3 outlet, Mixer 2 inlet"

\[ T_{18} = \text{temperature}(\text{Water}, \ P=P_{18}, \ h=h_{18}) \]
\[ P_{18} = P_{2} \]
\[ s_{18} = \text{entropy}(\text{Water}, \ P=P_{18}, \ h=h_{18}) \]
\[ e_{18} = h_{18} - h[0] - T[0]*(s_{18} - s[0]) \]
\[ E_.{18} = \text{convert}(\text{kW,MW})*e_{18} * m_.4 \]

"Control Volumes: Mass and Energy Balances"

"Pump 1"
\[ \text{eta}_\text{Pump}_1 = 0.86 \]
\[ h_{2\_s} = \text{enthalpy}(\text{Water}, \ P=P_{2}, \ s=s[1]) \]
\[ \text{eta}_\text{Pump}_1 = (h_{2\_s} - h[1]) / (h[2] - h[1]) \]
\[ W_.\text{Pump}_1 = m_.3*(h[2] - h[1])*\text{convert}(\text{kW,MW}) \]

"Closed FWH 1"
\[ Q_.\text{FWH}_1 = m_.5*(h[12] - h[15])*\text{convert}(\text{kW,MW}) \]
\[ Q_.\text{FWH}_1 = m_.3*(h[3] - h[2])*\text{convert}(\text{kW,MW}) \]

"Mixer 1"

"Pump 2"
\[ \text{eta}_\text{Pump}_2 = 0.86 \]
h_16_s = enthalpy(Water, P=P[16], s=s[15])
eta_Pump_2 = (h_16_s - h[15]) / (h[16] - h[15])
W_.Pump_2 = m_.5*(h[16] - h[15])*convert(kW,MW)

"Closed FWH 2"
Q_.FWH_2 = m_.4*(h[10] - h[17])*convert(kW,MW)
Q_.FWH_2 = m_.2*(h[5] - h[4])*convert(kW,MW)

"Mixer 2"

"Pump 3"
eta_Pump_3 = 0.86
h_18_s = enthalpy(Water, P=P[18], s=s[17])
eta_Pump_3 = (h_18_s - h[17]) / (h[18] - h[17])
W_.Pump_3 = m_.4*(h[18] - h[17])*convert(kW,MW)

"Boiler & Reheater"
eto_boiler = 0.85
HHV_coal = 27.135 [MJ/kg] "Coal Heating Value"
eto_burner = 1 - 0.077
Q_.boiler = (HHV_coal* m_.c)*eta_boiler*eta_burner
Q_.effluent = (Q_.boiler/eta_boiler)*(1 - eta_boiler)

"C51.2 H43 O4.2 N0.9 S0.8 + (O2 + N2 + H2O) --> CO2 + H2O + SO2 + N2"
e_coal = 27680 [kJ/kg] "Exergetic Value of Coal"
T_air = 298.15 [K] "Temperature of incoming air"
P_air = 101.13 [kPa] "Pressure of incoming air"
h_air = enthalpy(Air, T=T_air) "Enthalpy of incoming air"
s_air = entropy(Air, T=T_air) "Entropy of incoming air"
e_air = h_air - enthalpy(Air, T=T[0], P=P[0]) -(T[0]*s_air-entropy(Air, T=T[0],P=P[0]))
AFR_theoretical = 11.36
AFR_theoretical = m_.air_th / m_.c
m_.air = 1.15 * m_.air_th "Air Flow Rate"
E_.air = (e_air*m_.air)*convert(kW,MW) "Exergy of Air"
E_.coal = e_coal*m_.c*convert(kW,MW) "Exergy of Coal"
c_coal = 55 [$/ton]*convert(1/ton, 1/kg) "Specific Cost of Coal"

"Turbine 1"
eto_Turb = 0.86 "Isentropic Efficiency"
h_10_s = enthalpy(Water, P=P[10], s=s[9])
eto_Turb = (h[9] - h[10]) / (h[9] - h_10_s) "Enthalpy Calculations"
W_.Turb_1 = m_.1*(h[9] - h[10])*convert(kW,MW) "Work output of Stage 1"

"Split 1"
m_.1 = m_.2 + m_.4
m_.2 = m_.1*(1-y)
m_.4 = m_.1*(y)

"Turbine 2"
h_12_s = enthalpy(Water, P=P[12], s=s[11])
"Split 2"
m_.2 = m_.3 + m_.5
m_.3 = m_.2*(1-z)
m_.5 = m_.2*(z)

"Turbine 3"
h_14_s = enthalpy(Water, P=P[14], s=s[13])
W_.Turb_3 = m_.3*(h[13] - h[14])*convert(kW,MW)

"Condenser"
Q_.Condenser = (m_.3*(h[14] - h[1]))*convert(kW,MW)
W_.Cond_fans = (7.4133[kW/MW]*Q_.Condenser + 2.9342[kW])*convert(kW,MW)

W_.out = 1000 [MW]
W_.out = W_.Turb - W_.Pump

"Total Turbine Work output"
"Total Pump Work input"

eta_th = W_.out / Q_.Boiler

"Cost Calculations"
i=0.04
i_eff = ((1+(i/m))^m)-1
r_n=0.04
r_n_coal = 0.039
UF = 0.90
n = 22
m = 12
k=(1+r_n)/(1+i_eff)

CELF = ((k*(1-k^m))/((1-k)^(m-1)))
CRF = (i_eff*(1+i_eff)^n)/((1+i_eff)^(n-1))
C_.coal = c_coal * m_.c * convert($/s,$/hr)
C_.air = 0 [$/hr]

DC = PEC_total*2.69
IC = PEC_total * 0.72
FCI = IC + DC + 0.56*PEC_Total
TCI = FCI + 0.7*PEC_Total
TCI_L = (TCI /(8776[hr]*UF*n))

Var = 53000 [$/yr] * convert($/yr, $/hr)
OM_L = (0.2*PEC_total)*CELF
CC_L = (4*PEC_total)*CRF

TRR_L = (OM_L + CC_L + Var)/(8776[hr]*UF) + C_.coal
Cost_elec = ((TRR_L+TCI_L) / W_.out)*convert($/MW-hr, cents/kW-hr)

PEC_Boiler = 0.367[$/(Btu/hr)^.77]*((Q_.Boiler*convert(MW,Btu/hr))^0.77)
PEC_Pump_1=-0.0988[$/(kg/s)^2]*m_.3^2+232.61[$/(kg/s)]*m_.3+45142[$]
PEC_Pump_2=-0.0988[$/(kg/s)^2]*m_.5^2+232.61[$/(kg/s)]*m_.5+45142[$]
PEC_Pump_3=-0.0988[$/(kg/s)^2]*m_.4^2+232.61[$/(kg/s)]*m_.4+45142[$]
PEC_Turb_total = 2* 1.7537e6[$/MW^.7] *(W_.Turb^0.7)"Turbine Purchased Equipment Cost Function"
PEC_Turb_1 = PEC_Turb_total*(W_.Turb_1 / W_.Turb)
PEC_Turb_2 = PEC_Turb_total*(W_.Turb_2 / W_.Turb)
PEC_Turb_3 = PEC_Turb_total*(W_.Turb_3 / W_.Turb)

U_Steel = .680 [kW/m^2-K]
a_cost_Steel=1.75
b=.13

PEC_FWH_1=92400[$]
"Determined in separate spreadsheet"

PEC_FWH_2=479000[$]
"Determined in separate spreadsheet"

PEC_condenser=28750000[$]
"Determined in separate spreadsheet"

PEC_Purification = 1.3E6 [$]
PEC_Environmental = 1300000 [$]
14"

PEC_total=PEC_Boiler+PEC_Pump_1+PEC_Pump_2+PEC_Pump_3+PEC_Turb_total+PEC_condenser +PEC_FWH_1+PEC_FWH_2 + PEC_Purification + PEC_Environmental
"Sum Total of Purchased Equipment Costs""Costs associated with each Stream - used to find the cost of Electricity"
C_.[1] = c[1] * E_.[1]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[5] = c[5] * E_.[5]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[9] = c[9] * E_.[9]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[12] = c[12] * E_.[12]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[14] = c[14] * E_.[14]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[16] = c[16] * E_.[16]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[17] = c[17] * E_.[17]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.[18] = c[18] * E_.[18]*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Pump_1 = c_work * W_.Pump_1*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Pump_2 = c_work * W_.Pump_2*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Pump_3 = c_work * W_.Pump_3*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Turb_1 = c_work * W_.Turb_1*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Turb_2 = c_work * W_.Turb_2*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Turb_3 = c_work * W_.Turb_3*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.Condenser = c_work * W_.Cond_fans*convert($/s, $/hr)*convert($/GJ,$/MJ)
C_.work_out = c_work * W_.out*convert($/s, $/hr)*convert($/GJ,$/MJ)

"Cost Accounting"
C_.[1] + C_.pump_1 + z_.Pump_1 = C_.[2]
C_.[15] + C_.pump_2 + z_.Pump_2 = C_.[16]
C_.[17] + C_.pump_3 + z_.Pump_3 = C_.[18]
C_.[2] + C_.[12]*(z) + z_.FWH_1 = C_.[3] + C_.[15]
C_.[3] + C_.[16] + z_.Mixer_1 = C_.[4]
C_.[9] + z_.Turb_1 = C_.[10] + C_.Turb_1  
C_.[11] + z_.Turb_2 = C_.[12] + C_.Turb_2  
C_.[14] + C_.air + z_.condenser + C_.condenser = C_.[1] + C_.air_out

c[9] = c[10]  
c[13] = c[14]  
c[10] = c[17]  
(C_.[9]-C_.[6]) / (E_.[9]-E_.[6]) = (C_.[11]-C_.[10]*(1-y)) / (E_.[11]-E_.[10]*(1-y))  
(C_.[9]-C_.[6]) / (E_.[9]-E_.[6]) = (C_.[13]-C_.[12]*(1-z)) / (E_.[13]-E_.[12]*(1-z))

"Exergoeconomic Parameters"

"Pump 1"
epsilon_pump_1 = (E_.[2] - E_.[1]) / (W_.pump_1)  
E_.D_Pump_1 = (E_.[1] + W_.Pump_1) - (E_.[2])  
y_D_Pump_1 = E_.D_Pump_1 / (E_.[1] + W_.Pump_1)  
c_prod_Pump_1 = (C_.[2]-C_.[1]) / (E_.[2]-E_.[1])*convert($/hr-MW,$/GJ)  
C_.D_Pump_1 = c_work * E_.D_Pump_1*convert(MW/GJ, 1/hr)  
z_.Pump_1 = ((CC_L+OM_L)*(PEC_Pump_1/PEC_total))/(8776[hr]*UF)  
r_Pump_1= (c_prod_Pump_1 - c_work) / c_work  
f_Pump_1 = z_.Pump_1 / (z_.Pump_1 + C_.D_Pump_1)

"Closed FW 1"
epsilon_FWH_1 = (E_.[3] - E_.[2]) / (E_.[12] - E_.[15])  
E_.D_FWH_1 = (E_.[2] + E_.[12]) - (E_.[3] + E_.[15])  
y_D_FWH_1 = E_.D_FWH_1 / (E_.[2] + E_.[12])  
c_fuel_FWH_1 = (C_.[12]-C_.[15]) / (E_.[12]-E_.[15])*convert($/hr-MW,$/GJ)  
c_prod_FWH_1 = (C_.[3]-C_.[2]) / (E_.[3]-E_.[2])*convert($/hr-MW,$/GJ)  
C_.D_FWH_1 = c_fuel_FWH_1 * E_.D_FWH_1*convert(MW/GJ, 1/hr)  
z_.FWH_1 = ((CC_L+OM_L)*(PEC_FWH_1/PEC_total))/(8776[hr]*UF)  
r_FWH_1= (c_prod_FWH_1 - c_fuel_FWH_1) / c_fuel_FWH_1  
f_FWH_1 = z_.FWH_1 / (z_.FWH_1 + C_.D_FWH_1)

"Pump 2"
epsilon_Pump_2 = (E_.[16] - E_.[15]) / (W_.Pump_2)  
E_.D_Pump_2 = (E_.[15] + W_.Pump_2) - (E_.[16])  
y_D_Pump_2 = E_.D_Pump_2 / (E_.[15] + W_.Pump_2)  
c_prod_Pump_2 = (C_.[16]-C_.[15]) / (E_.[16]-E_.[15])*convert($/hr-MW,$/GJ)  
C_.D_Pump_2 = c_work * E_.D_Pump_2*convert(MW/GJ, 1/hr)  
z_.Pump_2 = ((CC_L+OM_L)*(PEC_Pump_2/PEC_total))/(8776[hr]*UF)  
r_Pump_2= (c_prod_Pump_2 - c_work) / c_work  
f_Pump_2 = z_.Pump_2 / (z_.Pump_2 + C_.D_Pump_2)  
z_.Mixer_1 = 0 [$/hr]

"Closed FW 2"
epsilon_FWH_2 = (E_.[5] - E_.[4]) / (E_.[10] - E_.[17])  
E_.D_FWH_2 = (E_.[4] + E_.[10]) - (E_.[5] + E_.[17])  
y_D_FWH_2 = E_.D_FWH_2 / (E_.[4] + E_.[10])  
c_fuel_FWH_2 = (C_.[10]-C_.[17]) / (E_.[10]-E_.[17])*convert($/hr-MW,$/GJ)  
c_prod_FWH_2 = (C_.[5]-C_.[4]) / (E_.[5]-E_.[4])*convert($/hr-MW,$/GJ)  
C_.D_FWH_2 = c_fuel_FWH_2 * E_.D_FWH_2*convert(MW/GJ, 1/hr)  
z_.FWH_2 = ((CC_L+OM_L)*(PEC_FWH_2/PEC_total))/(8776[hr]*UF)
\[
\begin{align*}
\text{FWH}_2 &= (c_{\text{prod}_2} - c_{\text{fuel}_2}) / c_{\text{fuel}_2} \\
\text{FWH}_2 &= z_{\text{FWH}_2} / (z_{\text{FWH}_2} + C_{\text{D_FWH}_2})
\end{align*}
\]

"Pump 3"
\[
\begin{align*}
\epsilon_{\text{pump}_3} &= \frac{(E_{[18]} - E_{[17]})}{W_{\text{pump}_3}} \\
E_{\text{D_Pump}_3} &= E_{\text{D_FWH}_2} / (E_{[17]} + E_{[18]}) \\
c_{\text{prod}_2} &= (C_{[18]} - C_{[17]}) / (E_{[18]} - E_{[17]}) \times \text{conv}($/hr - MW/$/GJ) \\
C_{\text{D_Pump}_3} &= c_{\text{work}} \times E_{\text{D_Pump}_3} \times \text{conv}(MW/GJ, 1/hr) \\
z_{\text{Pump}_3} &= \frac{(CC_{L} + OM_{L}) \times (PEC_{\text{Pump}_3} / PEC_{\text{total}})}{(8776 \times \text{UF})} \\
r_{\text{Pump}_3} &= \frac{(c_{\text{prod}_3} - c_{\text{work}})}{c_{\text{work}}} \\
f_{\text{Pump}_3} &= \frac{z_{\text{Pump}_3}}{z_{\text{Pump}_3} + C_{\text{D_Pump}_3}}
\end{align*}
\]

"Mixer 2"
\[
\begin{align*}
\epsilon_{\text{mixer}_2} &= \frac{E_{[6]}}{E_{[5]} + E_{[18]}} \\
E_{\text{D_mixer}_2} &= (E_{[5]} + E_{[18]}) - (E_{[6]}) \\
y_{\text{D_mixer}_2} &= \frac{E_{\text{D_mixer}_2}}{E_{[5]} + E_{[18]}} \\
c_{\text{fuel}_2} &= \frac{(C_{[5]} + C_{[18]})}{(E_{[5]} + E_{[18]})} \times \text{conv}($/hr - MW/$/GJ) \\
c_{\text{prod}_2} &= \frac{C_{[6]}}{E_{[6]}} \times \text{conv}($/hr - MW/$/GJ) \\
C_{\text{D_mixer}_2} &= c_{\text{fuel}_2} \times E_{\text{D_mixer}_2} \times \text{conv}(MW/GJ, 1/hr) \\
z_{\text{mixer}_2} &= \frac{(CC_{L} + OM_{L}) \times (PEC_{\text{mixer}_2} / PEC_{\text{total}})}{(8776 \times \text{UF})} \\
r_{\text{mixer}_2} &= \frac{(c_{\text{prod}_2} - c_{\text{fuel}_2})}{c_{\text{fuel}_2}} \\
f_{\text{mixer}_2} &= \frac{z_{\text{mixer}_2}}{z_{\text{mixer}_2} + C_{\text{D_mixer}_2}}
\end{align*}
\]

"Boiler & Reheaters"
\[
\begin{align*}
\epsilon_{\text{Boiler}} &= \frac{(E_{[9]} - E_{[6]} + (E_{[11]} - E_{[10]}) \times (1 - y) + (E_{[13]} - E_{[12]}) \times (1 - z))}{E_{\text{air}} + E_{\text{coal}}} \\
E_{\text{D_Boiler}} &= (E_{[6]} + E_{[10]} \times (1 - y) + E_{[12]} \times (1 - z) + E_{\text{coal}} + E_{\text{air}}) - (E_{[9]} + E_{[11]} + E_{[13]}) \\
y_{\text{D_Boiler}} &= \frac{E_{\text{D_Boiler}}}{E_{\text{coal}} + E_{\text{air}} + E_{[6]} + E_{[10]} \times (1 - y) + E_{[12]} \times (1 - z)} \\
c_{\text{fuel}_2} &= \frac{(C_{\text{coal}} + C_{\text{air}})}{(E_{\text{coal}} + E_{\text{air}})} \times \text{conv}($/hr-MW/$/GJ) \\
c_{\text{prod}_2} &= \frac{(C_{[9]} - C_{[11]} - C_{\text{coal}} \times (1 - y) + (C_{[13]} - C_{[12]} \times (1 - z)))}{(E_{[9]} - E_{[6]} + E_{[10]} \times (1 - y) + E_{[12]} \times (1 - z))} \times \text{conv}($/hr-MW/$/GJ) \\
C_{\text{D_Boiler}} &= c_{\text{fuel}_2} \times E_{\text{D_Boiler}} \times \text{conv}(MW/GJ, 1/hr) \\
z_{\text{Boiler}} &= \frac{(CC_{L} + OM_{L}) \times (PEC_{\text{Boiler}} / PEC_{\text{total}})}{(8776 \times \text{UF})} \\
r_{\text{Boiler}} &= \frac{(c_{\text{prod}_2} - c_{\text{fuel}_2})}{c_{\text{fuel}_2}} \\
f_{\text{Boiler}} &= \frac{z_{\text{Boiler}}}{z_{\text{Boiler}} + C_{\text{D_Boiler}}}
\end{align*}
\]

"Turbine 1"
\[
\begin{align*}
\epsilon_{\text{Turb}_1} &= \frac{W_{\text{Turb}_1}}{E_{[9]} - E_{[10]}} \\
E_{\text{D_Turb}_1} &= (E_{[9]} - E_{[10]} + W_{\text{Turb}_1}) \\
y_{\text{Turb}_1} &= \frac{E_{\text{D_Turb}_1}}{E_{[9]}} \\
c_{\text{fuel}_2} &= \frac{(C_{\text{coal}} - C_{\text{air}})}{(E_{\text{coal}} - E_{\text{air}})} \times \text{conv}($/hr-MW/$/GJ) \\
C_{\text{D_Turb}_1} &= c_{\text{fuel}_2} \times E_{\text{D_Turb}_1} \times \text{conv}(MW/GJ, 1/hr) \\
z_{\text{Turb}_1} &= \frac{(CC_{L} + OM_{L}) \times (PEC_{\text{Turb}_1} / PEC_{\text{total}})}{(8776 \times \text{UF})} \\
r_{\text{Turb}_1} &= \frac{(c_{\text{work}} - c_{\text{fuel}_2})}{c_{\text{fuel}_2}} \\
f_{\text{Turb}_1} &= \frac{z_{\text{Turb}_1}}{z_{\text{Turb}_1} + C_{\text{D_Turb}_1}}
\end{align*}
\]

"Turbine 2"
\[
\begin{align*}
\epsilon_{\text{Turb}_2} &= \frac{W_{\text{Turb}_2}}{E_{[11]} - E_{[12]}} \\
E_{\text{D_Turb}_2} &= (E_{[11]} - E_{[12]} + W_{\text{Turb}_2}) \\
y_{\text{Turb}_2} &= \frac{E_{\text{D_Turb}_2}}{E_{[11]}} \\
c_{\text{fuel}_2} &= \frac{(C_{\text{coal}} - C_{\text{air}})}{(E_{\text{coal}} - E_{\text{air}})} \times \text{conv}($/hr-MW/$/GJ) \\
C_{\text{D_Turb}_2} &= c_{\text{fuel}_2} \times E_{\text{D_Turb}_2} \times \text{conv}(MW/GJ, 1/hr) \\
z_{\text{Turb}_2} &= \frac{(CC_{L} + OM_{L}) \times (PEC_{\text{Turb}_2} / PEC_{\text{total}})}{(8776 \times \text{UF})} \\
r_{\text{Turb}_2} &= \frac{(c_{\text{work}} - c_{\text{fuel}_2})}{c_{\text{fuel}_2}} \\
f_{\text{Turb}_2} &= \frac{z_{\text{Turb}_2}}{z_{\text{Turb}_2} + C_{\text{D_Turb}_2}}
\end{align*}
\]
“Turbine 3”

\[ \epsilon_{\text{turb}_3} = \frac{W_{\text{turb}_3}}{(E_{\text{[13]}} - E_{\text{[14]}})} \]
\[ E_{\text{D}_{\text{Turb}_3}} = (E_{\text{[13]}}) - (E_{\text{[14]}} + W_{\text{Turb}_3}) \]
\[ y_{\text{D}_{\text{Turb}_3}} = E_{\text{D}_{\text{Turb}_3}} / E_{\text{[13]}} \]
\[ c_{\text{fuel}_{\text{Turb}_3}} = \frac{(C_{\text{[13]}} - C_{\text{[14]}})}{(E_{\text{[13]}} - E_{\text{[14]}})} \times \text{convert}(\$/\text{hr-MW}, \$/\text{GJ}) \]
\[ C_{\text{D}_{\text{Turb}_3}} = c_{\text{fuel}_{\text{Turb}_3}} \times E_{\text{D}_{\text{Turb}_3}} \times \text{convert}(\text{MW/GJ}, 1/\text{hr}) \]
\[ z_{\text{Turb}_3} = (CC_{\text{L}} + OM_{\text{L}}) \times (PEC_{\text{Turb}_3} / PEC_{\text{total}}) / (8776[\text{hr}] \times UF) \]
\[ r_{\text{Turb}_3} = \frac{c_{\text{work}} - c_{\text{fuel}_{\text{Turb}_3}}}{c_{\text{fuel}_{\text{Turb}_3}}} \]
\[ f_{\text{Turb}_3} = \frac{z_{\text{Turb}_3}}{z_{\text{Turb}_3} + C_{\text{D}_{\text{Turb}_3}}} \]

“Condenser”

\[ c_{\text{fuel}_{\text{condenser}}} = \frac{(C_{\text{[14]}} - C_{\text{[1]}})}{(E_{\text{[14]}} - E_{\text{[1]}})} \times \text{convert}(\$/\text{hr-MW}, \$/\text{GJ}) \]
\[ z_{\text{condenser}} = (CC_{\text{L}} + OM_{\text{L}}) \times (PEC_{\text{condenser}} / PEC_{\text{total}}) / (7446[\text{hr}]) \]

"Overall System Exergetic Performance"

\[ \epsilon_{\text{system}} = \frac{W_{\text{out}}}{(E_{\text{coal}} + E_{\text{air}})} \]
\[ z_{\text{system}} = (CC_{\text{L}} + OM_{\text{L}}) / (7446[\text{hr}]) \]

"Exergetic Efficiency of System"

"Overall System Cash flow"
15.2 Appendix 2: EES Code of Some Rejected Thermodynamic Designs

"3 Reheats & 3 Regenerations"

"State Point 0 - Dead State"
\[ T[0] = \text{converttemp}(C,K,25) \]
\[ P[0] = 10 \text{ [kPa]} \]
\[ h[0] = \text{enthalpy}(\text{Water}, T=T[0], P=P[0]) \]
\[ s[0] = \text{entropy}(\text{Water}, T=T[0], P=P[0]) \]

"State Point 1 - Condenser Outlet, Pump 1 Inlet"
\[ T[1] = \text{temperature}(\text{Water}, P=P[1], x=x[1]) \]
\[ P[1] = 10 \text{ [kPa]} \]
\[ h[1] = \text{enthalpy}(\text{Water}, P=P[1], x=x[1]) \]
\[ s[1] = \text{entropy}(\text{Water}, P=P[1], x=x[1]) \]
\[ x[1] = 0 \]
\[ e[1] = h[1] - h[0] - T[0]*(s[1] - s[0]) \]
\[ E_.[1] = \text{convert}(\text{kW,MW})*e[1] * \text{m}_.4 \]

"State Point 2 - Pump 1 Outlet, Open FWH 1 inlet"
\[ T[2] = \text{temperature}(\text{Water}, P=P[2], h=h[2]) \]
\[ P[2] = 500 \text{ [kPa]} \]
\[ s[2] = \text{entropy}(\text{Water}, P=P[2], h=h[2]) \]
\[ E_.[2] = \text{convert}(\text{kW,MW})*e[2] * \text{m}_.4 \]

"State Point 3 - Open FWH 1 Outlet, Pump 2"
\[ T[3] = \text{temperature}(\text{Water}, P=P[3], h=h[3]) \]
\[ s[3] = \text{entropy}(\text{Water}, P=P[3], h=h[3]) \]
\[ E_.[3] = \text{convert}(\text{kW,MW})*e[3] * \text{m}_.3 \]

"State Point 4 - Pump 2 outlet, Open FWH 2 inlet"
\[ T[4] = \text{temperature}(\text{Water}, P=P[4], h=h[4]) \]
\[ P[4] = 2000 \text{ [kPa]} \]
\[ s[4] = \text{entropy}(\text{Water}, P=P[4], h=h[4]) \]
\[ E_.[4] = \text{convert}(\text{kW,MW})*e[4] * \text{m}_.3 \]

"State Point 5 - Open FWH 2 outlet, Pump 3 inlet"
\[ T[5] = \text{temperature}(\text{Water}, P=P[5], h=h[5]) \]
\[ s[5] = \text{entropy}(\text{Water}, P=P[5], h=h[5]) \]
\[ E_.[5] = \text{convert}(\text{kW,MW})*e[5] * \text{m}_.2 \]

"State Point 6 - Pump 3 outlet, Closed FWH 3 inlet"
\[ T[6] = \text{temperature}(\text{Water}, P=P[6], h=h[6]) \]
\[ P[6] = 15000 \text{ [kPa]} \]
\[ s[6] = \text{entropy}(\text{Water}, P=P[6], h=h[6]) \]
\[ E_.[6] = \text{convert}(\text{kW,MW})*e[6] * \text{m}_.2 \]

"State Point 7 - Closed FWH 3 outlet, mixer inlet"
\[ T[7] = \text{temperature}(\text{Water}, P=P[7], h=h[7]) \]
s[7] = entropy(Water, P=P[7], h=h[7])
E_.[7] = convert(kW,MW)*e[7] * m_.2

"State Point 8 - Mixer outlet, Boiler inlet"
T[8] = temperature(Water, P=P[8], h=h[8])
s[8] = entropy(Water, P=P[8], h=h[8])
e[8] = h[8] - h[0] - T[0]*s[8] - s[0])
E_.[8] = convert(kW,MW)*e[8] * m_.1

"State Point 9 - Saturated Liquid within Boiler"
T[9] = temperature(Water, P=P[9], h=h[9])
h[9] = enthalpy(Water, P=P[9], x=x[9])
s[9] = entropy(Water, P=P[9], h=h[9])
x[9] = 0
e[9] = h[9] - h[0] - T[0]*s[9] - s[0])
E_.[9] = convert(kW,MW)*e[9] * m_.1

"State Point 10 - Saturated Steam within Boiler"
T[10] = temperature(Water, P=P[10], h=h[10])
h[10] = enthalpy(Water, P=P[10], x=x[10])
s[10] = entropy(Water, P=P[10], h=h[10])
x[10] = 1
e[10] = h[10] - h[0] - T[0]*s[10] - s[0])
E_.[10] = convert(kW,MW)*e[10] * m_.1

"State Point 11 - Boiler outlet, Turbine 1 inlet"

"State Point 12 - Turbine 1 outlet, Reheat 1 inlet, Closed FWH 3 inlet"
T[12] = temperature(Water, P=P[12], h=h[12])
P[12] = 6000 [kPa]
s[12] = entropy(Water, P=P[12], h=h[12])
e[12] = h[12] - h[0] - T[0]*s[12] - s[0])
E_.[12] = convert(kW,MW)*e[12] * m_.1

"State Point 13 - Reheat 1 outlet, Turbine 2 inlet"
T[13] = converttemp(C,K,600)
h[13] = enthalpy(Water, T=T[13], P=P[13])
s[13] = entropy(Water, T=T[13], P=P[13])

"State Point 14 - Turbine 2 outlet, Reheat 2 inlet, Open FWH 2 inlet"
T[14] = temperature(Water, P=P[14], h=h[14])
s[14] = entropy(Water, P=P[14], h=h[14])
\( e_{14} = h_{14} - h[0] - T[0] \cdot (s[14] - s[0]) \)
\( E_{e_{14}} = \text{convert(kW,MW)} \cdot e_{14} \cdot m_{\cdot 2} \)

"State Point 15 - Reheat 2 outlet, Turbine 3 inlet"
\( T[15] = \text{converttemp(C,K,600)} \)
\( h[15] = \text{enthalpy(Water, T=T[15], P=P[15])} \)
\( s[15] = \text{entropy(Water, T=T[15], P=P[15])} \)
\( e[15] = h[15] - h[0] - T[0] \cdot (s[15] - s[0]) \)
\( E_{e_{15}} = \text{convert(kW,MW)} \cdot e[15] \cdot m_{\cdot 3} \)

"State Point 16 - Turbine 3 outlet, Reheat 3 inlet, Open FWH 1 inlet"
\( T[16] = \text{temperature(Water, P=P[16], h=h[16])} \)
\( P[16] = P[2] \)
\( s[16] = \text{entropy(Water, P=P[16], h=h[16])} \)
\( e[16] = h[16] - h[0] - T[0] \cdot (s[16] - s[0]) \)
\( E_{e_{16}} = \text{convert(kW,MW)} \cdot e[16] \cdot m_{\cdot 3} \)

"State Point 17 - Reheat 3 outlet, Turbine 4 inlet"
\( T[17] = \text{converttemp(C,K,600)} \)
\( h[17] = \text{enthalpy(Water, T=T[17], P=P[17])} \)
\( s[17] = \text{entropy(Water, T=T[17], P=P[17])} \)
\( e[17] = h[17] - h[0] - T[0] \cdot (s[17] - s[0]) \)
\( E_{e_{17}} = \text{convert(kW,MW)} \cdot e[17] \cdot m_{\cdot 4} \)

"State Point 18 - Turbine 4 outlet, Condenser inlet"
\( T[18] = \text{temperature(Water, P=P[18], h=h[18])} \)
\( P[18] = P[1] \)
\( s[18] = \text{entropy(Water, P=P[18], h=h[18])} \)
\( e[18] = h[18] - h[0] - T[0] \cdot (s[18] - s[0]) \)
\( E_{e_{18}} = \text{convert(kW,MW)} \cdot e[18] \cdot m_{\cdot 4} \)

"State Point 19 - Closed FWH 3 outlet, Pump 4 inlet"
\( T[19] = \text{temperature(Water, P=P[19], x=x[19])} \)
\( P[19] = P[12] \)
\( h[19] = \text{enthalpy(Water, P=P[19], x=x[19])} \)
\( s[19] = \text{entropy(Water, P=P[19], x=x[19])} \)
\( x[19] = 0 \)
\( e[19] = h[19] - h[0] - T[0] \cdot (s[19] - s[0]) \)
\( E_{e_{19}} = \text{convert(kW,MW)} \cdot e[19] \cdot m_{\cdot 5} \)

"State Point 20 - Pump 4 outlet, Mixer inlet"
\( T[20] = \text{temperature(Water, P=P[20], h=h[20])} \)
\( s[20] = \text{entropy(Water, P=P[20], h=h[20])} \)
\( e[20] = h[20] - h[0] - T[0] \cdot (s[20] - s[0]) \)
\( E_{e_{20}} = \text{convert(kW,MW)} \cdot e[20] \cdot m_{\cdot 5} \)

"Control Volumes"

"Pump 1"
\( \eta_{\text{Pump}_1} = 0.86 \)
\( h_{\cdot 2\_s} = \text{enthalpy(Water, P=P[2], s=s[1])} \)
\( \eta_{\text{Pump}_1} = (h_{\cdot 2\_s} - h[1]) / (h[2] - h[1]) \)
\( W_{\cdot \text{Pump}_1} = m_{\cdot 4} \cdot (h[2] - h[1]) \cdot \text{convert(kW,MW)} \)
"Open FWH 1"
m_.3*h[3] = m_.7*h[16] + m_.4*h[2]

"Pump 2"
eta_Pump_2 = 0.86
h_4_s = enthalpy(Water, P=P[4], s=s[3])
eta_Pump_2 = (h_4_s - h[3]) / (h[4] - h[3])
W__Pump_2 = m_.3*(h[4] - h[3])*convert(kW,MW)

"Open FWH 2"

"Pump 3"
eta_Pump_3 = 0.86
h_6_s = enthalpy(Water, P=P[6], s=s[5])
eta_Pump_3 = (h_6_s - h[5]) / (h[6] - h[5])
W__Pump_3 = m_.2*(h[6] - h[5])*convert(kW,MW)

"Closed FWH 3"
Q__FWH_3 = m_.5*(h[12] - h[19])*convert(kW,MW)
Q__FWH_3 = m_.2*(h[7] - h[6])*convert(kW,MW)

"Mixer"

"Pump 4"
eta_Pump_4 = 0.86
h_20_s = enthalpy(Water, P=P[20], s=s[19])
eta_Pump_4 = (h_20_s - h[19]) / (h[20] - h[19])
W__Pump_4 = m_.5*(h[20] - h[19])*convert(kW,MW)

"Boiler & Reheater"
eta_boiler = 0.85

"Typical Boiler Efficiency"
HHV_coal = 27.135 [MJ/kg]
Q__boiler = (HHV_coal* m_.c)*eta_boiler
Q__effluent = (Q__boiler/eta_boiler)*(1 - eta_boiler)

"C51.2 H43 O4.2 N0.9 S0.8 + (O2 + N2 + H2O) --> CO2 + H2O + SO2 + N2"
e_Coal = 27680 [kJ/kg]
T_air = 298.15 [K]
P_air = 101.13 [kPa]
h_air = enthalpy(Air, T=T_air)
s_air = entropy(Air, T=T_air, P=P_air)
e_air = h_air - enthalpy(Air, T=T[0] - T[0]*s_air-entropy(Air, T=T[0],P=P[0]))
AFR_theoretical = 11.36
AFR_theoretical = m_.air_th / m_.c
m_.air = 1.15 * m_.air_th
E__air = (e_air*m_.air)*convert(kW,MW)
E__coal = e_coal*m_.c*convert(kW,MW)
c_coal = 55 [$/ton]*convert(1/ton, 1/kg)

"Turbine 1"
eta_Turb = 0.86
h_12_s = enthalpy(Water, P=P[12], s=s[11])
W_.Turb_1 = m_.1*(h[11] - h[12])*convert(kW,MW)

"Split 1"

m_.1 = m_.2 + m_.5
m_.2 = m_.1*(1-x)

m_.5 = m_.1*(x)

"Turbine 2"

h_14_s = enthalpy(Water, P=P[14], s=s[13])
W_.Turb_2 = m_.2*(h[13] - h[14])*convert(kW,MW)

"Split 2"

m_.2 = m_.3 + m_.6
m_.3 = m_.2*(1-y)

m_.6 = m_.2*(y)

"Turbine 3"

h_16_s = enthalpy(Water, P=P[16], s=s[15])
eta_Turb = (h[15] - h[16]) / (h[15] - h_16_s)
W_.Turb_3 = m_.3*(h[15] - h[16])*convert(kW,MW)

"Split 3"

m_.3 = m_.4 + m_.7
m_.4 = m_.3*(1-z)

m_.7 = m_.3*(z)

"Turbine 4"

h_18_s = enthalpy(Water, P=P[18], s=s[17])
eta_Turb = (h[17] - h[18]) / (h[17] - h_18_s)
W_.Turb_4 = m_.4*(h[17] - h[18])*convert(kW,MW)

"Condenser"

Q_.Condenser = (m_.4*(h[18] - h[1]))*convert(kW,MW)
W_.cond_fans = (7.4133[kW/MW]*Q_.condenser+2.9342[kW])*convert(kW,MW)
W_.out = 1000 [MW]
W_.out = W_.Turb - W_.Pump-W_.cond_fans
W_.Turb = W_.Turb_1 + W_.Turb_2 + W_.Turb_3 + W_.Turb_4
W_.Pump = W_.Pump_1 + W_.Pump_2 + W_.Pump_3 + W_.Pump_4
teta_th = W_.out / Q_.Boiler

"Cost Calculations"

i=0.05
i_eff = ((1+(i/m))^m)-1
r_n=0.04
r_n_coal = 0.039
UF = 0.85
n = 20
m = 12
k= (1+r_n)/(1+i_eff)
CELF = ((k*(1-k^n))/(1-k)) CRF
CRF = (i_eff*(1+i_eff)^n)/((1+i_eff)^n-1)
C_.coal = c_coal * m_.c * convert($/s,$/hr)
C_.air = 0 [$/hr]
OM_L = (0.2*PEC_total)*CELF
CC_L = (4*PEC_total)*CRF
TRR_L = (OM_L + CC_L)/(8776[hr]*UF) + C_.coal
Cost_elec = (TRR_L / W_.out)*convert($/MW-hr, $/kW-hr)

PEC_Boiler = 0.367[$/(Btu/hr)^.77]*((Q_.Boiler*convert(MW,Btu/hr))^0.77)
PEC_Pump_1 = -0.0988[$/(kg/s)^2]*m_.1^2+232.61[$/(kg/s)]*(m_.1)+45142[$]
Pump_2 = -0.0988[$/(kg/s)^2]*(m_.2^2)+232.61[$/(kg/s)]*(m_.2)+45142[$]
Pump_3 = -0.0988[$/(kg/s)^2]*(m_.3^2)+232.61[$/(kg/s)]*(m_.3)+45142[$]
Pump_4 = -0.0988[$/(kg/s)^2]*(m_.4^2)+232.61[$/(kg/s)]*(m_.4)+45142[$]

PEC_Turb_total = 2.1874e6[$/MW^0.7]*W_.Turb^0.7
PEC_Turb_1 = PEC_Turb_total*(W_.Turb_1 / W_.Turb)
Pump_2 = PEC_Turb_total*(W_.Turb_2 / W_.Turb)
Pump_3 = PEC_Turb_total*(W_.Turb_3 / W_.Turb)
Pump_4 = PEC_Turb_total*(W_.Turb_4 / W_.Turb)

Q_.FWH_3*convert(MW,kW) = U_Steel*A_FWH_3*LMTD_FWH_3
C_B_FWH_3=exp(11.0545-.9228*ln(A_FWH_3*1[m^2])+.09861*(ln(A_FWH_3*1[m^-2])^2))
F_M_FWH_3=(a_cost_Steel+(A_FWH_3/100[m^2])^b)
F_P_FWH_3=.9803+.018*(P[2]/100[kPa])+.0017[kPa^-2]*(P[2])^2
PEC_FWH_3=0.79[$]*C_B_FWH_3*F_M_FWH_3*F_P_FWH_3
U_steel = 0.680[kW/m^2-K]
a_cost_steel = 1.75
b=0.13
PEC_condenser=68.761[$/MW]*Q_.condenser+137592[$]
PEC_total=PEC_Boiler+PEC_Pump_1+PEC_Pump_2+PEC_Pump_3+PEC_Pump_4+PEC_Turb_total+
PEC_condenser+PEC_FWH_3
DC = PEC_total*2.49
IC = DC * 0.2645
TCI = IC + DC + ((OM_L/ 12) + (0.1*OM_L*UF)/12 + C_.coal*142.8[hr] + (0.02*(IC+DC)) +
(C_.coal*1241[hr])*1.25

"Cost Streams"
C_[1] = c[1] * E_[1]*convert($/s, $/hr)
C_[2] = c[2] * E_[2]*convert($/s, $/hr)
C_[3] = c[3] * E_[3]*convert($/s, $/hr)
C_[5] = c[5] * E_[5]*convert($/s, $/hr)
C_[6] = c[6] * E_[6]*convert($/s, $/hr)
C_[7] = c[7] * E_[7]*convert($/s, $/hr)
C_[8] = c[8] * E_[8]*convert($/s, $/hr)
C_[12] = c[12] * E_[12]*convert($/s, $/hr)
C_[14] = c[14] * E_[14]*convert($/s, $/hr)
C_[16] = c[16] * E_[16]*convert($/s, $/hr)
C_[17] = c[17] * E_[17]*convert($/s, $/hr)
C_[18] = c[18] * E_[18]*convert($/s, $/hr)
C_[19] = c[19] * E_[19]*convert($/s, $/hr)

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\[
C_\text{[20]} = c[20] \times E_\text{[20]} \times \text{convert(s/hr, $/s)}
\]
\[
C_\text{Pump}_1 = c_\text{work} \times W_\text{Pump}_1 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Pump}_2 = c_\text{work} \times W_\text{Pump}_2 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Pump}_3 = c_\text{work} \times W_\text{Pump}_3 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Pump}_4 = c_\text{work} \times W_\text{Pump}_4 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Turb}_1 = c_\text{work} \times W_\text{Turb}_1 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Turb}_2 = c_\text{work} \times W_\text{Turb}_2 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Turb}_3 = c_\text{work} \times W_\text{Turb}_3 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{Turb}_4 = c_\text{work} \times W_\text{Turb}_4 \times \text{convert($/s, $/hr)}
\]
\[
C_\text{condenser} = c_\text{work} \times W_\text{cond_fans} \times \text{convert($/s, $/hr)}
\]

"Cost Accounting"
\[
(C_\text{[8]} + C_\text{[12]}(1-x) + C_\text{[14]}(1-y) + C_\text{[16]}(1-z) + C_\text{coal} + C_\text{air} + z_\text{Boiler} = C_\text{[11]} + C_\text{[13]} + C_\text{[15]} + C_\text{[17]}
\]
\[
C_\text{[1]} + C_\text{[pump}_1 + z_\text{Pump}_1 = C_\text{[2]}
\]
\[
C_\text{[3]} + C_\text{[pump}_2 + z_\text{Pump}_2 = C_\text{[4]}
\]
\[
C_\text{[5]} + C_\text{[pump}_3 + z_\text{Pump}_3 = C_\text{[6]}
\]
\[
C_\text{[19]} + C_\text{[pump}_4 + z_\text{Pump}_4 = C_\text{[20]}
\]
\[
C_\text{[2]} + C_\text{[16]}(z) + z_\text{FWH}_1 = C_\text{[3]}
\]
\[
C_\text{[4]} + C_\text{[14]}(y) + z_\text{FWH}_2 = C_\text{[5]}
\]
\[
C_\text{[6]} + C_\text{[12]}(x) + z_\text{FWH}_3 = C_\text{[7]} + C_\text{[19]}
\]
\[
C_\text{[7]} + C_\text{[20]} + z_\text{Mixer} = C_\text{[8]}
\]
\[
C_\text{[11]} + z_\text{Turb}_1 = C_\text{[12]} + C_\text{Turb}_1
\]
\[
C_\text{[13]} + z_\text{Turb}_2 = C_\text{[14]} + C_\text{Turb}_2
\]
\[
C_\text{[15]} + z_\text{Turb}_3 = C_\text{[16]} + C_\text{Turb}_3
\]
\[
C_\text{[17]} + z_\text{Turb}_4 = C_\text{[18]} + C_\text{Turb}_4
\]
\[
C_\text{[18]} + C_\text{[coolant}_in + z_\text{condenser} + C_\text{Condenser} = C_\text{[1]} + C_\text{coolant_out}
\]
\[
C_\text{coolant}_in = 0[$/hr]
\]
\[
\]
\[
c[13] = c[14]
\]
\[
c[15] = c[16]
\]
\[
c[17] = c[18]
\]
\[
\]

\[(C_\text{[11]}-C_\text{[8]})/(E_\text{[11]}-E_\text{[8]}) = (C_\text{[13]}-C_\text{[12]}(1-x))/(E_\text{[13]}-E_\text{[12]}(1-x))
\]
\[(C_\text{[11]}-C_\text{[8]})/(E_\text{[11]}-E_\text{[8]}) = (C_\text{[15]}-C_\text{[14]}(1-y))/(E_\text{[15]}-E_\text{[14]}(1-y))
\]
\[(C_\text{[13]}-C_\text{[12]})/(E_\text{[13]}-E_\text{[12]}) = (C_\text{[17]}-C_\text{[16]}(1-z))/(E_\text{[17]}-E_\text{[16]}(1-z))\]

"Exergoeconomic Parameters"

"Pump 1"
\[
\epsilon_\text{pump}_1 = (E_\text{[2]} - E_\text{[1]}) / (W_\text{pump}_1)
\]
\[
E_\text{D_Pump}_1 = (E_\text{[1]} + W_\text{Pump}_1) / (E_\text{[1] + W_\text{Pump}_1})
\]
\[
c_\text{fuel_Pump}_1 = (C_\text{Pump}_1) / (W_\text{Pump}_1) \times \text{convert($/hr-MW, $/GJ)}
\]
\[
c_\text{prod_Pump}_1 = (C_\text{[2]}-C_\text{[1]}) / (E_\text{[2]}-E_\text{[1]}) \times \text{convert($/hr-MW, $/GJ)}
\]
\[
C_\text{D_Pump}_1 = c_\text{fuel_Pump}_1 \times E_\text{D_Pump}_1 \times \text{convert(MW/GJ, 1/hr)}
\]
\[
z_\text{Pump}_1 = ((CC_L+OM_L) \times (PEC_\text{Pump}_1/PEC_\text{total}))/\text{(8776[hr]*UF)}
\]
\[
f_\text{Pump}_1 = (c_\text{prod_Pump}_1 - c_\text{fuel_Pump}_1) / c_\text{fuel_Pump}_1
\]
\[
f_\text{Pump}_1 = z_\text{Pump}_1 / (z_\text{Pump}_1 + C_\text{D_Pump}_1)
\]

"Open FWH 1"
\[
\text{z_open_FWH1} = 60[$/hr]
\]

"Pump 2"
\[
\epsilon_\text{pump}_2 = (E_\text{[4]} - E_\text{[3]}) / (W_\text{pump}_2)
\]
E_.D_Pump_2 = (E_[3] + W_.Pump_2) - (E_[4])
y_D_Pump_2 = E_.D_Pump_2 / (E_[3] + W_.Pump_2)
\[
c_.fuel_Pump_2 = (C_.Pump_2) / (W_.Pump_2) \times \text{convert}($/hr-MW,$/GJ)
\]
c_.prod_Pump_2 = (C_[4]-C_[3]) / (E_[4]-E_[3]) \times \text{convert}($/hr-MW,$/GJ)
\[
C_.D_Pump_2 = c_.fuel_Pump_2 \times E_.D_Pump_2 \times \text{convert}(MW/GJ, 1/hr)
\]
z_.Pump_2 =((CC_L+OM_L) \times (PEC_Pump_2/PEC\text{total}))/8776[hr] \times \text{UF}
\[
\{r\}_Pump_2 = (c_.prod_Pump_2 - c_.fuel_Pump_2) / c_.fuel_Pump_2
\]
f_.Pump_2 = z_.Pump_2 / (z_.Pump_2 + C_.D_Pump_2)

"Open FWH 2"
"
!"z_.open_FWH2 = 60[$/hr]

"Pump 3"
\[
epsilon_{\text{pump}}_3 = (E_[6] - E_[5]) / (W_.pump_3)
\]
E_.D_Pump_3 = (E_[5] + W_.Pump_3) - (E_[6])
\[
c_.fuel_Pump_3 = (C_.Pump_3) / (W_.Pump_3) \times \text{convert}($/hr-MW,$/GJ)
\]
c_.prod_Pump_3 = (C_[6]-C_[5]) / (E_[6]-E_[5]) \times \text{convert}($/hr-MW,$/GJ)
\[
C_.D_Pump_3 = c_.fuel_Pump_3 \times E_.D_Pump_3 \times \text{convert}(MW/GJ, 1/hr)
\]
z_.Pump_3 =((CC_L+OM_L) \times (PEC_Pump_3/PEC\text{total}))/8776[hr] \times \text{UF}
\[
\{r\}_Pump_3 = (c_.prod_Pump_3 - c_.fuel_Pump_3) / c_.fuel_Pump_3
\]
f_.Pump_3 = z_.Pump_3 / (z_.Pump_3 + C_.D_Pump_3)

"Closed FWH 3"
\[
epsilon_{\text{FWH}}_3 = (E_[6] - E_[5]) / (E_[12] - E_[19])
\]
E_.D_FWH_3 = (E_[6] + E_[12]) - (E_[19])
y_D_FWH_3 = E_.D_FWH_3 / (E_[6] + E_[12])
\[
c_.fuel_FWH_3 = (C_[12]-C_[19]) / (E_[12] - E_[19]) \times \text{convert}($/hr-MW,$/GJ)
\]
c_.prod_FWH_3 = (C_[7]-C_[6]) / (E_[7]-E_[6]) \times \text{convert}($/hr-MW,$/GJ)
\[
C_.D_FWH_3 = c_.fuel_FWH_3 \times E_.D_FWH_3 \times \text{convert}(MW/GJ, 1/hr)
\]
z_.FWH_3 =((CC_L+OM_L) \times (PEC_FWH_3/PEC\text{total}))/8776[hr] \times \text{UF}
\[
\{r\}_FWH_3 = (c_.prod_FWH_3 - c_.fuel_FWH_3) / c_.fuel_FWH_3
\]
f_.FWH_3 = z_.FWH_3 / (z_.FWH_3 + C_.D_FWH_3)

"Mixer"
"
!"z_.mixer = 50[$/hr]

"Pump 4"
\[
epsilon_{\text{pump}}_4 = (E_[19] - E_[18]) / (W_.pump_4)
\]
E_.D_Pump_4 = (E_[19] + W_.Pump_4) - (E_[20])
\[
c_.fuel_Pump_4 = (C_.Pump_4) / (W_.Pump_4) \times \text{convert}($/hr-MW,$/GJ)
\]
c_.prod_Pump_4 = (C_[20]-C_[19]) / (E_[20]-E_[19]) \times \text{convert}($/hr-MW,$/GJ)
\[
C_.D_Pump_4 = c_.fuel_Pump_4 \times E_.D_Pump_4 \times \text{convert}(MW/GJ, 1/hr)
\]
z_.Pump_4 =((CC_L+OM_L) \times (PEC_Pump_4/PEC\text{total}))/8776[hr] \times \text{UF}
\[
\{r\}_Pump_4 = (c_.prod_Pump_4 - c_.fuel_Pump_4) / c_.fuel_Pump_4
\]
f_.Pump_4 = z_.Pump_4 / (z_.Pump_4 + C_.D_Pump_4)

"Boiler & Reheaters"
\[
\]
\[
c_.fuel_Boiler = (C_.coal + C_.air) / (E_.coal + E_.air) \times \text{convert}($/hr-MW,$/GJ)
\[
c_{ prod\_Boiler} = ((C_[11]-C_[8])+(C_[14]-C_[13])) / ((E_[11]-E_[8])+(E_[14]-E_[13])) \times \text{convert}(\$/hr-MW,\$/GJ) \\
C_{d\_Boiler} = c_{fuel\_Boiler} \times E_{d\_Boiler} \times \text{convert}(MW/GJ, 1/hr) \\
z_{Boiler} = (CC_L+OM_L)/(PEC_{Boiler}/PEC_{total})/(8776[hr]*UF) \\
\{ \_Boiler = (c_{prod\_Boiler} - c_{fuel\_Boiler}) / c_{fuel\_Boiler} \\
f_{Boiler} = z_{Boiler} / (z_{Boiler} + C_{d\_Boiler}) \\
\]

**Turbine 1**

\[
epsilon_{turb\_1} = (W_{turb\_1}) / (E_[11] - E_[12]) \\
E_{d\_Turb\_1} = (E_[11]) - (E_[12] + W_{Turb\_1}) \\
y_{d\_Turb\_1} = E_{d\_Turb\_1} / E_[11] \\
c_{fuel\_Turb\_1} = (C_[11] - C_[12]) / (E_[11] - E_[12]) \times \text{convert}(\$/hr-MW,\$/GJ) \\
c_{prod\_Turb\_1} = (C_{Turb\_1}) / (W_{Turb\_1}) \times \text{convert}(MW/GJ, 1/hr) \\
z_{Turb\_1} = ((CC_L+OM_L)*(PEC_{Turb\_1}/PEC_{total})/(8776[hr]*UF) \\
\{ \_Turb\_1 = (c_{prod\_Turb\_1} - c_{fuel\_Turb\_1}) / c_{fuel\_Turb\_1} \\
f_{Turb\_1} = z_{Turb\_1} / (z_{Turb\_1} + C_{d\_Turb\_1}) \\
\]

**Turbine 2**

\[
epsilon_{turb\_2} = (W_{turb\_2}) / (E_[13] - E_[14]) \\
E_{d\_Turb\_2} = (E_[13]) - (E_[14] + W_{Turb\_2}) \\
y_{d\_Turb\_2} = E_{d\_Turb\_2} / E_[13] \\
c_{fuel\_Turb\_2} = (C_[13] - C_[14]) / (E_[13] - E_[14]) \times \text{convert}(\$/hr-MW,\$/GJ) \\
c_{prod\_Turb\_2} = (C_{Turb\_2}) / (W_{Turb\_2}) \times \text{convert}(MW/GJ, 1/hr) \\
z_{Turb\_2} = ((CC_L+OM_L)*(PEC_{Turb\_2}/PEC_{total})/(8776[hr]*UF) \\
\{ \_Turb\_2 = (c_{prod\_Turb\_2} - c_{fuel\_Turb\_2}) / c_{fuel\_Turb\_2} \\
f_{Turb\_2} = z_{Turb\_2} / (z_{Turb\_2} + C_{d\_Turb\_2}) \\
\]

**Turbine 3**

\[
epsilon_{turb\_3} = (W_{turb\_3}) / (E_[15] - E_[16]) \\
E_{d\_Turb\_3} = (E_[15]) - (E_[16] + W_{Turb\_3}) \\
y_{d\_Turb\_3} = E_{d\_Turb\_3} / E_[15] \\
c_{fuel\_Turb\_3} = (C_[15] - C_[16]) / (E_[15] - E_[16]) \times \text{convert}(\$/hr-MW,\$/GJ) \\
c_{prod\_Turb\_3} = (C_{Turb\_3}) / (W_{Turb\_3}) \times \text{convert}(MW/GJ, 1/hr) \\
z_{Turb\_3} = ((CC_L+OM_L)*(PEC_{Turb\_3}/PEC_{total})/(8776[hr]*UF) \\
\{ \_Turb\_3 = (c_{prod\_Turb\_3} - c_{fuel\_Turb\_3}) / c_{fuel\_Turb\_3} \\
f_{Turb\_3} = z_{Turb\_3} / (z_{Turb\_3} + C_{d\_Turb\_3}) \\
\]

**Turbine 4**

\[
epsilon_{turb\_4} = (W_{turb\_4}) / (E_[17] - E_[18]) \\
E_{d\_Turb\_4} = (E_[17]) - (E_[18] + W_{Turb\_4}) \\
y_{d\_Turb\_4} = E_{d\_Turb\_4} / E_[17] \\
c_{fuel\_Turb\_4} = (C_[17] - C_[18]) / (E_[17] - E_[18]) \times \text{convert}(\$/hr-MW,\$/GJ) \\
c_{prod\_Turb\_4} = (C_{Turb\_4}) / (W_{Turb\_4}) \times \text{convert}(MW/GJ, 1/hr) \\
z_{Turb\_4} = ((CC_L+OM_L)*(PEC_{Turb\_4}/PEC_{total})/(8776[hr]*UF) \\
\{ \_Turb\_4 = (c_{prod\_Turb\_4} - c_{fuel\_Turb\_4}) / c_{fuel\_Turb\_4} \\
f_{Turb\_4} = z_{Turb\_4} / (z_{Turb\_4} + C_{d\_Turb\_4}) \\
\]

**Condenser**

\[
\{ \epsilon_{condenser} = (E_[coolant\ out] - E_[coolant\ in]) / (E_[18] - E_[1]) \\
E_{d\_condenser} = (E_[18] + E_[coolant\ in]) - (E_[1] + E_[coolant\ out]) \\
y_{d\_condenser} = E_{d\_condenser} / (E_[18] + E_[coolant\ in]) \\
c_{fuel\_condenser} = (C_[18] - C_[1]) / (E_[18] - E_[1]) \times \text{convert}(\$/hr-MW,\$/GJ) \\
\]

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$$c_{\text{prod\_condenser}} = (C_{\text{\_coolant\ out}}-C_{\text{\_coolant\ in}}) / (E_{\text{\_coolant\ out}}-E_{\text{\_coolant\ in}}) \cdot \text{convert}(\$/\text{hr-MW,}\$/\text{GJ})$$

$$C_{\text{\_D\_condenser}} = c_{\text{\_fuel\_condenser}} \cdot E_{\text{\_D\_condenser}} \cdot \text{convert}(\text{MW/GJ, 1/hr})$$

$$z_{\text{\_condenser}} = ((CC_{\_L}+OM_{\_L}) \cdot (\text{PEC_{\_condenser}}/\text{PEC_{\_total}})) / (7446 [\text{hr}])$$

$$r_{\text{\_condenser}} = (c_{\text{\_prod\_condenser}} - c_{\text{\_fuel\_condenser}}) / c_{\text{\_fuel\_condenser}}$$

$$f_{\text{\_condenser}} = z_{\text{\_condenser}} / (z_{\text{\_condenser}} + C_{\text{\_D\_condenser}})$$
"4 Reheats & 4 Regenerations"
"IState Point 0 - Dead State"
\[ T[0] = \text{converttemp}(C,K,25) \]
\[ P[0] = 1 \text{ [atm]} \times \text{convert(atm,kPa)} \]
\[ h[0] = \text{enthalpy(Water, } T=T[0], P=P[0]) \]
\[ s[0] = \text{entropy(Water, } T=T[0], P=P[0]) \]

"IState Point 1 - Condenser Outlet, Pump 1 Inlet"
\[ T[1] = \text{temperature(Water, } P=P[1], x=x[1]) \]
\[ P[1] = 10 \text{ [kPa]} \]
\[ h[1] = \text{enthalpy(Water, } P=P[1], x=x[1]) \]
\[ s[1] = \text{entropy(Water, } P=P[1], x=x[1]) \]
\[ x[1] = 0 \]
\[ e[1] = h[1] - h[0] - T[0] \times (s[1] - s[0]) \]
\[ E_{\text{.1}} = \text{convert(kW,MW)} \times e[1] \times m_{\text{.5}} \]

"IState Point 2 - Pump 1 Outlet, Open FWH 1 inlet"
\[ T[2] = \text{temperature(Water, } P=P[2], h=h[2]) \]
\[ P[2] = 450 \text{[kPa]} \]
\[ s[2] = \text{entropy(Water, } P=P[2], h=h[2]) \]
\[ e[2] = h[2] - h[0] - T[0] \times (s[2] - s[0]) \]
\[ E_{\text{.2}} = \text{convert(kW,MW)} \times e[2] \times m_{\text{.5}} \]

"IState Point 3 - Open FWH 1 Outlet, Pump 2"
\[ T[3] = \text{temperature(Water, } P=P[3], h=h[3]) \]
\[ s[3] = \text{entropy(Water, } P=P[3], h=h[3]) \]
\[ e[3] = h[3] - h[0] - T[0] \times (s[3] - s[0]) \]
\[ E_{\text{.3}} = \text{convert(kW,MW)} \times e[3] \times m_{\text{.4}} \]

"IState Point 4 - Pump 2 outlet, Open FWH 2 inlet"
\[ T[4] = \text{temperature(Water, } P=P[4], h=h[4]) \]
\[ P[4] = 1500 \text{[kPa]} \]
\[ s[4] = \text{entropy(Water, } P=P[4], h=h[4]) \]
\[ E_{\text{.4}} = \text{convert(kW,MW)} \times e[4] \times m_{\text{.4}} \]

"IState Point 5 - Open FWH 2 outlet, Pump 3 inlet"
\[ T[5] = \text{temperature(Water, } P=P[5], h=h[5]) \]
\[ s[5] = \text{entropy(Water, } P=P[5], h=h[5]) \]
\[ e[5] = h[5] - h[0] - T[0] \times (s[5] - s[0]) \]
\[ E_{\text{.5}} = \text{convert(kW,MW)} \times e[5] \times m_{\text{.3}} \]

"IState Point 6 - Pump 3 outlet, Open FWH 3 inlet"
\[ T[6] = \text{temperature(Water, } P=P[6], h=h[6]) \]
\[ P[6] = 5000 \text{[kPa]} \]
\[ s[6] = \text{entropy(Water, } P=P[6], h=h[6]) \]
\[ E_{\text{.6}} = \text{convert(kW,MW)} \times e[6] \times m_{\text{.3}} \]

"IState Point 7 - Open FWH 3 outlet, Pump 4 inlet"
\[ T[7] = \text{temperature(Water, } P=P[7], h=h[7]) \]
\[ s[7] = \text{entropy(Water, } P=P[7], h=h[7]) \]
\[ e[7] = h[7] - h[0] - T[0] \times (s[7] - s[0]) \]
E_[7] = convert(kW,MW)*e[7] * m_.2

"State Point 8 - Pump 4 outlet, Closed FWH 4 inlet"
T[8] = temperature(Water, P=P[8], h=h[8])
P[8] = 15000 [kPa]
s[8] = entropy(Water, P=P[8], h=h[8])
e[8] = h[8] - h[0] - T[0]*(s[8] - s[0])
E_.[8] = convert(kW,MW)*e[8] * m_.2

"State Point 9 - Closed FWH outlet, Mixer inlet"
T[9] = temperature(Water, P=P[9], h=h[9])
P[9] = P[8]
s[9] = entropy(Water, P=P[9], h=h[9])
e[9] = h[9] - h[0] - T[0]*(s[9] - s[0])
E_.[9] = convert(kW,MW)*e[9] * m_.1

"State Point 10 - Mixer outlet, Boiler inlet"
T[10] = temperature(Water, P=P[10], h=h[10])
P[10] = P[8]
s[10] = entropy(Water, P=P[10], h=h[10])
e[10] = h[10] - h[0] - T[0]*(s[10] - s[0])
E_.[10] = convert(kW,MW)*e[10] * m_.1

"State Point 11 - Saturated liquid within Boiler"
x[11] = 0

"State Point 12 - Saturated Steam within Boiler"
T[12] = temperature(Water, P=P[12], x=x[12])
P[12] = P[8]
h[12] = enthalpy(Water, P=P[12], x=x[12])
s[12] = entropy(Water, P=P[12], x=x[12])
x[12] = 1
e[12] = h[12] - h[0] - T[0]*(s[12] - s[0])
E_.[12] = convert(kW,MW)*e[12] * m_.1

"State Point 13 - Boiler outlet, Turbine 1 inlet"
T[13] = converttemp(C,K,600)
h[13] = enthalpy(Water, T=T[13], P=P[13])
s[13] = entropy(Water, T=T[13], P=P[13])
E_.[13] = convert(kW,MW)*e[13] * m_.1

"State Point 14 - Turbine 1 outlet, Reheat 1 inlet, Closed FWH 4 inlet"
T[14] = temperature(Water, P=P[14], h=h[14])
P[14] = 14900 [kPa]
s[14] = entropy(Water, P=P[14], h=h[14])
e[14] = h[14] - h[0] - T[0]*(s[14] - s[0])
E_.[14] = convert(kW,MW)*e[14] * m_.1
"State Point 15 - Reheat 1 outlet, Turbine 2 inlet"
\[ T[15] = \text{converttemp}(C,K,600) \]
\[ h[15] = \text{enthalpy}(\text{Water}, T=T[15], P=P[15]) \]
\[ s[15] = \text{entropy}(\text{Water}, T=T[15], P=P[15]) \]
\[ E_-.[15] = \text{convert}(\text{kW, MW})*e[15] \times m_.2 \]

"State Point 16 - Turbine 2 outlet, Reheat 2 inlet, Open FWH 3 inlet"
\[ T[16] = \text{temperature}(\text{Water}, P=P[16], h=h[16]) \]
\[ s[16] = \text{entropy}(\text{Water}, P=P[16], h=h[16]) \]
\[ e[16] = h[16] - h[0] - T[0]*(s[16] - s[0]) \]
\[ E_-.[16] = \text{convert}(\text{kW, MW})*e[16] \times m_.2 \]

"State Point 17 - Reheat 2 outlet, Turbine 3 inlet"
\[ T[17] = \text{converttemp}(C,K,600) \]
\[ h[17] = \text{enthalpy}(\text{Water}, T=T[17], P=P[17]) \]
\[ s[17] = \text{entropy}(\text{Water}, T=T[17], P=P[17]) \]
\[ e[17] = h[17] - h[0] - T[0]*(s[17] - s[0]) \]
\[ E_-.[17] = \text{convert}(\text{kW, MW})*e[17] \times m_.3 \]

"State Point 18 - Turbine 3 outlet, Reheat 3 inlet, Open FWH 2 inlet"
\[ T[18] = \text{temperature}(\text{Water}, P=P[18], h=h[18]) \]
\[ s[18] = \text{entropy}(\text{Water}, P=P[18], h=h[18]) \]
\[ e[18] = h[18] - h[0] - T[0]*(s[18] - s[0]) \]
\[ E_-.[18] = \text{convert}(\text{kW, MW})*e[18] \times m_.3 \]

"State Point 19 - Reheat 3 outlet, Turbine 4 inlet"
\[ T[19] = \text{converttemp}(C,K,600) \]
\[ h[19] = \text{enthalpy}(\text{Water}, T=T[19], P=P[19]) \]
\[ s[19] = \text{entropy}(\text{Water}, T=T[19], P=P[19]) \]
\[ e[19] = h[19] - h[0] - T[0]*(s[19] - s[0]) \]
\[ E_-.[19] = \text{convert}(\text{kW, MW})*e[19] \times m_.4 \]

"State Point 20 - Turbine 4 outlet, Reheat 4 inlet, Open FWH 1 inlet"
\[ T[20] = \text{temperature}(\text{Water}, P=P[20], h=h[20]) \]
\[ s[20] = \text{entropy}(\text{Water}, P=P[20], h=h[20]) \]
\[ e[20] = h[20] - h[0] - T[0]*(s[20] - s[0]) \]
\[ E_-.[20] = \text{convert}(\text{kW, MW})*e[20] \times m_.4 \]

"State Point 21 - Reheat 4 outlet, Turbine 5 outlet"
\[ T[21] = \text{converttemp}(C,K,600) \]
\[ h[21] = \text{enthalpy}(\text{Water}, T=T[21], P=P[21]) \]
\[ s[21] = \text{entropy}(\text{Water}, T=T[21], P=P[21]) \]
\[ e[21] = h[21] - h[0] - T[0]*(s[21] - s[0]) \]
\[ E_-.[21] = \text{convert}(\text{kW, MW})*e[21] \times m_.5 \]

"State Point 22 - Turbine 5 outlet, condenser inlet"
\[ T[22] = \text{temperature}(\text{Water}, P=P[22], h=h[22]) \]
\[ P[22] = P[1] \]
s[22] = entropy(Water, P=P[22], h=h[22])
e[22] = h[22] - h[0] - T[0]*(s[22] - s[0])
E_[.][22] = convert(kW,MW)*e[22] * m_.5

"!State Point 23 - Closed FWH 4 outlet, Pump 5 inlet"
T[23] = temperature(Water, P=P[23], x=x[23])
P[23] = P[14]
h[23] = enthalpy(Water, P=P[23], x=x[23])
s[23] = entropy(Water, P=P[23], x=x[23])
x[23] = 0
e[23] = h[23] - h[0] - T[0]*(s[23] - s[0])
E_[.][23] = convert(kW,MW)*e[23] * m_.6

"!State Point 24 - Pump 5 outlet, Mixer inlet"
T[24] = temperature(Water, P=P[24], h=h[24])
s[24] = entropy(Water, P=P[24], h=h[24])

"!Control Volumes"

"Pump 1"
eta_Pump_1 = 0.86
h_2_s = enthalpy(Water, P=P[2], s=s[1])
eta_Pump_1 = (h_2_s - h[1]) / (h[2] - h[1])
W_.Pump_1 = m_.5*(h[2] - h[1])*convert(kW,MW)

"Open FWH 1"

"Pump 2"
eta_Pump_2 = 0.86
h_4_s = enthalpy(Water, P=P[4], s=s[3])
eta_Pump_2 = (h_4_s - h[3]) / (h[4] - h[3])
W_.Pump_2 = m_.4*(h[4] - h[3])*convert(kW,MW)

"Open FWH 2"

"Pump 3"
eta_Pump_3 = 0.86
h_6_s = enthalpy(Water, P=P[6], s=s[5])
eta_Pump_3 = (h_6_s - h[5]) / (h[6] - h[5])
W_.Pump_3 = m_.3*(h[6] - h[5])*convert(kW,MW)

"Open FWH 3"

"Pump 4"
eta_Pump_4 = 0.86
h_8_s = enthalpy(Water, P=P[8], s=s[7])
eta_Pump_4 = (h_8_s - h[7]) / (h[8] - h[7])
W_.Pump_4 = m_.2*(h[8] - h[7])*convert(kW,MW)

"Closed FWH 4"
Q_.FWH_4 = m_.6*(h[14] - h[23])*convert(kW,MW)
Q_.FWH_4 = m_.2*(h[9] - h[8])*convert(kW,MW)

"Mixer"

"Pump 5"
eta_Pump_5 = 0.86
h_24_s = enthalpy(Water, P=P[24], s=s[23])
eta_Pump_5 = (h_24_s - h[23]) / (h[24] - h[23])
W_.Pump_5 = m_.6*(h[24] - h[23])*convert(kW,MW)

"Boiler & Reheater"
eta_boiler = 0.85
HHV_coal = 27.135 [MJ/kg]
Q_.boiler = (HHV_coal* m_.c)*eta_boiler
Q_.effluent = (Q_.boiler/eta_boiler)*(1 - eta_boiler)

"C51.2 H43 O4.2 N0.9 S0.8 + (O2 + N2 + H2O) --> CO2 + H2O + SO2 + N2"
e_Coil = 27680 [kJ/kg]
T_air = 298.15 [K]
P_air = 101.13 [kPa]
h_air = enthalpy(Air, T=T_air)
s_air = entropy(Air, T=T_air, P=P_air)
e_air = h_air - enthalpy(Air, T=T[0]) - T[0]*(s_air-entropy(Air, T=T[0],P=P[0]))
AFR_theoretical = 11.36
AFR_theoretical = m_.air_th / m_.c
m_.air = 1.15 * m_.air_th
E_air = (e_air*m_.air)*convert(kW,MW)
E_coal = e_coal*m_.c*convert(kW,MW)
c_coal = 55 ['$/ton']*convert(1/ton, 1/kg)

"Turbine 1"
eta_Turb = 0.86
h_14_s = enthalpy(Water, P=P[14], s=s[13])
eta_Turb = (h[15] - h[16]) / (h[15] - h_16_s)
W_.Turb_1 = m_.2*(h[15] - h[16])*convert(kW,MW)

"Split 1"
m_.1 = m_.2 + m_.6
m_.2 = m_.1*(1-w)
m_.6 = m_.1*(w)

"Turbine 2"
h_16_s = enthalpy(Water, P=P[16], s=s[15])
eta_Turb = (h[15] - h[16]) / (h[15] - h_16_s)
W_.Turb_2 = m_.2*(h[15] - h[16])*convert(kW,MW)

"Split 2"
m_.2 = m_.3 + m_.7
m_.3 = m_.2*(1-x)
m_.7 = m_.2*(x)

"Turbine 3"
h_18_s = enthalpy(Water, P=P[18], s=s[17])
eta_Turb = (h[17] - h[18]) / (h[17] - h_16_s)
W_.Turb_3 = m_.3*(h[17] - h[18])*convert(kW,MW)

"Split 3"
m_.3 = m_.4 + m_.8
m_.4 = m_.3*(1-y)
m_.8 = m_.3*(y)

"Turbine 4"
h_20_s = enthalpy(Water, P=P[20], s=s[19])
eta_Turb = (h[19] - h[20]) / (h[19] - h_20_s)
W_.Turb_4 = m_.4*(h[19] - h[20])*convert(kW,MW)

"Split 4"
m_.4 = m_.5 + m_.9
m_.5 = m_.4*(1-z)
m_.9 = m_.4*(z)

"Turbine 5"
h_22_s = enthalpy(Water, P=P[22], s=s[21])
eta_Turb = (h[21] - h[22]) / (h[21] - h_22_s)
W_.Turb_5 = m_.5*(h[21] - h[22])*convert(kW,MW)

"Condenser"
Q_.Condenser = (m_.5*(h[22] - h[1]))*convert(kW,MW)
W_.Cond_fans = (7.4133[kW/MW]*Q_.Condenser + 2.9342[kW])*convert(kW,MW)

W_.out = 1000 [MW]
W_.out = W_.Turb - W_.Pump
W_.Turb = W_.Turb_1 + W_.Turb_2 + W_.Turb_3 + W_.Turb_4 + W_.Turb_5
W_.Pump = W_.Pump_1 + W_.Pump_2 + W_.Pump_3 + W_.Pump_4 + W_.Pump_5
eta_th = W_.out / Q_.Boiler

"Cost Calculations"
i=0.05
i_eff = ((1+(i/m))^m)-1
r_n=0.04
r_n_coal = 0.039
UF = 0.85
n = 20
m = 12
k= (1+r_n)/(1+i_eff)
CELF = ((k*(1-k^n))/(1-k))*CRF
CRF = (i_eff*(1+i_eff)^n)/(1+i_eff)^n-1
C_.coal = c_coal * m_.c * convert($/s,$/hr)
C_.air = 0 [$/hr]
OM_L = (0.2*PEC_total)*CELF
CC_L = (4*PEC_total)*CRF
TRR_L = (OM_L + CC_L)/(8776[hr])*UF + C_.coal
Cost_min_elec = (TRR_L / W_.out)*convert($/MW-hr, $/kW-hr)
PEC_Boiler = 0.367[$/(Btu/hr)^.77]*((Q_.Boiler*convert(MW,Btu/hr))^0.77)
PEC_Pump_1=-0.0988[($/(kg/s)^2]*((m_.5)^2)+232.61[($/(kg/s))*m_.5]+45142[$]
PEC_Pump_2=-0.0988[($/(kg/s)^2)*((m_.4)^2)+232.61[($/(kg/s))*m_.4]+45142[$]
PEC\_Pump\_3=-0.0988\[$/(kg/s)^2\]*(m\_3^2)+232.61\[$/(kg/s)]*(m\_3)+45142\[$]
PEC\_Pump\_4=-0.0988\[$/(kg/s)^2\]*(m\_2^2)+232.61\[$/(kg/s)]*(m\_2)+45142\[$]
PEC\_Pump\_5=-0.0988\[$/(kg/s)^2\]*(m\_6^2)+232.61\[$/(kg/s)]*(m\_6)+45142\[$]

PEC\_Turb\_total = 2.6175e6\[$/MW^0.7\]*(W\_Turb^0.7)
PEC\_Turb\_1 = PEC\_Turb\_total*(W\_Turb\_1 / W\_Turb)
PEC\_Turb\_2 = PEC\_Turb\_total*(W\_Turb\_2 / W\_Turb)
PEC\_Turb\_3 = PEC\_Turb\_total*(W\_Turb\_3 / W\_Turb)
PEC\_Turb\_4 = PEC\_Turb\_total*(W\_Turb\_4 / W\_Turb)
PEC\_Turb\_5 = PEC\_Turb\_total*(W\_Turb\_5 / W\_Turb)

U\_Steel = .680 [kW/m^2-K]
a\_cost\_Steel=1.75 "Stainless Steel"
b=.13

LMTD\_FWH\_4= ((T[14]-T[23])-(T[9]-T[8]))/ln((T[14]-T[23])/(T[9]-T[8]))
Q\_FWH\_4*convert(MW,kW) = U\_Steel*A\_FWH\_4*LMTD\_FWH\_4
C\_B\_FWH\_4=exp(11.0545-.9228*ln(A\_FWH\_4*1[m^2]))+0.9861*(ln(A\_FWH\_4*1[m^2]))^2)
F\_M\_FWH\_4=(a\_cost\_Steel+(A\_FWH\_4/100[m^2]))^b
F\_P\_FWH\_4=.9803+.018*(P[8]/100[kPa])+.0017[kPa^2]*(P[8])^2
PEC\_FWH\_4=0.79\[$]*C\_B\_FWH\_4*F\_M\_FWH\_4*F\_P\_FWH\_4

PEC\_condenser=68.761\[$/MW]*Q\_Condenser+137592\[$]

PEC\_total=PEC\_Boiler+PEC\_Pump\_1+PEC\_Pump\_2+PEC\_Pump\_3+PEC\_Pump\_4+PEC\_Pump\_5+PEC\_Turb\_total+PEC\_FWH\_4+PEC\_condenser

DC = PEC\_total*2.49
IC = DC * 0.2645
TCI = IC + DC + ((OM\_L/ 12) + (0.1*OM\_L*UF)/12 + C\_.coal*142.8[hr] + (0.02*(IC+DC))) + (C\_.coal*1241[hr])*1.25

{C\_[1] = c[1] * E\_[1]*convert($/s, $/hr)
C\_[2] = c[2] * E\_[2]*convert($/s, $/hr)
C\_[3] = c[3] * E\_[3]*convert($/s, $/hr)
C\_[5] = c[5] * E\_[5]*convert($/s, $/hr)
C\_[6] = c[6] * E\_[6]*convert($/s, $/hr)
C\_[7] = c[7] * E\_[7]*convert($/s, $/hr)
C\_[8] = c[8] * E\_[8]*convert($/s, $/hr)
C\_[12] = c[12] * E\_[12]*convert($/s, $/hr)
C\_[14] = c[14] * E\_[14]*convert($/s, $/hr)
C\_[15] = c[15] * E\_[15]*convert($/s, $/hr)
C\_[16] = c[16] * E\_[16]*convert($/s, $/hr)
C\_[17] = c[17] * E\_[17]*convert($/s, $/hr)
C\_[18] = c[18] * E\_[18]*convert($/s, $/hr)
C\_[19] = c[19] * E\_[19]*convert($/s, $/hr)
C\_[20] = c[20] * E\_[20]*convert($/s, $/hr)
C\_[21] = c[21] * E\_[21]*convert($/s, $/hr)
C\_[22] = c[22] * E\_[22]*convert($/s, $/hr)
C\_[23] = c[23] * E\_[23]*convert($/s, $/hr)
C\_[24] = c[24] * E\_[24]*convert($/s, $/hr)
C\_.Pump\_1 = c\_work * W\_.Pump\_1*convert($/s, $/hr)
C\_.Pump\_2 = c\_work * W\_.Pump\_2*convert($/s, $/hr)
\[ C_{\text{Pump}}_3 = c_{\text{work}} \times W_{\text{Pump}}_3 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Pump}}_4 = c_{\text{work}} \times W_{\text{Pump}}_4 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Pump}}_5 = c_{\text{work}} \times W_{\text{Pump}}_5 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Turb}}_1 = c_{\text{work}} \times W_{\text{Turb}}_1 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Turb}}_2 = c_{\text{work}} \times W_{\text{Turb}}_2 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Turb}}_3 = c_{\text{work}} \times W_{\text{Turb}}_3 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Turb}}_4 = c_{\text{work}} \times W_{\text{Turb}}_4 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Turb}}_5 = c_{\text{work}} \times W_{\text{Turb}}_5 \times \text{convert}($/s, $/hr) \]
\[ C_{\text{Condenser}} = c_{\text{work}} \times W_{\text{Cond_fans}} \times \text{convert}($/s, $/hr) \times \text{convert}($/GJ, $/MJ) \]
\[ C_{\text{work_out}} = c_{\text{work}} \times W_{\text{out}} \times \text{convert}($/s, $/hr) \times \text{convert}($/GJ, $/MJ) \]

"Cost Accounting"
\[ C_{[10]} + C_{[14]} \times (1-w) + C_{[16]} \times (1-x) + C_{[18]} \times (1-y) + C_{[20]} \times (1-z) + C_{\text{coal}} + C_{\text{air}} + z_{\text{Boiler}} = \]
\[ C_{[13]} + C_{[15]} + C_{[17]} + C_{[19]} + C_{[21]} \]
\[ C_{[1]} + C_{\text{pump}}_1 + z_{\text{Pump}}_1 = C_{[2]} \]
\[ C_{[3]} + C_{\text{pump}}_2 + z_{\text{Pump}}_2 = C_{[4]} \]
\[ C_{[5]} + C_{\text{pump}}_3 + z_{\text{Pump}}_3 = C_{[6]} \]
\[ C_{[7]} + C_{\text{pump}}_4 + z_{\text{Pump}}_4 = C_{[8]} \]
\[ C_{[23]} + C_{\text{pump}}_5 + z_{\text{Pump}}_5 = C_{[24]} \]
\[ C_{[2]} + C_{[20]} \times (x) + z_{\text{FWH}}_1 = C_{[3]} \]
\[ C_{[4]} + C_{[18]} \times (y) + z_{\text{FWH}}_2 = C_{[5]} \]
\[ C_{[6]} + C_{[16]} \times (x) + z_{\text{FWH}}_3 = C_{[7]} \]
\[ C_{[8]} + C_{[14]} \times (w) + z_{\text{FWH}}_4 = C_{[9]} + C_{[23]} \]
\[ C_{[9]} + C_{[24]} + z_{\text{Mixer}} = C_{[10]} \]
\[ C_{[13]} + z_{\text{Turb}}_1 = C_{[14]} + C_{\text{Turb}}_1 \]
\[ C_{[15]} + z_{\text{Turb}}_2 = C_{[16]} + C_{\text{Turb}}_2 \]
\[ C_{[17]} + z_{\text{Turb}}_3 = C_{[18]} + C_{\text{Turb}}_3 \]
\[ C_{[19]} + z_{\text{Turb}}_4 = C_{[20]} + C_{\text{Turb}}_4 \]
\[ C_{[21]} + z_{\text{Turb}}_5 = C_{[22]} + C_{\text{Turb}}_5 \]
\[ C_{[22]} + C_{\text{air}} + z_{\text{condenser}} + C_{\text{condenser}} = C_{[1]} + C_{\text{air_out}} \]

\[ c_{[13]} = c_{[14]} \]
\[ c_{[15]} = c_{[16]} \]
\[ c_{[17]} = c_{[18]} \]
\[ c_{[19]} = c_{[20]} \]
\[ c_{[21]} = c_{[22]} \]
\[ c_{[14]} = c_{[23]} \]

Variable = \( (C_{[13]} - C_{[10]}) / (E_{[13]} - E_{[10]}) \)

Variable = \( (C_{[15]} - C_{[14]} \times (1-w)) / (E_{[15]} - E_{[14]} \times (1-w)) \)

Variable = \( (C_{[17]} - C_{[16]} \times (1-x)) / (E_{[17]} - E_{[16]} \times (1-x)) \)

Variable = \( (C_{[19]} - C_{[18]} \times (1-y)) / (E_{[19]} - E_{[18]} \times (1-y)) \)

Variable = \( (C_{[21]} - C_{[20]} \times (1-z)) / (E_{[21]} - E_{[20]} \times (1-z)) \)

"Exergoeconomic Parameters"

"Pump 1"
\[ \epsilon_{\text{Pump}}_1 = (E_{[2]} - E_{[1]}) / (W_{\text{pump}}_1) \]
\[ E_{D_{\text{Pump}}_1} = (E_{[1]} + W_{\text{pump}}_1) - (E_{[2]}) \]
\[ y_{D_{\text{Pump}}_1} = E_{D_{\text{Pump}}_1} / (E_{[1]} + W_{\text{pump}}_1) \]
\[ (c_{\text{fuel}}_{\text{Pump}}_1) = (C_{\text{Pump}}_1) / (W_{\text{pump}}_1) \times \text{convert}($/hr-MW, $/GJ) \]
\[ c_{\text{prod}}_{\text{Pump}}_1 = (C_{[2]} - C_{[1]}) / (E_{[2]} - E_{[1]}) \times \text{convert}($/hr-MW, $/GJ) \]
\[ C_{D_{\text{Pump}}_1} = c_{\text{fuel}}_{\text{Pump}}_1 \times E_{D_{\text{Pump}}_1} \times \text{convert}(MW/GJ, 1/hr) \]
\[ z_{\text{Pump}}_1 = (CC_{L} + OM_{L}) \times \text{PEC}_{\text{Pump}}_1 / \text{PEC}_{\text{total}} \times \text{convert}(8776 [hr] \times UF) \]
\[ (r_{\text{Pump}}_1 = (c_{\text{prod}}_{\text{Pump}}_1 - c_{\text{fuel}}_{\text{Pump}}_1) / c_{\text{fuel}}_{\text{Pump}}_1 \]

"Open FWH 1"
"I" z_.FWH_1 = 50 [$/hr]

"Pump 2"
epsilon_pump_2 = (E_.[4] - E_.[3]) / (W_.pump_2)
E_.D_Pump_2 = (E_.[3] + W_.Pump_4) - (E_.[4])
y_D_Pump_2 = E_.D_Pump_2 / (E_.[3] + W_.Pump_2)
{c_fuel_Pump_2 = (E_.D_Pump_2) / (W_.Pump_2)*convert($/hr-MW,$/GJ)
c_prod_Pump_2 = (E_.[4]-E_.[3]) / (E_.[4]-E_.[3])*convert($/hr-MW,$/GJ)
C_.D_Pump_2 = c_fuel_Pump_2 * E_.D_Pump_2*convert(MW/GJ, 1/hr)
z_.Pump_2=((CC_L+OM_L)*(PEC_Pump_2/PEC_total))/(8776/hr)*UF
{r_Pump_2= (c_prod_Pump_2 - c_fuel_Pump_2) / c_fuel_Pump_2
f_Pump_2 = z_.Pump_2 / (z_.Pump_2 + C_.D_Pump_2)}

"Open FWH 2"
"I" z_.FWH_2 = 50 [$/hr]

"Pump 3"
epsilon_pump_3 = (E_.[6] - E_.[5]) / (W_.pump_3)
E_.D_Pump_3 = (E_.[5] + W_.Pump_3) - (E_.[6])
{c_fuel_Pump_3 = (E_.D_Pump_3) / (W_.Pump_3)*convert($/hr-MW,$/GJ)
c_prod_Pump_3 = (E_.[6]-E_.[5]) / (E_.[6]-E_.[5])*convert($/hr-MW,$/GJ)
C_.D_Pump_3 = c_fuel_Pump_3 * E_.D_Pump_3*convert(MW/GJ, 1/hr)
z_.Pump_3=((CC_L+OM_L)*(PEC_Pump_3/PEC_total))/(8776/hr)*UF
{r_Pump_3= (c_prod_Pump_3 - c_fuel_Pump_3) / c_fuel_Pump_3
f_Pump_3 = z_.Pump_3 / (z_.Pump_3 + C_.D_Pump_3)}

"Open FWH 3"
"I" z_.FWH_3 = 50 [$/hr]

"Pump 4"
epsilon_pump_4 = (E_.[8] - E_.[7]) / (W_.pump_4)
E_.D_Pump_4 = (E_.[7] + W_.Pump_4) - (E_.[8])
{c_fuel_Pump_4 = (E_.D_Pump_4) / (W_.Pump_4)*convert($/hr-MW,$/GJ)
c_prod_Pump_4 = (E_.[8]-E_.[7]) / (E_.[8]-E_.[7])*convert($/hr-MW,$/GJ)
C_.D_Pump_4 = c_fuel_Pump_4 * E_.D_Pump_4*convert(MW/GJ, 1/hr)
z_.Pump_4=((CC_L+OM_L)*(PEC_Pump_4/PEC_total))/(8776/hr)*UF
{r_Pump_4= (c_prod_Pump_4 - c_fuel_Pump_4) / c_fuel_Pump_4
f_Pump_4 = z_.Pump_4 / (z_.Pump_4 + C_.D_Pump_4)}

"Closed FWH 4"
epsilon_FWH_4 = (E_.[9] - E_.[8]) / (E_.[9]-E_.[8])
E_.D_FWH_4 = (E_.[8] + E_.[14]) - (E_.[9] + E_.[23])
y_D_FWH_4 = E_.D_FWH_4 / (E_.[8] + E_.[14])
{c_fuel_FWH_4 = (E_.[9]-E_.[8]) / (E_.[9]-E_.[8])*convert($/hr-MW,$/GJ)
c_prod_FWH_4 = (E_.[9]-E_.[8]) / (E_.[9]-E_.[8])*convert($/hr-MW,$/GJ)
C_.D_FWH_4 = c_fuel_FWH_4 * E_.D_FWH_4*convert(MW/GJ, 1/hr)
z_.FWH_4=((CC_L+OM_L)*(PEC_FWH_4/PEC_total))/(8776/hr)*UF
{r_FWH_4= (c_prod_FWH_4 - c_fuel_FWH_4) / c_fuel_FWH_4
f_FWH_4 = z_.FWH_4 / (z_.FWH_4 + C_.D_FWH_4)}

"Mixer"
"I" z_.Mixer = 50 [$/hr]

"Pump 5"
\[ \epsilon_{\text{pump}_5} = \frac{E_{\text{-}[24]} - E_{\text{-}[23]}}{W_{\text{pump}_5}} \]
\[ y_{\text{D}_\text{Pump}_5} = \frac{E_{\text{-}[23]} + W_{\text{pump}_5} - (E_{\text{-}[24])}{E_{\text{-}[23]} + W_{\text{pump}_5}} \]
\[ c_{\text{fuel}_\text{Pump}_5} = \frac{(C_{\text{-}[13]} - C_{\text{-}[14]})(PEC_{\text{Pump}_5}/PEC_{\text{total}})}{(8776 \text{[hr]})} \]
\[ f_{\text{Pump}_5} = \frac{z_{\text{Pump}_5}}{(z_{\text{Pump}_5} + C_{\text{-}[13]} - C_{\text{-}[14]})} \]

\[ \epsilon_{\text{Boiler}} = \frac{((E_{\text{-}[13]} - E_{\text{-}[10]} + (E_{\text{-}[15]} - E_{\text{-}[14]})(1-w) + (E_{\text{-}[17]} - E_{\text{-}[16]})(1-x) + (E_{\text{-}[19]} - E_{\text{-}[18]})(1-y) + (E_{\text{-}[21]} - E_{\text{-}[20]})(1-z))}{(E_{\text{air}} + E_{\text{coal}})} \]
\[ E_{\text{-}[13]} + E_{\text{-}[15]} + E_{\text{-}[17]} + E_{\text{-}[19]} + E_{\text{-}[21]} \]
\[ y_{\text{D}_\text{Boiler}} = \frac{E_{\text{-}[13]} + E_{\text{-}[15]} + E_{\text{-}[17]} + E_{\text{-}[19]} + E_{\text{-}[21]} + E_{\text{coal}} + E_{\text{air}}}{E_{\text{-}[13]} + E_{\text{-}[15]} + E_{\text{-}[17]} + E_{\text{-}[19]} + E_{\text{-}[21]} + E_{\text{coal}} + E_{\text{air}} - (E_{\text{-}[13]} + E_{\text{-}[15]} + E_{\text{-}[17]} + E_{\text{-}[19]} + E_{\text{-}[21]}} \]
\[ c_{\text{fuel}_\text{Boiler}} = \frac{(C_{\text{coal}} + C_{\text{air}})}{(E_{\text{coal}} + E_{\text{air}})} \]
\[ c_{\text{prod}_\text{Boiler}} = \frac{(C_{\text{-}[11]} - C_{\text{-}[8]} + (C_{\text{-}[14]} - C_{\text{-}[13]}))}{((E_{\text{-}[11]} - E_{\text{-}[8]} + (E_{\text{-}[14]} - E_{\text{-}[13]}))} \]
\[ f_{\text{Boiler}} = \frac{z_{\text{Boiler}}}{(z_{\text{Boiler}} + C_{\text{-}[13]} - C_{\text{-}[14]})} \]

\[ \epsilon_{\text{turb}_1} = \frac{W_{\text{turb}_1}}{E_{\text{-}[13]} - E_{\text{-}[14]}} \]
\[ E_{\text{-}[13]} - E_{\text{-}[14]} + W_{\text{Turb}_1} \]
\[ y_{\text{D}_\text{Turb}_1} = \frac{E_{\text{-}[13]} - E_{\text{-}[14]} + W_{\text{Turb}_1}}{E_{\text{-}[13]} - E_{\text{-}[14]}} \]
\[ c_{\text{fuel}_\text{Turb}_1} = \frac{(C_{\text{-}[13]} - C_{\text{-}[14]})(PEC_{\text{Turb}_1}/PEC_{\text{total}})}{(8776 \text{[hr]})} \]
\[ f_{\text{Turb}_1} = \frac{z_{\text{Turb}_1}}{(z_{\text{Turb}_1} + C_{\text{-}[13]} - C_{\text{-}[14]})} \]

\[ \epsilon_{\text{turb}_2} = \frac{W_{\text{turb}_2}}{E_{\text{-}[15]} - E_{\text{-}[16]}} \]
\[ E_{\text{-}[15]} - E_{\text{-}[16]} + W_{\text{Turb}_2} \]
\[ y_{\text{D}_\text{Turb}_2} = \frac{W_{\text{Turb}_2}}{E_{\text{-}[15]} - E_{\text{-}[16]}} \]
\[ c_{\text{fuel}_\text{Turb}_2} = \frac{(C_{\text{-}[15]} - C_{\text{-}[16]})(PEC_{\text{Turb}_2}/PEC_{\text{total}})}{(8776 \text{[hr]})} \]
\[ f_{\text{Turb}_2} = \frac{z_{\text{Turb}_2}}{(z_{\text{Turb}_2} + C_{\text{-}[15]} - C_{\text{-}[16]})} \]

\[ \epsilon_{\text{turb}_3} = \frac{W_{\text{turb}_3}}{E_{\text{-}[17]} - E_{\text{-}[18]}} \]
\[ E_{\text{-}[17]} - E_{\text{-}[18]} + W_{\text{Turb}_3} \]
\[ y_{\text{D}_\text{Turb}_3} = \frac{W_{\text{Turb}_3}}{E_{\text{-}[17]} - E_{\text{-}[18]}} \]
\[ c_{\text{fuel}_\text{Turb}_3} = \frac{(C_{\text{-}[17]} - C_{\text{-}[18]})(PEC_{\text{Turb}_3}/PEC_{\text{total}})}{(8776 \text{[hr]})} \]
\[ f_{\text{Turb}_3} = \frac{z_{\text{Turb}_3}}{(z_{\text{Turb}_3} + C_{\text{-}[17]} - C_{\text{-}[18]})} \]
\[
f_{\text{Turb}_3} = \frac{z_{\text{Turb}_3}}{z_{\text{Turb}_3} + C_{\text{D.Turb}_3}}
\]

"Turbine 4"

\[
\text{epsilon}_\text{turb}_4 = \frac{W_{\text{turb}_4}}{E_{[19]} - E_{[20]}}
\]

\[
E_{\text{D.Turb}_4} = \frac{(E_{[19]} - (E_{[20]} + W_{\text{Turb}_4})}{E_{[19]}}
\]

\[
y_{\text{D.Turb}_4} = \frac{E_{\text{D.Turb}_4}}{E_{[19]} - (E_{[20]} + W_{\text{Turb}_4})
\]

\[
c_{\text{fuel.Turb}_4} = \frac{(C_{[19]} - C_{[20]})}{E_{[19]} - E_{[20]}} * \text{convert}\left(\frac{\$/hr-MW}{\$/GJ}\right)
\]

\[
c_{\text{prod.Turb}_4} = \frac{C_{\text{Turb}_4}}{W_{\text{Turb}_4}} * \text{convert}\left(\frac{\$/hr-MW}{\$/GJ}\right)
\]

\[
C_{\text{D.Turb}_4} = c_{\text{fuel.Turb}_4} * E_{\text{D.Turb}_4} * \text{convert}\left(\frac{MW}{GJ}, \frac{1}{hr}\right)
\]

\[
z_{\text{Turb}_4} = \frac{((CC_L + OM_L) * (PEC_{\text{Turb}_4}/PEC_{\text{total}}))}{8776[hr]*UF}
\]

\[
\text{r.Turb}_4 = \frac{(c_{\text{prod.Turb}_4} - c_{\text{fuel.Turb}_4})}{c_{\text{fuel.Turb}_4}
\]

\[
f_{\text{Turb}_4} = \frac{z_{\text{Turb}_4}}{z_{\text{Turb}_4} + C_{\text{D.Turb}_4}}
\]

"Turbine 5"

\[
\text{epsilon}_\text{turb}_5 = \frac{W_{\text{turb}_5}}{E_{[21]} - E_{[22]}}
\]

\[
E_{\text{D.Turb}_5} = \frac{(E_{[21]} - (E_{[22]} + W_{\text{Turb}_5})}{E_{[21]}}
\]

\[
y_{\text{D.Turb}_5} = \frac{E_{\text{D.Turb}_5}}{E_{[21]} - (E_{[22]} + W_{\text{Turb}_5})}
\]

\[
c_{\text{fuel.Turb}_5} = \frac{(C_{[21]} - C_{[22]})}{E_{[21]} - E_{[22]}} * \text{convert}\left(\frac{\$/hr-MW}{\$/GJ}\right)
\]

\[
c_{\text{prod.Turb}_5} = \frac{C_{\text{Turb}_5}}{W_{\text{Turb}_5}} * \text{convert}\left(\frac{\$/hr-MW}{\$/GJ}\right)
\]

\[
C_{\text{D.Turb}_5} = c_{\text{fuel.Turb}_5} * E_{\text{D.Turb}_5} * \text{convert}\left(\frac{MW}{GJ}, 1/hr\right)
\]

\[
z_{\text{Turb}_5} = \frac{((CC_L + OM_L) * (PEC_{\text{Turb}_5}/PEC_{\text{total}}))}{8776[hr]*UF}
\]

\[
\text{r.Turb}_5 = \frac{(c_{\text{prod.Turb}_5} - c_{\text{fuel.Turb}_5})}{c_{\text{fuel.Turb}_5}
\]

\[
f_{\text{Turb}_5} = \frac{z_{\text{Turb}_5}}{z_{\text{Turb}_5} + C_{\text{D.Turb}_5}}
\]
"Basic Rankine Cycle"
"State Point 0 - Dead State"
T[0] = 280 [K]
P[0] = 1 [atm] \* convert(atm,kPa)
h[0] = enthalpy(Water, T=T[0], P=P[0])
s[0] = entropy(Water, T=T[0], P=P[0])

"State Point 1 - Condenser Outlet, Pump 1 inlet"
T[1] = temperature(Water, P=P[1], x=x[1])
P[1] = 10 [kPa]
h[1] = enthalpy(Water, P=P[1], x=x[1])
s[1] = entropy(Water, P=P[1], x=x[1])
x[1] = 0
e[1] = h[1] - h[0] - T[0]*(s[1] - s[0])
E_.[1] = convert(kW,MW)\*e[1] * m_.w

"Pump 1"
eta_pump_1 = (h_2_s - h[1]) / (h[2] - h[1])
h_2_s = enthalpy(Water, P=P[2], s=s[1])
eta_pump_1 = 0.86

"State Point 2 - Pump 1 outlet, Boiler Inlet"
T[2] = temperature(Water, P=P[2], h=h[2])
P[2] = 15000 [kPa]
s[2] = entropy(Water, P=P[2], h=h[2])
E_.[2] = convert(kW,MW)\*e[2] * m_.w

"State Point 3 - Sat Liquid, Boiler"
T[3] = temperature(Water, P=P[3], x=0)
h[3] = enthalpy(Water, P=P[3], x=0)
s[3] = entropy(Water, P=P[3], x=0)
E_.[3] = convert(kW,MW)\*e[3] * m_.w

"State Point 4 - Sat Steam, Boiler"
T[4] = temperature(Water, P=P[4], x=1)
h[4] = enthalpy(Water, P=P[4], x=1)
s[4] = entropy(Water, P=P[4], x=1)
E_.[4] = convert(kW,MW)\*e[4] * m_.w

"State Point 5 - Boiler Outlet, Turbine Inlet"
T[5] = converttemp(C,K, 600)
P[5] = 15000[kPa]
h[5] = enthalpy(Water, T=T[5], P=P[5])
s[5] = entropy(Water, T=T[5], P=P[5])
E_.[5] = convert(kW,MW)\*e[5] * m_.w

"Turbine 1 - High Pressure Turbine"
h_6_s = enthalpy(Water, P=P[6], s=s[5])
$$\eta_{turb\_1} = 0.86$$

"State Point 6 - Turbine Outlet, Condenser Inlet"

\[ T[6] = \text{temperature(Water, } P=P[6], \ h=h[6]) \]
\[ P[6] = 10 \ [\text{kPa}] \]
\[ \{h[6] = \text{enthalpy(Water, } P=P[6], \ s=s[6])\} \]
\[ s[6] = \text{entropy(Water, } x=1, \ P=P[6]) \]
\[ E\_{\_6} = \text{convert(kW,MW)}*e[6] \times m\_w \]

"Boiler Calculations"

\[ Q\_\text{boiler} = m\_w*(h[5]-h[2])*\text{convert(kW,MW)} \]
\[ W\_\text{turb} = m\_w*(h[5] - h[6])*\text{convert(kW,MW)} \]
\[ Q\_\text{cond} = m\_w*(h[6] - h[1])*\text{convert(kW,MW)} \]
\[ W\_\text{pump} = m\_w*(h[2] - h[1])*\text{convert(kW,MW)} \]

\{W\_\text{pump} + Q\_\text{boiler} = W\_\text{turb} + Q\_\text{cond}\}
\[ W\_\text{turb} = 1000[\text{MW}] \]
\[ W\_\text{net} = W\_\text{turb} + W\_\text{pump} \]

$$\eta_{th} = 1 - (Q\_\text{cond}/Q\_\text{boiler})$$
"5 Reheats"
"!State Point 0 - Dead State"
T[0] = 280 [K]
P[0] = 1 [atm] * convert(atm,kPa)
h[0] = enthalpy(Water, T=T[0], P=P[0])
s[0] = entropy(Water, T=T[0], P=P[0])

"!State Point 1 - Condenser Outlet, Pump 1 inlet"
T[1] = temperature(Water, P=P[1], x=x[1])
P[1] = 10 [kPa]
h[1] = enthalpy(Water, P=P[1], x=x[1])
s[1] = entropy(Water, P=P[1], x=x[1])
x[1] = 0
e[1] = h[1] - h[0] - T[0]*(s[1] - s[0])
E_.[1] = convert(kW,MW)*e[1] * m_.w

"Pump 1"
eta_pump_1 = (h_2_s - h[1]) / (h[2] - h[1])
h_2_s = enthalpy(Water, P=P[2], s=s[1])
eta_pump_1 = 0.86

"!State Point 2 - Pump 1 outlet, Boiler Inlet"
T[2] = temperature(Water, P=P[2], h=h[2])
P[2] = 15000 [kPa]
s[2] = entropy(Water, P=P[2], h=h[2])
E_.[2] = convert(kW,MW)*e[2] * m_.w

"!State Point 3 - Sat Liquid, Boiler"
T[3] = temperature(Water, P=P[3], x=0)
h[3] = enthalpy(Water,P=P[3],x=0)
s[3] = entropy(Water, P=P[3], x=0)
E_.[3] = convert(kW,MW)*e[3] * m_.w

"!State Point 4 - Sat Steam, Boiler"
T[4] = temperature(Water, P=P[4],x=1)
h[4] = enthalpy(Water,P=P[4],x=1)
s[4] = entropy(Water, P=P[4], x=1)

"!State Point 5 - Boiler Outlet, Turbine 1 Inlet"
T[5] = converttemp(C,K, 600)
h[5] = enthalpy(Water, T=T[5], P=P[5])
s[5] = entropy(Water, T=T[5], P=P[5])
E_.[5] = convert(kW,MW)*e[5] * m_.w

"Turbine 1 - Highest Pressure Turbine"
h_6_s = enthalpy(Water, P=P[6], s=s[5])
\[
\text{eta}_\text{turb}_1 = 0.86
\]

"State Point 6 - Turbine 1 Outlet, Reheat 1 Inlet"
\[
T[6] = \text{temperature}(\text{Water}, P=P[6], h=h[6])
\]
\[
P[6] = 7108 \text{ [kPa]}
\]
\[
s[6] = \text{entropy}(\text{Water}, T=T[6], P=P[6])
\]
\[
\]
\[
E_.[6] = \text{convert}(\text{kW/MW}) \cdot e[6] \cdot m_\text{w}
\]

"State Point 7 - Reheat 1 Outlet, Turbine 2 Inlet"
\[
\]
\[
\]
\[
h[7] = \text{enthalpy}(\text{Water}, T=T[7], P=P[7])
\]
\[
s[7] = \text{entropy}(\text{Water}, T=T[7], P=P[7])
\]
\[
\]
\[
E_.[7] = \text{convert}(\text{kW/MW}) \cdot e[7] \cdot m_\text{w}
\]

"Turbine 2 - High Pressure Turbine"
\[
\text{eta}_\text{turb}_2 = \frac{(h[7] - h[8])}{(h[7] - h_8_s)}
\]
\[
h_8_s = \text{enthalpy}(\text{Water}, P=P[8], s=s[7])
\]
\[
\text{eta}_\text{turb}_2 = 0.86
\]

"State Point 8 - Turbine 2 Outlet, Reheat 2 Inlet"
\[
T[8] = \text{temperature}(\text{Water}, P=P[8], h=h[8])
\]
\[
P[8] = 3204 \text{ [kPa]}
\]
\[
s[8] = \text{entropy}(\text{Water}, P=P[8], T=T[8])
\]
\[
e[8] = h[8] - h[0] - T[0]*(s[8] - s[0])
\]
\[
E_.[8] = \text{convert}(\text{kW/MW}) \cdot e[8] \cdot m_\text{w}
\]

"State Point 9 - Reheat 2 Outlet, Turbine 3 Inlet"
\[
\]
\[
P[9] = P[8]
\]
\[
h[9] = \text{enthalpy}(\text{Water}, T=T[9], P=P[9])
\]
\[
s[9] = \text{entropy}(\text{Water}, T=T[9], P=P[9])
\]
\[
e[9] = h[9] - h[0] - T[0]*(s[9] - s[0])
\]
\[
E_.[9] = \text{convert}(\text{kW/MW}) \cdot e[9] \cdot m_\text{w}
\]

"Turbine 3 - Medium High Pressure Turbine"
\[
\text{eta}_\text{turb}_3 = \frac{(h[9] - h[10])}{(h[9] - h_10_s)}
\]
\[
h_10_s = \text{enthalpy}(\text{Water}, P=P[10], s=s[9])
\]
\[
\text{eta}_\text{turb}_3 = 0.86
\]

"State Point 10 - Turbine 3 Outlet, Reheat 3 Inlet"
\[
T[10] = \text{temperature}(\text{Water}, P=P[10], h=h[10])
\]
\[
P[10] = 1408 \text{ [kPa]}
\]
\[
s[10] = \text{entropy}(\text{Water}, P=P[10], T=T[10])
\]
\[
e[10] = h[10] - h[0] - T[0]*(s[10] - s[0])
\]
\[
E_.[10] = \text{convert}(\text{kW/MW}) \cdot e[10] \cdot m_\text{w}
\]

"State Point 11 - Reheat 3 Outlet, Turbine 4 Inlet"
\[
\]
\[
\]
\[
\]
\[
\]
\[
\]
\[
E_.[11] = \text{convert}(\text{kW/MW}) \cdot e[11] \cdot m_\text{w}
\]
**Turbine 4 - Medium Low Pressure Turbine**

\[
\eta_{turb\_4} = \frac{h[11] - h[12]}{h[11] - h_{12\_s}} \\
h_{12\_s} = \text{enthalpy(Water, P=P[12], s=s[11])} \\
\eta_{turb\_4} = 0.86
\]

**State Point 12 - Turbine 4 Outlet, Reheat 4 Inlet**

\[
T[12] = \text{temperature(Water, P=P[12], h=h[12])} \\
P[12] = 610.7 \text{[kPa]} \\
s[12] = \text{entropy(Water, P=P[12], T=T[12])} \\
e[12] = h[12] - h[0] - T[0]*(s[12] - s[0]) \\
E\_\{12\} = \text{convert(kW, MW)*e[12]*}m\_w
\]

**Turbine 5 - Low Pressure Turbine**

\[
\eta_{turb\_5} = \frac{h[13] - h[14]}{h[13] - h_{14\_s}} \\
h_{14\_s} = \text{enthalpy(Water, P=P[14], s=s[13])} \\
\eta_{turb\_5} = 0.86
\]

**State Point 13 - Reheat 4 Outlet, Turbine 5 Inlet**

\[
h[13] = \text{enthalpy(Water, T=T[13], P=P[13])} \\
s[13] = \text{entropy(Water, T=T[13], P=P[13])} \\
E\_\{13\} = \text{convert(kW, MW)*e[13]*}m\_w
\]

**Turbine 6 - Lowest Pressure Turbine**

\[
\eta_{turb\_6} = \frac{h[15] - h[16]}{h[15] - h_{16\_s}} \\
h_{16\_s} = \text{enthalpy(Water, P=P[16], s=s[15])} \\
\eta_{turb\_6} = 0.86
\]

**State Point 14 - Reheat 5 Outlet, Turbine 6 Inlet**

\[
h[14] = \text{enthalpy(Water, T=T[14], P=P[15])} \\
s[14] = \text{entropy(Water, T=T[14], P=P[15])} \\
e[14] = h[14] - h[0] - T[0]*(s[14] - s[0]) \\
E\_\{14\} = \text{convert(kW, MW)*e[14]*}m\_w
\]

**Turbine Work**

\[
W\_\{turb\_1\} = (m\_w*(h[5] - h[6]))*\text{convert(kW, MW)} \\
W\_\{turb\_2\} = (m\_w*(h[7] - h[8]))*\text{convert(kW, MW)}
\]
\[
W_{\text{turb\_3}} = (m_\text{w}\cdot(h_{9} - h_{10}))\cdot\text{convert(kW,MW)}
\]
\[
W_{\text{turb\_4}} = (m_\text{w}\cdot(h_{11} - h_{12}))\cdot\text{convert(kW,MW)}
\]
\[
W_{\text{turb\_5}} = (m_\text{w}\cdot(h_{13} - h_{14}))\cdot\text{convert(kW,MW)}
\]
\[
W_{\text{turb\_6}} = (m_\text{w}\cdot(h_{15} - h_{16}))\cdot\text{convert(kW,MW)}
\]
\[
W_{\text{turb}} = W_{\text{turb\_1}} + W_{\text{turb\_2}} + W_{\text{turb\_3}} + W_{\text{turb\_4}} + W_{\text{turb\_5}} + W_{\text{turb\_6}}
\]

"Energy Calculations"

\[
Q_{\text{boiler}} = (m_\text{w}\cdot(h_{5} - h_{2})) + m_\text{w}\cdot(h_{7} - h_{6}) + m_\text{w}\cdot(h_{9} - h_{8}) + m_\text{w}\cdot(h_{11} - h_{10}) + m_\text{w}\cdot(h_{13} - h_{12}) + m_\text{w}\cdot(h_{15} - h_{14}))\cdot\text{convert(kW,MW)}
\]
\[
Q_{\text{cond}} = (m_\text{w}\cdot(h_{16} - h_{1}))\cdot\text{convert(kW,MW)}
\]
\[
W_{\text{pump}} = (m_\text{w}\cdot(h_{2} - h_{1}))\cdot\text{convert(kW,MW)}
\]

\[
W_{\text{out}} = 1000\text{[MW]}
\]
\[
W_{\text{out}} = W_{\text{turb}} - W_{\text{pump}}
\]

\[
\eta_{\text{th}} = W_{\text{out}}/Q_{\text{boiler}}
\]
"2 Regenerations"

"State Point 1 - Condenser outlet, Pump 1 inlet"
T[1] = temperature(Water, P=P[1], x=x[1])
P[1] = 10 [kPa]
h[1] = enthalpy(Water, P=P[1], x=x[1])
s[1] = entropy(Water, P=P[1], x=x[1])
x[1] = 0

"State Point 2 - Pump 1 outlet, Open FWH 1 inlet"
T[2] = temperature(Water, P=P[2], h=h[2])
P[2] = 500 [kPa]
s[2] = entropy(Water, P=P[2], h=h[2])

"State Point 3 - Open FWH 1 outlet, Pump 2 inlet"
T[3] = temperature(Water, P=P[2], h=h[3])
s[3] = entropy(Water, P=P[3], h=h[3])

"State Point 4 - Pump 2 outlet, Open FWH 2 inlet"
T[4] = temperature(Water, P=P[4], h=h[4])
P[4] = 4000 [kPa]
s[4] = entropy(Water, P=P[4], h=h[4])

"State Point 5 - Open FWH 2 outlet, Pump 3 inlet"
T[5] = temperature(Water, P=P[5], h=h[5])
s[5] = entropy(Water, P=P[5], h=h[5])

"State Point 6 - Pump 3 outlet, Boiler inlet"
T[6] = temperature(Water, P=P[6], h=h[6])
P[6] = 15000 [kPa]
s[6] = entropy(Water, P=P[6], h=h[6])

"State Point 7 - Saturated Liquid Within Boiler"
T[7] = temperature(Water, P=P[7], x=x[7])
h[7] = enthalpy(Water, P=P[7], x=x[7])
s[7] = entropy(Water, P=P[7], x=x[7])
x[7] = 0

"State Point 8 - Saturated Vapor Within Boiler"
T[8] = temperature(Water, P=P[8], x=x[8])
h[8] = enthalpy(Water, P=P[8], x=x[8])
s[8] = entropy(Water, P=P[8], x=x[8])
x[8] = 1

"State Point 9 - Boiler outlet, Turbine 1 inlet"
T[9] = converttemp(C,K,600)
h[9] = enthalpy(Water, T=T[9], P=P[9])
s[9] = entropy(Water, T=T[9], P=P[9])

"State Point 10 - Turbine 1 outlet, Turbine 2 inlet, Open FWH 2 inlet"
T[10] = temperature(Water, P=P[10], h=h[10])
s[10] = entropy(Water, P=P[10], h=h[10])

"!State Point 11 - Turbine 2 outlet, Turbine 3 inlet, Open FWH 1 inlet"

"!State Point 12 - Turbine 3 outlet, Condenser inlet"
T[12] = temperature(Water, P=P[12], h=h[12])
P[12] = P[1]
s[12] = entropy(Water, P=P[12], h=h[12])

"!Control Volumes"
"Pump 1"
eta_Pump_1 = 0.86
h_2_s = enthalpy(Water, P=P[2], s=s[1])
eta_Pump_1 = (h_2_s - h[1]) / (h[2] - h[1])
W_.Pump_1 = m_.3*(h[2] - h[1])*convert(kW,MW)

"Open FWH 1"

"Pump 2"
eta_Pump_2 = 0.86
h_4_s = enthalpy(Water, P=P[4], s=s[3])
eta_Pump_2 = (h_4_s - h[3]) / (h[4] - h[3])
W_.Pump_2 = m_.2*(h[4] - h[3])*convert(kW,MW)

"Open FWH 2"

"Pump 3"
eta_Pump_3 = 0.86
h_6_s = enthalpy(Water, P=P[6], s=s[5])
eta_Pump_3 = (h_6_s - h[5]) / (h[6] - h[5])
W_.Pump_3 = m_.1*(h[6] - h[5])*convert(kW,MW)

"Boiler"
Q_.Boiler = m_.1*(h[9] - h[6])*convert(kW,MW)

"Turbine 1"
eta_Turb = 0.86
h_10_s = enthalpy(Water, P=P[10], s=s[9])
eta_Turb = (h[9] - h[10]) / (h[9] - h_10_s)
W_.Turb_1 = m_.1*(h[9] - h[10])*convert(kW,MW)

"Split 1"
m_.1 = m_.2 + m_.4
m_.2 = m_.1*(1-y)
m_.4 = m_.1*(y)

"Turbine 2"
h_11_s = enthalpy(Water, P=P[11], s=s[10])
W_.Turb_2 = m_.2*(h[10] - h[11])*convert(kW,MW)
"Split 2"
m_.2 = m_.3 + m_.5
m_.3 = m_.1*(1-y-z)
m_.5 = m_.1*(z)

"Turbine 3"
h_12_s = enthalpy(Water, P=P[12], s=s[11])

"Condenser"
Q_.Condenser = m_.3*(h[12] - h[1])*convert(kW,MW)

W_.out = 1000 [MW]
W_.out = W_.Turb - W_.Pump
W_.Turb = W_.Turb_1 + W_.Turb_2 + W_.Turb_3
W_.Pump = W_.Pump_1 + W_.Pump_2 + W_.Pump_3

eta_th = W_.out / Q_.Boiler
### 15.3 Appendix 3: Specification Sheets of Components

#### Boiler

<table>
<thead>
<tr>
<th>Identification:</th>
<th>Item Name:</th>
<th>Date:</th>
<th>Item Number:</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Boiler</td>
<td>4/18/2012</td>
<td>B-100W</td>
</tr>
</tbody>
</table>

**Function:** Boils water to be used in turbines

**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-006</td>
<td>610.6</td>
<td>610.6</td>
</tr>
<tr>
<td>S-007</td>
<td>610.6</td>
<td>610.6</td>
</tr>
<tr>
<td>S-008</td>
<td>610.6</td>
<td>610.6</td>
</tr>
<tr>
<td>S-009</td>
<td>610.6</td>
<td>610.6</td>
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</table>

**Composition:**

<table>
<thead>
<tr>
<th>Component</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>H$_2$O$_l$</td>
<td>610.6</td>
<td>610.6</td>
</tr>
<tr>
<td>H$_2$O$_v$</td>
<td>610.6</td>
<td>610.6</td>
</tr>
<tr>
<td>Air</td>
<td>1284</td>
<td></td>
</tr>
<tr>
<td>O$_2$</td>
<td>268.4</td>
<td></td>
</tr>
<tr>
<td>H</td>
<td>3.9</td>
<td></td>
</tr>
<tr>
<td>C</td>
<td>71.9</td>
<td></td>
</tr>
<tr>
<td>N$_2$</td>
<td>939.9</td>
<td></td>
</tr>
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</table>

**MATERIAL**

<table>
<thead>
<tr>
<th>Component</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>H$_2$O</td>
<td>38.5</td>
</tr>
<tr>
<td>CO$_2$</td>
<td>261.9</td>
</tr>
<tr>
<td>O$_2$</td>
<td>44.9</td>
</tr>
<tr>
<td>N$_2$</td>
<td>939.9</td>
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**Temperature (K):**

<table>
<thead>
<tr>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>474.4</td>
</tr>
<tr>
<td>615.3</td>
</tr>
<tr>
<td>615.3</td>
</tr>
<tr>
<td>873.2</td>
</tr>
</tbody>
</table>

**Pressure (kPa):**

<table>
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<tr>
<th>Value</th>
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<tbody>
<tr>
<td>15000</td>
</tr>
<tr>
<td>15000</td>
</tr>
<tr>
<td>15000</td>
</tr>
<tr>
<td>15000</td>
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**Design Data:**

<table>
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<tr>
<th>Type:</th>
<th>PC Tangentially Fired</th>
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<tr>
<td>Materials for Construction:</td>
<td><strong>tbd</strong></td>
</tr>
<tr>
<td>Temperature Change:</td>
<td>398.8 K</td>
</tr>
<tr>
<td>Heating Duty:</td>
<td>2266 MW</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.86</td>
</tr>
</tbody>
</table>

**Comments:** B-100W boils water in between M-101W and T-100W. The unit is a pulverized coal tangentially fired boiler that utilizes dual registry burners.
<table>
<thead>
<tr>
<th>Identification:</th>
<th>Item Name:</th>
<th>Pump</th>
<th>Date:</th>
<th>4/18/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>P-100W</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Function:** To pump from C-100W to HE-100W

**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>S-001</td>
<td>S-002</td>
</tr>
<tr>
<td>Quantity (kg/s)</td>
<td>474.6</td>
<td>474.6</td>
</tr>
<tr>
<td>Composition:</td>
<td>H₂O₁</td>
<td>474.6</td>
</tr>
</tbody>
</table>

| Temperature (K): | 318.9 | 320 |
| Pressure (kPa):  | 10    | 15000 |

**Design Data:**

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
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<tbody>
<tr>
<td>Mass Flow Rate:</td>
<td>470 kg/s</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>14990 kPa</td>
</tr>
<tr>
<td>Power Required:</td>
<td>8.3 MW</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.8</td>
</tr>
</tbody>
</table>

**Comments:** P-100W is a pump that increases water pressure to 15000kPa as it leaves C-100W.
### Identification

<table>
<thead>
<tr>
<th>Item Name:</th>
<th>Pump</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>P-101W</td>
</tr>
</tbody>
</table>

**Date:** 5/9/2012

### Function:
To pump from HE-100W to M-100W

### Operation:
Continuous

### Materials Handled:

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-015</td>
<td></td>
<td></td>
</tr>
<tr>
<td>S-016</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Quantity (kg/s)</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>129.9</td>
<td></td>
<td></td>
</tr>
<tr>
<td>129.9</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Composition:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₂O l</td>
<td>129.9</td>
<td>129.9</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temperature (K):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>414.9</td>
<td></td>
<td></td>
</tr>
<tr>
<td>417</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure (kPa):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>380</td>
<td></td>
<td></td>
</tr>
<tr>
<td>15000</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

### Design Data:

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass Flow Rate:</td>
<td>6.2 kg/s</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>14620 kPa</td>
</tr>
<tr>
<td>Power Required:</td>
<td>2.379 MW</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.8</td>
</tr>
</tbody>
</table>

### Comments:
P-101W is a pump that increases water pressure to 15000kPa as it leaves HE-100W and enters M-100W.
**Pump & Motor**

**Identification:** Item Name: Pump  
**Item Number:** P-102W  
**Date:** 4/18/2012

**Function:** To pump from HE-101W to M-101W  
**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-017</td>
<td>6.2</td>
<td>6.2</td>
</tr>
<tr>
<td>S-018</td>
<td>6.2</td>
<td>6.2</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Composition:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\text{H}_2\text{O}_l$</td>
<td>6.2</td>
<td>6.2</td>
</tr>
</tbody>
</table>

| Temperature (K): | 503.4 | 506.5 |
| Pressure (kPa):  | 2806  | 15000 |

**Design Data:**

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Flow Rate:</td>
<td>$\text{m}^3/\text{hr}$</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>14990 kPa</td>
</tr>
<tr>
<td>Power Required:</td>
<td>8.331 MW</td>
</tr>
<tr>
<td>Efficiencys:</td>
<td>0.8</td>
</tr>
</tbody>
</table>

**Comments:** P-102W is a pump that increases water pressure to 15000kPa as it leaves HE-101W and enters M-101W.
**Turbine**

<table>
<thead>
<tr>
<th>Identification: Item Name:</th>
<th>Turbine</th>
<th>Date:</th>
<th>4/18/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>T-100W</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Function:** Expander  

**Operation:** Continuous  

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-009</td>
<td></td>
<td>S-010</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Quantity (kg/s)</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>610.6</td>
<td></td>
<td>610.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Composition:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H_2O_i$</td>
<td>610.6</td>
<td></td>
</tr>
<tr>
<td>$H_2O_v$</td>
<td></td>
<td>610.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temperature (K):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>873.2</td>
<td></td>
<td>628.1</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure (kPa):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>15000</td>
<td></td>
<td>2806</td>
</tr>
</tbody>
</table>

**Design Data:**

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Flow Rate:</td>
<td>$m^3/hr$</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>-12194 kPa</td>
</tr>
<tr>
<td>Power Produced:</td>
<td>275.3 MW</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.86</td>
</tr>
</tbody>
</table>

**Comments:** T-100W is the first stage of a three stage turbine which reduces the pressure to 2806 kPa as it leaves B-100W and enters V-100W.
<table>
<thead>
<tr>
<th>Identification:</th>
<th>Item Name:</th>
<th>Turbine</th>
<th>Date:</th>
<th>4/18/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>T-101W</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Function:** Expander  
**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-011</td>
<td>604.5</td>
<td></td>
</tr>
<tr>
<td>S-012</td>
<td>604.5</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Quantity (kg/s)</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>H₂Oᵢᵥ</td>
<td>604.5</td>
<td>604.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temperature (K):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>873.2</td>
<td></td>
<td>603.5</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure (kPa):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>2806</td>
<td></td>
<td>380</td>
</tr>
</tbody>
</table>

**Design Data:**

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Flow Rate:</td>
<td>m³/hr</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>-2426 kPa</td>
</tr>
<tr>
<td>Power Produced:</td>
<td>335.2 MW</td>
</tr>
<tr>
<td>Efficiencies</td>
<td>0.86</td>
</tr>
</tbody>
</table>

**Comments:** T-101W is the second stage of a three stage turbine which reduces the pressure to 380 kPa as it leaves HE-100A and enters V-101W.
**Turbine**

<table>
<thead>
<tr>
<th>Identification:</th>
<th>Item Name:</th>
<th>Date:</th>
<th>4/18/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>Turbine</td>
<td></td>
<td>T-102W</td>
</tr>
</tbody>
</table>

**Function:** Expander  
**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-013</td>
<td>474.6</td>
<td>474.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Quantity (kg/s)</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>474.6</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Composition:</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>$H_2O_{i,v}$</td>
<td>474.6</td>
<td>474.6</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Temperature (K):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>873.2</td>
<td></td>
<td>380452.9</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Pressure (kPa):</th>
<th>Inlet</th>
<th>Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>380</td>
<td></td>
<td>10</td>
</tr>
</tbody>
</table>

**Design Data:**

<table>
<thead>
<tr>
<th>Type:</th>
<th>Centrifugal</th>
</tr>
</thead>
<tbody>
<tr>
<td>Volumetric Flow Rate:</td>
<td>m$^3$/hr</td>
</tr>
<tr>
<td>Pressure Change:</td>
<td>-370 kPa</td>
</tr>
<tr>
<td>Power Produced:</td>
<td>409.6 MW</td>
</tr>
<tr>
<td>Efficiency:</td>
<td>0.86</td>
</tr>
</tbody>
</table>

**Comments:** T-102W is the third stage of a three stage turbine which reduces the pressure to 10 kPa as it leaves HE-101A and enters C-100W.
**Condenser**

**Identification:** Heat Exchanger  
**Item Name:** Condenser  
**Item Number:** E-000  
**Date:** 5/1/2012

**Function:** To Condense steam off turbine T-002  
**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Cold in</th>
<th>Hot In</th>
<th>Hot Out</th>
<th>Cold Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-025</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>S-014</td>
<td>1.592*10^8</td>
<td>1706000</td>
<td>1706000</td>
<td>1.592*10^8</td>
</tr>
<tr>
<td>S-001</td>
<td>1</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
<tr>
<td>S-026</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Composition:** H2O

**Temperature (°C):**  
- $T_{\text{Hot In}}$: 47  
- $T_{\text{Hot Out}}$: 46.0  
- $T_{\text{Cold In}}$: 27.0  
- $T_{\text{Cold Out}}$: 33.0

**Pressure (kPa):** 10

**Design Data:**

- **Type:** Fixed Head Shell and Tube  
- **Materials of Construction:** Carbon Steel Shell/Carbon Steel Tube  
- **Heat Duty:** 1140 MW  
- **Shell Diameter:** 3.34 m  
- **Heat Transfer Coefficient:** 929.9 W/m^2-K  
- **Heat Transfer Area:** 19450 m^2

**Comments:** E-000 is used to condense steam off turbine T-002, in order for it to be pumped back to the boiler.
# Heat Exchanger

**Identification:** Heat Exchanger

**Item Name:** Heat Exchanger

**Item Number:** E-001

**Date:** 5/1/2012

**Function:** To regenerate steam off turbine T-001

**Operation:** Continuous

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID</th>
<th>Cold in</th>
<th>Hot In</th>
<th>Hot Out</th>
<th>Cold Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>S-025</td>
<td>1706000</td>
<td>475900</td>
<td>475900</td>
<td>1706000</td>
</tr>
<tr>
<td>S-014</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>S-001</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>S-026</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Composition:** H2O

<table>
<thead>
<tr>
<th>temperature (°C)</th>
<th>Hot In</th>
<th>Hot Out</th>
<th>Cold In</th>
<th>Cold Out</th>
</tr>
</thead>
<tbody>
<tr>
<td>T&lt;sub&gt;Hot In&lt;/sub&gt;</td>
<td>329.6</td>
<td>141.8</td>
<td>46.9</td>
<td>205.1</td>
</tr>
<tr>
<td>T&lt;sub&gt;Hot Out&lt;/sub&gt;</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T&lt;sub&gt;Cold In&lt;/sub&gt;</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>T&lt;sub&gt;Cold Out&lt;/sub&gt;</td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

**Pressure (kpa):** 380

**Design Data:**

- **Type:** Fixed Head Shell and Tube
- **Materials of Construction:** Carbon Steel Shell/Carbon Steel Tube
- **Heat Duty:** 335 MW
- **Shell Diameter:** 1.03 m
- **Heat Transfer Coefficient:** 1315 W/m^2-K
- **Heat Transfer Area:** 2685 m^2

**Comments:** E-001 is used to condense steam off turbine T-001, as well as preheat the water entering the boiler.
# Heat Exchanger

<table>
<thead>
<tr>
<th>Identification:</th>
<th>Item Name:</th>
<th>Heat Exchanger</th>
<th>Date:</th>
<th>5/1/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td></td>
<td>E-002</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Function:</th>
<th>To regenerate steam off turbine T-000</th>
</tr>
</thead>
<tbody>
<tr>
<td>Operation:</td>
<td>Continuous</td>
</tr>
</tbody>
</table>

**Materials Handled:**

<table>
<thead>
<tr>
<th>Stream ID:</th>
<th>Cold in</th>
<th>Hot In</th>
<th>Hot Out</th>
<th>Cold Out</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>S-025</td>
<td>S-014</td>
<td>S-001</td>
<td>S-026</td>
</tr>
<tr>
<td>Quantity (kg/hr)</td>
<td>2182000</td>
<td>57780</td>
<td>57780</td>
<td>2182000</td>
</tr>
<tr>
<td>Composition:</td>
<td>H2O</td>
<td>1</td>
<td>1</td>
<td>1</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>$T_{\text{Hot In}}$ (C):</th>
<th>346.7</th>
</tr>
</thead>
<tbody>
<tr>
<td>$T_{\text{Hot Out}}$ (C):</td>
<td>230.8</td>
</tr>
<tr>
<td>$T_{\text{Cold In}}$ (C):</td>
<td>192.1</td>
</tr>
<tr>
<td>$T_{\text{Cold Out}}$ (C):</td>
<td>203.7</td>
</tr>
</tbody>
</table>

| Pressure (kPa): | 2806 |

<table>
<thead>
<tr>
<th>Design Data:</th>
<th>Fixed Head Shell and Tube</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type:</td>
<td>Carbon Steel Shell/Carbon Steel</td>
</tr>
<tr>
<td>Materials of Construction:</td>
<td>Tube</td>
</tr>
<tr>
<td>Heat Duty:</td>
<td>34 MW</td>
</tr>
<tr>
<td>Shell Diameter:</td>
<td>1.03 m</td>
</tr>
<tr>
<td>Heat Transfer Coefficient:</td>
<td>935.3 W/m²·K</td>
</tr>
<tr>
<td>Heat Transfer Area:</td>
<td>473 m²</td>
</tr>
</tbody>
</table>

**Comments:** E-002 is used to condense steam off turbine T-000, as well as preheat the water entering the boiler.
# Fly Ash Removal

<table>
<thead>
<tr>
<th>Identification</th>
<th>Item Name: Electrostatic Precipitator</th>
<th>Date: 4/20/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>FA-100</td>
<td></td>
</tr>
</tbody>
</table>

**Function:** To precipitate out fly ash from the gasifier outlet  
**Operation:** Continuous  
**Materials Handled:**

<table>
<thead>
<tr>
<th>Inlet</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Stream ID:</td>
<td>B100</td>
<td></td>
</tr>
<tr>
<td>Quantity (lb/hr)</td>
<td>9.22E+06</td>
<td></td>
</tr>
<tr>
<td>Composition fly ash (lb/hr):</td>
<td>7.96E+04</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Outlet</th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Stream ID:</td>
<td>B101</td>
<td></td>
</tr>
<tr>
<td>Quantity (lb/hr)</td>
<td>9.14E+06</td>
<td></td>
</tr>
<tr>
<td>Composition (lb/hr):</td>
<td>3.98E+03</td>
<td></td>
</tr>
</tbody>
</table>

**Temp (F):** 2273  
**Pressure(psi):** 20  
**Design Data:**

- **Materials of Construction:** Carbon Steel  
- **Collector Plates:** 40 ft  
- **Plate Height:** 13 ft  
- **Plate Width:** 11 ft  
- **Plant Dimensions (WxLxH):** 39x45x30 ft  
- **Plate Spacing:** 2 ft  
- **Drift Velocity:** 1.3 ft  
- **Operating Voltage:** 40 kV  
- **Operating Power Draw:** 20.7 kW

**Comments:** The Electrostatic Precipitator attracts and collects particulate matter from the flue gas using an electrical current. The particulate is removed through the bottom of the plant through a process called rapping, which shakes the collected dust free.
## Fly Ash Removal

<table>
<thead>
<tr>
<th>Identification</th>
<th>Item Name: Electrostatic Precipitator</th>
<th>Date: 5/8/2012</th>
</tr>
</thead>
<tbody>
<tr>
<td>Item Number:</td>
<td>FA-100</td>
<td></td>
</tr>
</tbody>
</table>

### Function:
To precipitate out fly ash from the Boiler outlet

### Operation:
Continuous

### Materials Handled:

<table>
<thead>
<tr>
<th>Stream ID</th>
<th>Quantity (lb/hr)</th>
<th>Composition fly ash (lb/hr):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Inlet</td>
<td>4.92E+06</td>
<td>5.48E+04</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Stream ID</th>
<th>Quantity (lb/hr)</th>
<th>Composition (lb/hr):</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outlet</td>
<td>4.87E+06</td>
<td>1.64E+03</td>
</tr>
</tbody>
</table>

| Temp (F):   | 1607             |
| Pressure (psi): | 14.5             |

### Design Data:

| Materials of Construction: Carbon Steel |
| Collector Plates: 18 |
| Plate Height: 13 ft |
| Plate Width: 11 ft |
| Plant Dimensions (WxLxH): 17x45x30 ft |
| Plate Spacing: 2 ft |
| Drift Velocity: 1.3 ft |
| Operating Voltage: 40 kW |
| Operating Power Draw: 8.7 kW |

### Comments:
The Electrostatic Precipitator attracts and collects particulate matter from the exhaust gas using an electrical current. The particulate is removed through the bottom of the plant through a process called rapping, which shakes the collected dust free.
Appendix 4: Heat Exchanger Calculations

**Shell and Tube Heat Exchanger Design**

*Heat Exchanger Specifications - Based on Table 18.6*

One Pass - Triangular Pitch
Shell Diameter - 37 inches (0.9398 m)
Tube Diameter - 3/4 inches (0.0191 m)
Triangle Pitch - 1 inch (0.0254 m)
Number of Tubes - 1074

Number of Tubes: \( n = \frac{7500}{1074} \approx 7.01 \)

**Tube Side Properties**

- \( D_t = 0.03048 \) (Tube Inside Diameter (m))
- \( k = 0.62 \) (Thermal Conductivity of Tube Side Fluid (W/m·K))
- \( C_p = 4318 \) (Mass Heat Capacity of Tube Side Fluid (J/kg·K))
- \( F = \frac{11057}{14} = 791.29 \) (Mass Flow Rate of 1 Tube (kg/s))

\[ F = 1.474 \]

\[ \Lambda = \pi \left( \frac{D_t}{2} \right)^2 \]

\[ A = 7.297 \times 10^{-4} \text{ m}^2 \text{ross Sectional Area for Flow} \]

\[ G = F \frac{A}{\Lambda} \]

\[ \mu = 7.972 \times 10^{-4} \text{ kg/m·s} \text{Viscosity} \]

\[ n = 0.4 \]

\[ h_t = 0.023 \frac{k}{D_t} \left( \frac{D_t G}{\mu} \right)^{0.8} \left( \frac{C_p \mu}{k} \right)^{n} \]

\[ h_t = 7.555 \times 10^{3} \text{ W/m}^2\text{K} \text{Sieder-Tate Equation (18.13)} \]

**Shell Side Properties**

- \( D_o = 0.0381 \) (Tube Outside Diameter (m))
- \( D_s = 3.34 \) (Shell Diameter (m))
- \( k = 333 \) (Thermal Conductivity of Shell Side Fluid (W/m·K))
- \( C_p = 3127 \) (Mass Heat Capacity of Fluid (J/kg·K))
- \( F = 1185 \) (Mass Flow Rate of Fluid (kg/s))
Cross Sectional Area for Flow (m²2)

A = \pi \left( \frac{D_s}{2} \right)^2 - \left( \frac{D_s}{2} \right) \left[ \pi \left( \frac{D_o}{2} \right)^2 \right] \\

Fluid Mass Velocit (kg/m²-s)

G = \frac{F}{A} \\

\mu = 2.99 \times 10^{-4} \\

A = 0.211 \\

Viscosity (Pa-s)

G = 561.88 \\

Kern Correlation Constants

C = 0.36 \\

n = 0.55 \\

K = 0.333 \\

Kern Correlation (18.21)

h_o = C \left( \frac{D_o \mu}{k} \right)^{n \left( \frac{C - \mu}{k} \right)^{0.333}} \\

Convective Heat Transfer Coefficient (W/m²)

Inlet and Outlet Temperatures

<table>
<thead>
<tr>
<th>Side</th>
<th>Temperature (°C)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hot Side</td>
<td>T_{hi} = 47</td>
</tr>
<tr>
<td></td>
<td>T_{ho} = 46.1</td>
</tr>
<tr>
<td>Cold Side</td>
<td>T_{ci} = 27</td>
</tr>
<tr>
<td></td>
<td>T_{co} = 33</td>
</tr>
</tbody>
</table>

\[ T_{lm} = \frac{(T_{hi} - T_{co}) - (T_{ho} - T_{ci})}{\ln \left( \frac{T_{hi} - T_{co}}{T_{ho} - T_{ci}} \right)} \]

\[ T_{lm} = 16.418 \]

Materials of Construction

Shell Side - Carbon Steel
Tube Side - Carbon Steel

k_s = 19.76 \\

L = 20.64 \\

Thermal Conductivity of Tubes (W/m-K)

Tube length (m)

Overall Heat Transfer Coefficient and Fm Correction Factor

\[ A_i = \pi \cdot D_i \cdot L \]
\[ A_o = \pi \cdot D_o \cdot L \]
\[ A_m = \frac{\pi \cdot L \cdot (D_o - D_i)}{\ln \left( \frac{D_o}{D_i} \right)} \]
\[ t_w = D_o - D_i \]
\[ A_i = 1.976 \]
\[ A_o = 2.47 \]
\[ U := \frac{1}{\frac{1}{h_0} + \frac{t_{w\cdot A_o}}{k\cdot A_m} + \frac{A_o}{h_1\cdot A_i}} \]

\[ U = 929.896 \quad \text{Heat Transfer Coefficient (W/m}^2\text{-K)} \]

\[ R := \frac{T_{hi} - T_{ho}}{T_{co} - T_{ci}} \quad S := \frac{T_{co} - T_{ci}}{T_{hi} - T_{ci}} \]

\[ R = 0.15 \quad S = 0.3 \]

From the \( R \) and \( S \) Values, determine an appropriate shell pass configuration using graphs in Seider on page 485.

\[ F_T := .96 \quad \text{Ft Correction Factor for 2-4 Shell Pass Heat Exchanger} \]

**Heat Transfer Area Needed**

\[ A := \frac{Q}{U\cdot F_T\cdot T_{im}} \]

\[ A = 1.945 \times 10^4 \quad \text{Heat Exchange Area Needed (m}^2\text{)} \]

\[ A_{R} := A \cdot 10.76 = 2.092 \times 10^5 \quad \text{Heat Exchange Area Needed (ft}^2\text{)} \]

\[ L := \frac{A}{3.14\cdot T:.034} \quad L = 24.285 \]
15.5 Appendix 5: Process Flow Diagram
15.6 Appendix 6: Gantt Chart

Figure 27: Gantt Chart First Semester
Figure 28: Gantt Chart Second Semester