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Low Temperature Heat and Water Recovery from Supercritical Coal Plant Flue Gas

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Low Temperature Heat and Water Recovery from Supercritical Coal Plant Flue Gas

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This thesis is dedicated my friends and family who have been a constant source of love and encouragement — my parents, Joe and Laurie, who have always been my biggest fans, my brothers, Erich and Christopher, my current and former roommates, John and Sean, all the rowdy fellows in my fantasy football league, and my beautiful girlfriend Erin, without whom the last two years would not have been nearly as much fun.

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Low Temperature Heat and Water Recovery from Supercritical Coal Plant Flue Gas

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The University of Texas at Austin, 2015

Supervisor: Michael E. Webber

For this work, I constructed an original thermodynamic model to estimate waste heat and water recovery from the flue gas of a supercritical coal plant burning lignite, subbituminous, or bittuminous coal. This model was written in MATLAB as a list of linear equations based on first and second law analyses of the power plant components. This research is relevant because coal accounted for the largest increase in primary energy consumption worldwide as recently as 2013 [1]. Coal-fired electricity generation is particularly water intensive. As populations increase, especially in the developing world, much of the increased demand for electricity will be provided by new coal-fired power plants [2].

One way to improve the efficiency of a coal-fired power plant is to recover the low temperature waste heat from the flue gas and use it to preheat combustion air or boiler feedwater. A low temperature economizer or flue gas cooler can be used for this purpose to achieve overall efficiency improvements as high as 0.4%. However, a side effect of the efficiency improvements is an increase in water consumption factor of nearly 10%.

The water consumption factor can be reduced with the addition of a flue gas dryer after the flue gas cooler. The flue gas dryer is a condensing heat exchanger

between the flue gas and ambient air. As the flue gas cools, its water content condenses and can be recovered and treated for use within the plant.

In general, the results indicate that low temperature waste heat and water recover from boiler flue gas would be more feasible and beneficial for coal plants burning lignite as opposed to higher quality coal. Because these plants already have a lower efficiency, the relative increase in efficiency is somewhat higher. Similarly, the relative increase in water consumption factor is somewhat lower for a lignite plant. The high moisture content and dew point of the flue gas produced from lignite combustion makes it easier to recover water with a flue gas dryer. The higher water recovery factor along with the lower water consumption factor means that a greater percentage of the water evaporated in the cooling tower can be recovered in the flue gas dryer of a lignite plant than for a plant burning higher quality coal.

Table of Contents

List of Tables

List of Figures

Chapter 1

Introduction

For this work, I constructed an original thermodynamic model to estimate waste heat and water recovery from the flue gas of a supercritical coal plant burning lignite, subbituminous, or bituminous coal. This topic is important because the majority of global electricity demand continues to be met by thermoelectric generation [3]. In the U.S., thermoelectric power generation accounts for nearly half of all water withdrawals [4]. Coal-fired electricity generation is particularly water intensive. As populations increase, especially in the developing world, much of the increased demand for electricity will be provided by new coal-fired power plants [2]. As recently as 2013, coal accounted for the largest increase in primary energy consumption worldwide [1].

Fuel costs and environmental regulations already incentivize power generators to have a high fuel efficiency. At the same time, worldwide demand for water for domestic use, manufacturing, and electricity production will continue to increase, particularly in the developing world as shown in Figure 1.1 [5]. The higher demand for water will incentivize power generators to be as efficient as possible in their water use.

Typically, heat is already extracted from the flue gas until it has cooled to a few degrees above the acid dew point, around 150 °C, beyond which the acids in the flue gas start to condense. Acid condensation results in corrosion and scale buildup, the latter of which can obstruct the flow of flue gas through the exhaust, increasing

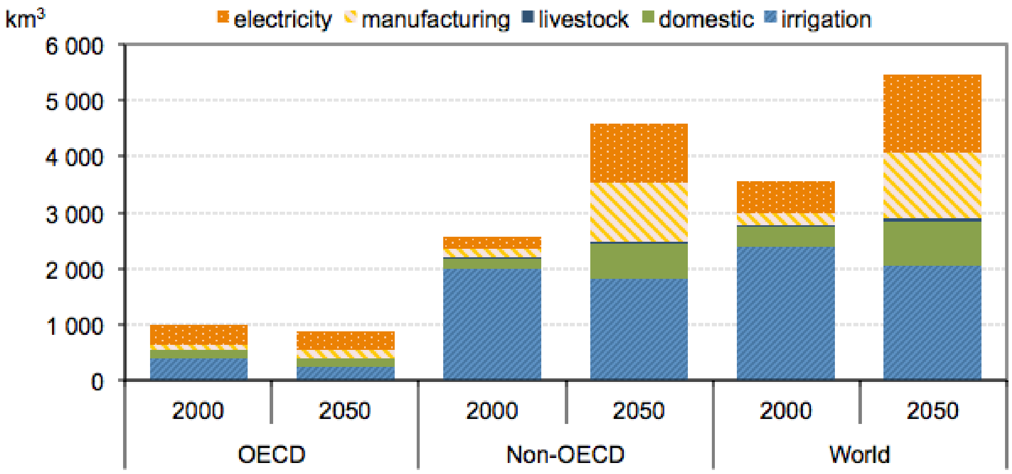

Figure 1.1: Global water demand is expected to increase dramatically between 2000 and 2050, particularly in the developing world [Source: OECD]

the back-pressure on the exhaust fan and decreasing plant efficiency. However, efforts to reduce fuel costs and carbon emissions might drive power generators to invest in the corrosion resistant heat exchangers necessary for recovering low-temperature waste heat from coal plant flue gas.

Low-temperature waste heat can be utilized for either preheating combustion air or boiler feedwater. Pre-heating combustion air has the effect of improving the boiler efficiency. Pre-heating boiler feedwater has the effect of increasing the temperature at which heat is added and, thus, increasing the thermodynamic efficiency of the cycle. Heat for feedwater heaters is typically extracted from bleed-off from the steam turbine[1]. By extracting low-temperature waste heat from the flue gas to preheat feedwater, the steam that would have been bled off can instead perform expansion work in a turbine, increasing plant power output and efficiency.

[1] A steam turbine for a modern Rankine Cycle power plant consists of several individual steam turbines mounted on a single shaft. The most common configuration consists of a single high pressure turbine, a single intermediate pressure turbine and two or more low pressure turbines.

The down-side of the waste heat recovered from the flue gas is that most of it ends up in the condenser. Typically, less than half of the waste heat recovered from the flue gas is converted to additional power output. The rest, which would have been rejected to the environment through the flue, is diverted to the condenser. As with all of the waste heat rejected to the condenser, it is then rejected to the cooling tower and then to the environment through evaporation. However, waste heat recovery from the flue gas can lead to reduced net water consumption if the moisture in the flue gas is also recovered. By extracting low-temperature waste heat from the flue gas, the flue gas is cooled closer to or below its dew point, allowing for the possibility of drying it further by cooling the flue gas in a heat exchanger. Using ambient air as the cold side of the heat exchanger by which heat from the flue gas is rejected to the environment, any water that condenses out can be collected, treated, and used as boiler feedwater, recirculating water for the cooling tower, or in a pollution control system, e.g. ash handling system or flue gas scrubber[2].

The analysis in this report is based on an original thermodynamic model of low temperature waste heat and water recovery for a super-critical coal plant with a capacity of approximately 600MW featuring multiple feedwater heaters, a regenerative air preheater and a steam-air heater. Baseline parameters are used to estimate the efficiency and water consumption for such a plant using lignite, subbituminous, or bituminous coal as a fuel. Three scenarios for waste heat recovery are then considered. In scenario A, the steam-air heater is replaced with a flue gas cooler. In scenario B, the lowest temperature feedwater heater is replaced with a flue gas cooler. In scenario C, the two lowest temperature feedwater heaters are both replaced with a flue gas

[2]Note that the cost of water treatment relates to both the quality of the water recovered and the process for which it is going to be used. Scrubber makeup water doesn't require much treatment as recirculating water. Feedwater must be extremely pure.

cooler.

For each of the scenarios and fuel types, the efficiency, power output, condenser cooling load, and water consumption factor are calculated using original thermodynamic models. Each of these factors is expected to increase with each successive scenario. The minimum heat exchanger effectiveness required for the flue gas cooler for each scenario and fuel type is also calculated as a proxy for the cost of waste heat recovery. It is expected that the minimum heat exchanger effectiveness will increase for each successive scenario. The minimum effectiveness will be lower, however, for the plants using lower quality coal because the higher moisture content results in an exhaust gas that transfers more heat for a given temperature gradient.

After the flue gas cooler, the exhaust goes through a flue gas dryer. The extent to which moisture is recovered is calculated according to the effectiveness of the flue gas dryer. As a result, metrics such as the rate of water recovery, the ratio of water recovery to water consumption, and the net water consumption rate can be quantified as functions of the effectiveness of the flue gas dryer.

These results can be used to assess feasibility and value of incorporating different schemes for low temperature waste heat and water recovery into the design of super-critical coal plants burning lignite, subbituminous or bituminous coal.

Chapter 2

Background

2.1 Coal Plant Energy Flows

Coal plants operate on a Rankine power cycle. The combustion of coal converts the chemical energy stored in the coal into heat. That heat is transferred to the steam cycle through a boiler with the intention of converting the thermal energy into mechanical energy with the hot steam driving a series of turbines. The turbines spin a shaft that drives a generator, converting the mechanical energy into electrical energy. The remaining heat in the steam is then rejected to the condenser at low pressure, and then the condensed feedwater is pumped back into the boiler. These processes are illustrated in Figure 2.1.

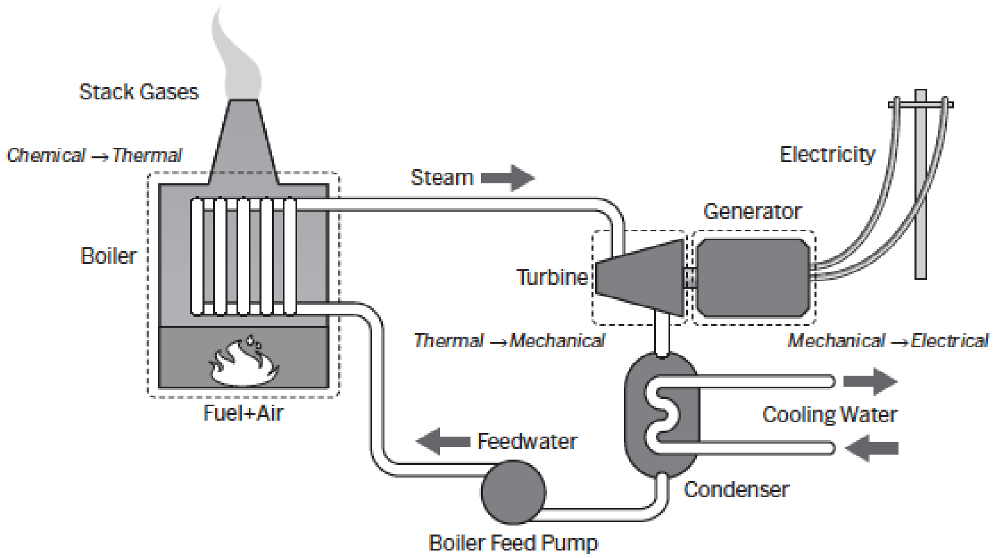

Figure 2.1: For a Rankine cycle power plant, chemical energy in the fuel is converted to thermal energy by combustion and transferred to a boiler. A steam turbine converts the thermal energy to mechanical energy, and a generator converts the mechanical energy to electrical energy. [Source: Philips, J]

In addition to the basic components shown in Figure 2.1, most coal plants also have one or more feedwater heaters. Feedwater heaters are used to increase the temperature of the feedwater on its way to the boiler, increasing boiler efficiency and reducing the potential for "thermal shock," wherein cracks form on the surface of the boiler as the result of thermal stress. The heat for the feedwater heater is extracted from steam "bleed-off" from one of the turbines. In an open feedwater heater, the turbine bleed-off is mixed directly with the feedwater as shown in Figure 2.2. Open feedwater heaters are also designed to act as deaerators that remove non-condensible gases from the feedwater. Some of these constituents, such as oxygen, can oxidize on metal surfaces if not removed, forming rust and obstructing the flow of feedwater through the system. In a closed feedwater heater, the bleed-off goes through a heat exchanger and is cooled until it condenses as a saturated fluid as shown in Figure 2.3.

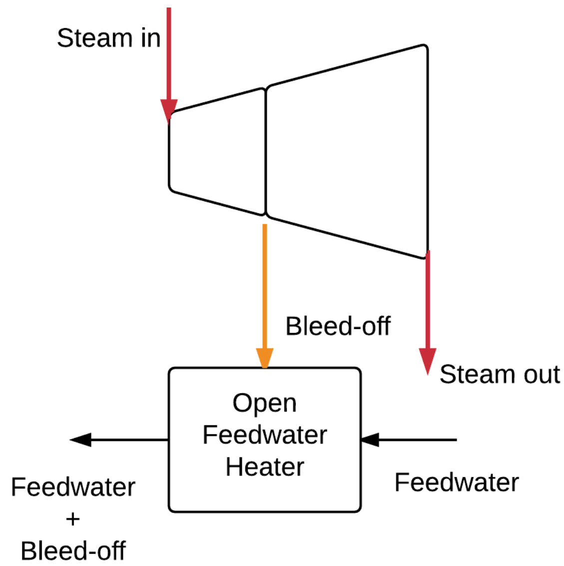

Figure 2.2: Schematic of turbine bleed-off mixing with feedwater in open feedwater heater

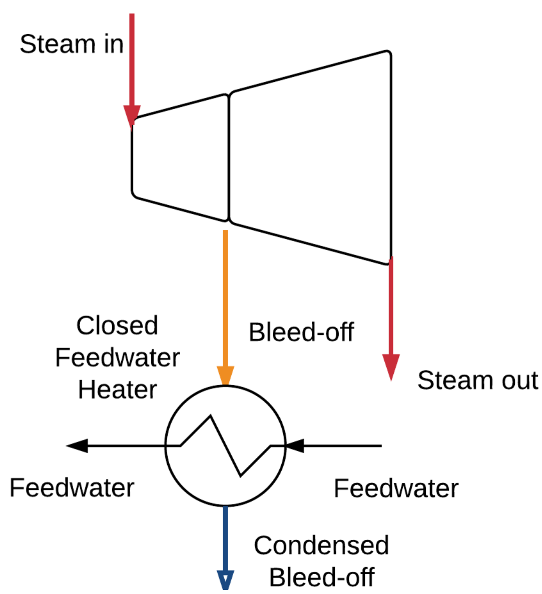

Figure 2.3: Schematic of turbine bleed-off being cooled in closed feedwater heater and exiting as a saturated fluid

Overall, 30–40% of the energy produced by the combustion of coal is converted into electrical power, around 50% is rejected to the condenser as heat, and another 5–15% is rejected through the flue. The rest is accounted for by losses between the turbine and generator, radiation losses and auxiliary power as shown in Figure 2.4 [6].

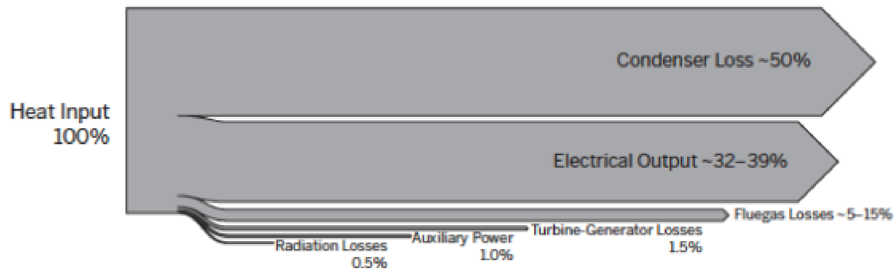

Figure 2.4: Less than half of the heat input to the boiler is converted to electrical output. Most of it is rejected to the condenser and through the flue.

2.2 Coal Plant Operation

Most coal plants are designed to combust pulverized coal. The coal is pulverized and often air dried before entering the combustor. The heat produced from the combustion of the coal is then transferred to the feedwater in the boiler. In a subcritical coal plant, the inlet pressure of the feedwater water is below 221 bar, the critical pressure of water, and the water is boiled at a constant temperature below 374 °C, the critical temperature. In a supercritical coal plant, the inlet pressure of the feedwater is above 221 bar, and the water heats up steadily until it reaches the critical temperature. Above the critical pressure and temperature, together referred to as the critical point, the liquid and vapor phases of the super-heated steam are indistinguishable. As a result, the steam continues to increase in temperature without having to undergo a phase change.

2.2.1 Subcritical

In a subcritical power plant, the inlet pressure of the feedwater is typically around 165 bar [2]. At this pressure, the phase change from liquid to vapor occurs at a constant temperature around 350 °C until all of the liquid has boiled into vapor as illustrated by a temperature-entropy digram in Figure 2.5. Throughout this phase change, the liquid and vapor phases are separated in a steam drum, where the liquid phase is collected and recirculated through the evaporative section of the boiler as shown in Figure 2.6. After exiting the steam drum, the superheated steam is heated to around 541 °C before exiting the boiler [7]. The efficiency of subcritical coal plants is in the 33 – 37% range depending on coal type, cooling system, and location.

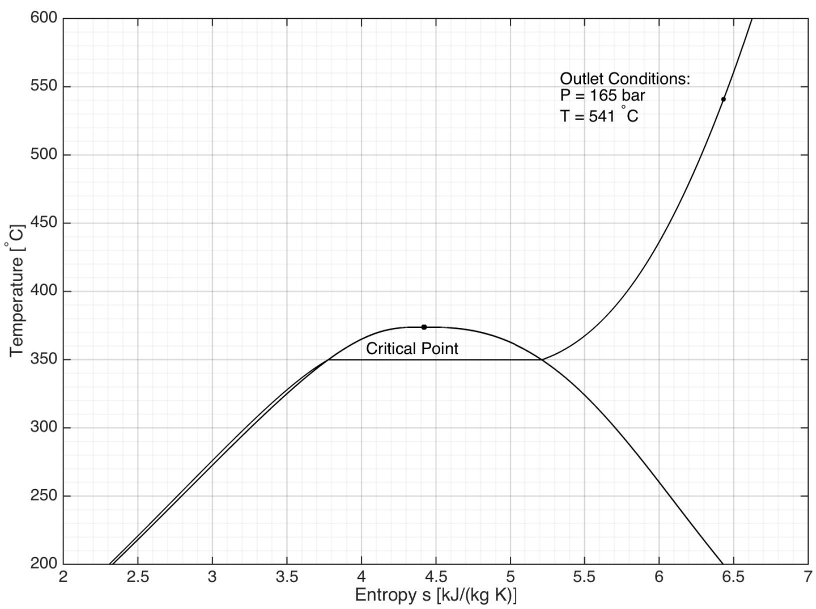

Figure 2.5: Temperature-entropy diagram for water being heated at a pressure of 165 bar and exiting as superheated steam at 541 °C.

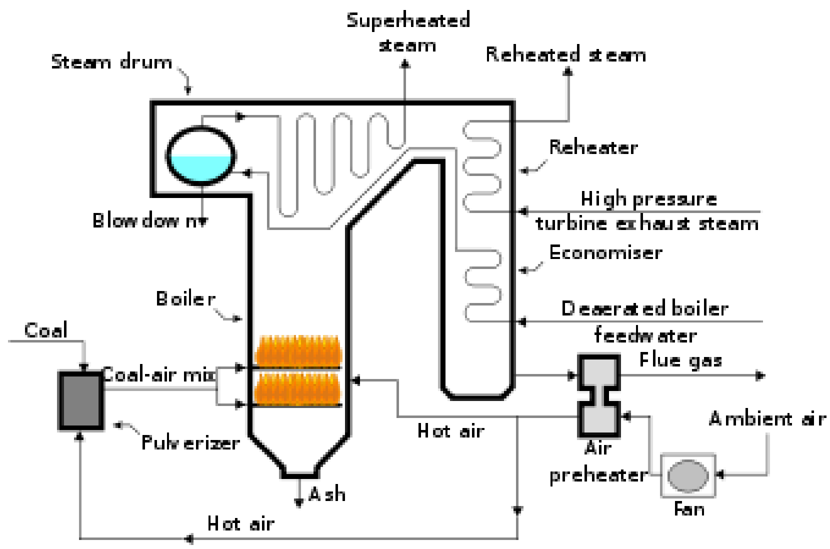

Figure 2.6: In a water-tube boiler, the liquid and solid phases are separated in a steam drum. The saturated vapor is then heated further and exits as superheated steam. [Source: Air Systems Ltd.]

2.2.2 Supercritical

In a supercritical plant, the boiler operates above the critical pressure of water, usually around 250 bar. As the steam is heated, the liquid and vapor phases become indistinguishable above the critical point, and so, rather than changing phase at a constant temperature, supercritical steam steadily increases in temperature. The steady increase in temperature increases the average temperature of heat addition from the boiler to the feedwater, increasing the boiler efficiency. The efficiency of supercritical coal plants is generally between 37 to 40% depending on coal type, cooling system, and location [2]. In the U.S., there were 170 super-critical coal plants representing 23% of the capacity of all fossil fuel generators as of 2000 [8]. The analysis presented in this report assumes that the steam exits the boiler at 550 °C with a pressure of 250 bar as illustrated in Figure 2.7.

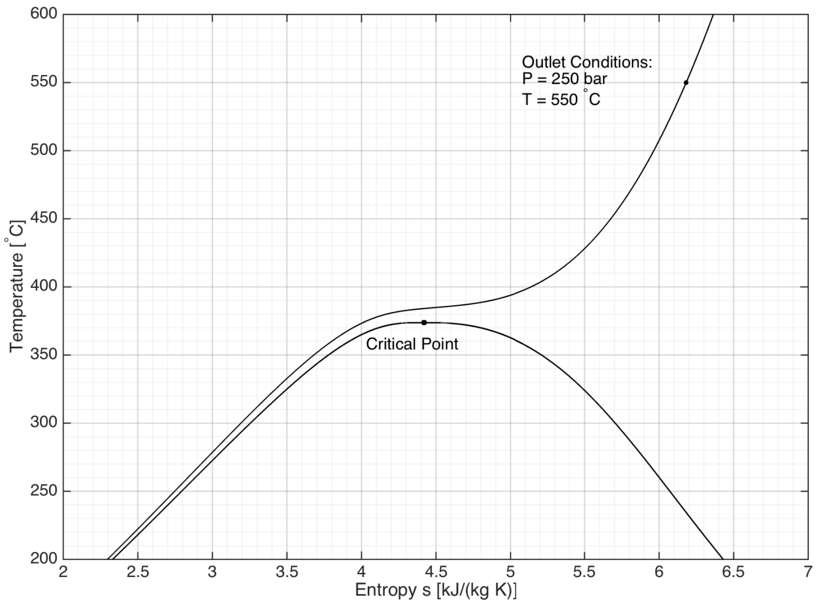

Figure 2.7: Temperature-entropy diagram for water being heated at pressure of 250 bar and exiting as superheated steam at 550 °C.

2.2.3 Other Components and Processes

In addition to the boiler, steam turbine, condenser, pumps, and feedwater heaters already discussed there are a few other components used in many coal plants that are included in the model used for this analysis.

Coal Pulverizer

The coal pulverizer is used to crush the coal before it is sent to the burner. "Primary air" is used to move the coal through the pulverizer and carry it to the burner. An example of a patent for a coal pulverizer assembly is shown in Figure 2.8. In that figure, primary air enters through opening 62, and the coal enters through opening 18. The coal is crushed by the rollers, 42, and the primary air carries the crushed coal to the burner through opening 66.

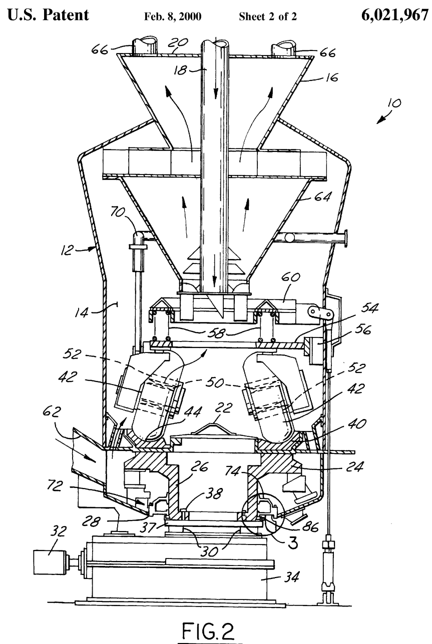

Figure 2.8: In this example of a patent for a coal pulverizer assembly, primary air moves coal through the pulverizer and into the burner. [Source: The Detroit Edison Company]

Reheater

It is desirable for the condenser to be at a low pressure to achieve a high expansion ratio through the turbines [9]. However, the minimum quality of steam exiting a turbine should be no less that 0.9 to avoid scale buildup as the result of water droplets condensing on the blades, causing them to rust [10]. The steam can be reheated and then passed through another series of turbines, exiting at the condenser pressure as saturated vapor as illustrated for a subcritical Rankine cycle in Figure 2.9. Note that while reheating the steam requires additional heat input to the boiler, it also increases the average temperature of heat addition, increasing the overall efficiency of the plant.

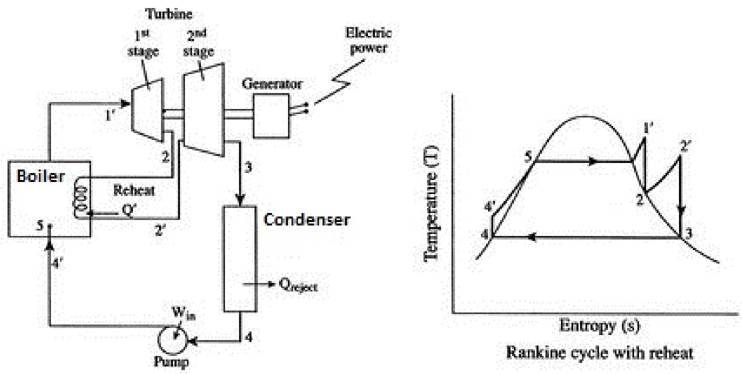

Figure 2.9: Temperature-entropy diagram of a Rankine cycle with reheat. Reheating the steam increases the average temperature of heat addition and the quality of the steam at condenser inlet.

Economizer

An economizer is a heat exchanger between the combustion gas and the feedwater as shown on the right side of Figure 2.6. In the economizer of a sub-critical coal plant, the feedwater is heated to the saturation temperature corresponding to boiler pressure [11]. The saturation temperature is the temperature at which liquid water starts to boil at a given pressure below the critical point. The flue gas exits the economizer at around 370°C.

Air Preheater

In the air pre-heater, the flue gas is cooled from 370 °C to about 155 °C, just above the acid dew point. The purpose of the air preheater is to heat the air exiting the steam-air heater to a temperature in the range of 150 – 420 °C. Preheating the air before it enters the combustor reduces the amount of energy spent raising the temperature of the air, thus increasing the rate at which heat is transferred to the boiler. The heated air can also be used to dry the coal in the pulzerizer, which increases the heating value of the coal.

Steam-Air Heater

The steam-air heater is a heat exchanger that, like the feedwater heaters, uses bleed-off from one of the steam turbines to preheat ambient air before it enters the air preheater. In general, steam-air heaters are only used during periods of extremely cold weather. However, one is included in this model for comparison with similar studies done by Sarunac and Levy wherein the steam-air heater is replaced with a flue gas cooler as described in section 2.6 [12], [13].

Forced and Induced Draft Fans

Air is brought into the the coal plant with a forced draft fan, and flue gas is blown out of the plant through an induced draft fan. The power requirements for these fans is based on the flow rates of air and flue gas and the pressure drops between the inlet and outlet of the plant and the furnace.

2.3 Coal Plant Water Use

Water is used in a variety of ways in a thermoelectric power plant. It is primarily used as the working fluid driving the steam turbines and as the coolant for the condenser[1]. Additionally for coal plants, water is used for wet desulfurization, a process used to remove sulfur from flue gas.

[1] Some power plants, particularly those in arid areas with a limited water supply, have dry cooling systems [14]. Dry cooling systems are attractive from a water use standpoint, but they can cost five times as much as water cooled systems and, in the hottest part of the year, produce 10-15% less power because of the increased condenser temperature compared to a water cooled system [15].

2.3.1 Once through Cooling

In the U.S., particularly in the east, older coal plants sited near an accessible water supply often use once-through cooling [16]. In a once-through plant situated in close proximity to a supply of cooling water, a significant amount of water is taken into the plant for cooling and then returned back to its source. Relatively little water is consumed by evaporation. The median withdrawal and consumption factors for supercritical coal plants with once-through cooling is 22,590 $\frac{gal}{MWh}$ and 103 $\frac{gal}{MWh}$, respectively [17].

With so much water being withdrawn, there are several environmental challenges associated with once-through cooling. The energy transferred from the power plant to the cooling water is a form of thermal pollution that produces areas of elevated temperatures within the body of water. The elevated temperature can be ecologically harmful, resulting in algal blooms, increased toxicity of substances such as cyanide and zinc, reduction of dissolved oxygen, and adverse effects on marine mating cycles [18]. For these reasons, the EPA regulates the temperature at which water can be discharged from a power plant [19]. These regulations are particularly problematic for plant operators in the summer when the demand for electricity for air conditioning is high and the intake temperature of cooling water tends to be higher. When the temperature of this water is too high, the plant has to reduce generation to avoid discharging water that is too hot.

Higher temperature cooling water also increases the operating pressure of the condenser, which limits the output of the turbine and decreases the efficiency of the plant. In addition to concerns about the environment and plant performance, plants that use once-through cooling are more vulnerable to water scarcity and have to curtail generation or acquire water from other sources when adequate cooling water is not available. For example, when the water level at Luminant's Martin Creek lignite plant in Texas got too low in 2011, water had to be pumped from the Sabine River eight miles away [20].

2.3.2 Closed Loop Cooling

Most newer coal plants, especially in the western U.S., use closed-loop cooling towers because they require the withdrawal of substantially less water for a given cooling load, making them more environmentally friendly and less vulnerable to water scarcity. The median withdrawal factor for a supercritical coal plant with a cooling tower is only 634 $\frac{gal}{MWh}$, less than 3% the amount required for once-through cooling [17].

Cooling towers can produce lower temperature cooling water than once-through cooling systems. Cold water from the cooling tower is circulated through the condenser to cool the steam from the power cycle. The heated up cooling water is then pumped back to the cooling tower. Because the ultimate mechanism of transferring waste heat to the environment is through evaporation, the median consumption factor for supercritical coal plants with a cooling tower is 493 $\frac{gal}{MWh}$, more than four times the consumption factor for plants with once-through cooling.

2.3.3 Flue Gas Desulfurization

A flue gas desulfurization system, also called a flue gas scrubber, as discussed in detail in the pollution control section, is a device used to remove SO_2 from the flue gas. In a wet flue gas desulfurizer, a slurry of water and lime or limestone is sprayed into the flue gas. The SO_2 reacts with the limestone, producing gypsum, which is then separated from the flue gas. Much of the water from the slurry is evaporated into the flue gas, saturating and cooling it to about 57 °C [21]. The moist, desulfurized flue gas is then rejected to the environment. A small of percentage of the water not evaporated into the flue gas is entrained in the produced gypsum, and the rest can be recovered and reused. Wet flue gas desulfurization accounts for about 10% of coal plant water consumption [21]. Dry desulfurization systems use less water, all of which is evaporated into the flue gas, leaving behind a dry waste product [7].

2.3.4 Efficiency vs. Water Consumption

It is reasonable to expect that improving the efficiency of a coal plant would also reduce the rate at which it consumes water per unit of electricity it generates. By converting a higher percentage of a heat input into electrical power, less heat is ultimately rejected to the condenser, reducing the rate at which water evaporates from the cooling tower. If, for example, a coal plant improves its efficiency by operating at higher pressures with reheat or by adding a series of feedwater heaters, the water consumption factor will decrease. However, for the same operating parameters and components, a coal plant burning higher quality coal will be more efficient and yet have a higher water consumption factor. That is because, as will be discussed in detail in the following section, the combustion gas produced by higher quality coal has much less moisture than that produced by low quality coal. As a result, much less of the heat input to the boiler is lost through the flue, improving the efficiency of the plant. However, even though the percentage of the heat input that is converted to power increases, there is an even greater increase in the percentage of the heat input that is ultimately rejected to the condenser rather than through the flue. Thus, improvements in efficiency can come at the cost of increased water consumption, and, in fact, increasing the water consumption can be a path to increasing the efficiency.

2.4 Coal Properties

Coal derives from prehistoric organic mater that, over long periods of time, has been buried, compressed at high temperatures, and progressively de-watered. As a result, coal is mostly composed of carbon and moisture in addition to small amounts of sulfur, mercury, and hydrogen. As coal matures, its mass percentage of carbon increases, improving its heating value. The types of coal considered in this report in order of quality are lignite, subbituminous, and bituminous. Typical ranges for the wet mass percentage of carbon, moisture, and sulfur and the heating value of each type of coal based on U.S. values are shown in Table 2.1 [2], [12].

Table 2.1: Higher heating value [$\frac{MJ}{kg}$] and wet mass composition [wt%] for different types of coal based on U.S. values

	HHV	Carbon	Moisture	Sulfur
Lignite	14.5	30 - 40	30 - 40	0.5 - 1.6
Subbituminous	20.8	45 - 50	28 - 30	0.3 - 0.5
Bituminous	28.2	60 - 70	3 - 13	2.0 - 5.0

Moisture in coal is described as either 'surface' or 'inherent' moisture. Surface moisture in coal results from the absorption of water upon exposure to moist air. and it can be mostly removed by drying the coal. Inherent moisture in coal derives from the original plant matter that formed the coal and is lower for more mature coals. Lower rank coals like lignite have the highest moisture content, while bituminous and anthracite, another high-carbon coal, have the lowest.

While bituminous coals are desirable because of their high heating value, they also have a higher sulfur content than other coals. The combustion of sulfur is problematic for the role it plays in the formation of acid rain, as discussed in subsequent sections. Hydrogen makes up 3 – 6% of the mass composition of coal. The water produced from the combustion of the hydrogen in the coal contributes to the moisture in the flue gas.

2.4.1 Boiler Flue Gas

The composition of the flue gas is greatly influenced by the type of coal burned in the furnace. The majority of moisture in the flue gas comes from water embedded in the coal. Combustion of hydrogen within the coal is another source of moisture for the flue gas. Lastly, a small amount of water is brought in with the ambient air depending on the local humidity. The higher the moisture content of the flue gas, the higher the dew point, and thus, the less the flue gas has to be cooled before water starts to condense out of it.

For a given drop temperature drop below the dew point, more energy is transferred from the latent heat released from the flue gas during condensation than from the sensible heat released in lowering the temperature of the flue gas. Thus, the minimum effectiveness of a flue gas cooler is lower if the flue gas is releasing latent heat in addition to sensible heat.

The combustion of sulfur produces SO_2 and SO_3, collectively referred to as SO_x. SO_x emissions, in addition to emissions of NO_x, mercury, particulate matter, and other pollutants, are particularly problematic from an environmental standpoint. As a result, emissions reductions systems such as electrostatic precipitators, flue gas scrubbers, and selective catalytic reduction systems are used in many coal plants to reduce emissions.

It is useful to differentiate between wet flue gas, that is all of flue gas including the water vapor, and dry flue gas, the flue gas constituents excluding the water vapor. Dry flue gas, despite differences in composition, has roughly similar thermal properties from smokestack to smokestack regardless of coal type.

2.4.2 Cost of Different Coal Types

Using low energy density, high moisture coals for electricity generation incurs higher capital costs and reduced generation efficiency. Compared to a coal plant built to burn Pittsburgh #8 bituminous coal, a pulverized coal plant built to burn Powder River Basin subbituminous coal costs an additional 14%. Similarly, a plant built to burn Texas lignite costs 24% more than a coal plant built to burn Pittsburgh #8 [2]. The reduction in generation efficiency, the result of reduced boiler efficiency due to flue gas moisture, is not as significant as the increase in capital cost. However, because the fuel cost is relatively lower, the levelized cost of electricity for plants burning Texas lignite or PRB are the same or lower than those burning Pittsburgh #8 [2].

2.5 Pollution Control

There are a variety of environmental impacts associated with coal-fired electricity generation. One of the most prominent of these issues is acid rain resulting from the emission of SO_x and NO_x into the atmosphere. The main precursor to acid rain is SO_2, and two-thirds of SO_2 emissions in the U.S. are the result of fossil fuel electricity generation [EPA]. Power plants also account for 25% of the NO_x emissions in the U.S. [22].

After combustion, SO_2 reacts with hydroxide and then oxygen to form SO_3, as described by the following chemical reactions.

$$S + O_2 \rightarrow SO_2 \tag{2.1}$$

$$SO_2 + OH \rightarrow HOSO_2 \tag{2.2}$$

$$HOSO_2 + O_2 \rightarrow HO_2 + SO_3 \tag{2.3}$$

In the presence of water, SO_3 reacts to form sulfuric acid.

$$SO_3(g) + H_2O(l) \rightarrow H_2SO_4 \tag{2.4}$$

Nitric acid, another major component of acid rain, is formed from the reaction of nitrogen dioxide with hydroxide.

$$NO_2 + OH \rightarrow HNO_3 \tag{2.5}$$

Wet acid deposition happens when sulfuric and nitric acid are removed from the atmosphere through any kind of precipitation. Dry deposition occurs when acids stick to the ground or other terrestrial surface such as trees or buildings in the absence of precipitation. Aspects of the environment negatively affected by acid rain include bodies of water and aquatic wildlife, soil, forests, and other plant life. Acid rain does not directly harm humans. However, the particles that cause acid rain, namely SO_2 and NO_x, react with the atmosphere to form fine sulfate and nitrate particles that can contribute to respiratory conditions such as asthma and bronchitis [23].

As a result of environmental concerns related to the combustion of coal, the U.S. Congress passed the Clean Air Act in 1970. The purpose of the Clean Air Act was to set enforceable air quality standards and regulate emissions. The power of the EPA and the federal government was expanded by amendments to the Clean Air Act in 1990 that set up a cap and trade system for SO_2 and emission reduction guidelines for NO_x [2]. There are a variety of components used in coal plants for the purpose of complying with these environmental standards as well as reducing emissions of particulate matter and other pollutants.

2.5.1 Electrostatic Precipitator

After exiting the boiler, the flue gas goes through an electrostatic precipitator, ESP, as shown in Figure 2.10.

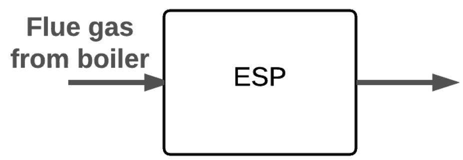

Figure 2.10: After exiting the boiler, the flue gas goes through an electrostatic precipitator, ESP, where fine solid particles are removed.

An ESP is used to remove fine particles, particulate matter, from the flue gas. A high voltage electrode, 20 – 70kV, is used to produce an electric field that negatively charges the particles in the flue gas. A positively charged metal plate is then used to attract the negatively charged particles, removing them from the flue gas. Nearly 25% of mercury in the flue gas is also removed by the ESP. Nearly all coal plants in the U.S. use an electrostatic precipitator that removes approximately 99.9% of all particulate matter from the flue gas [2]. A detailed illustration of an ESP is shown Figure 2.11.

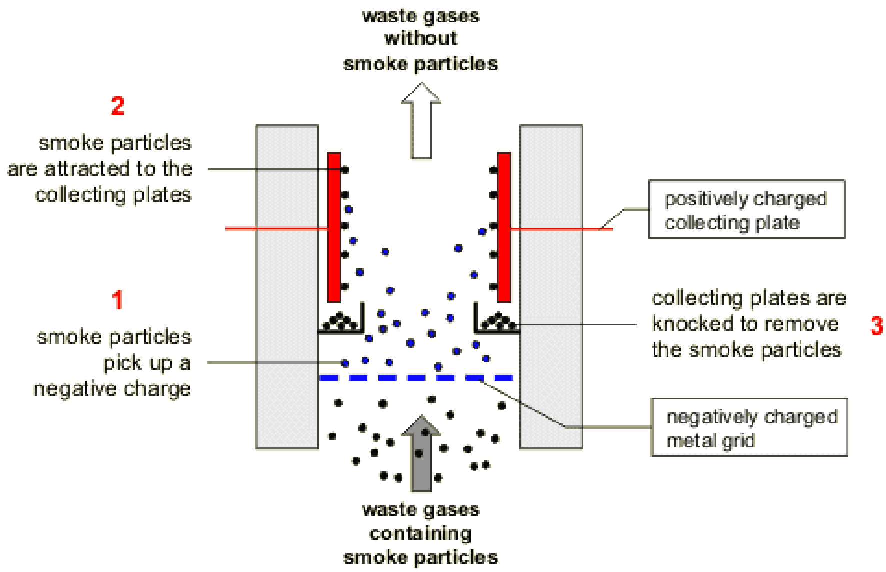

Figure 2.11: Illustration of an electrostatic precipitator removing particulate matter from flue gas [Source: BBC]

2.5.2 Flue Gas Desulfurization System

For plants with a flue gas desulfurization system, also called a flue gas scrubber, the flue gas enters the scrubber upon exiting the ESP as shown in Figure 2.12.

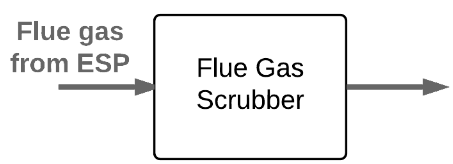

Figure 2.12: For plants with a flue gas desulfurization system, the flue gas enters the SO_x scrubber upon exiting the ESP.

In 2010, 60% of U.S. coal-fired electricity was generated in a plant with some kind of SO_2 scrubber [24]. However, more SO_2 is produced by coal plants than any other source, mostly from units without scrubbing technology. Coal plants without scrubbers only account for 40% of coal-fired electricity but produce 70% of the SO_2

emissions. The most commonly used flue gas desulfurization technology is wet limestone scrubbing [25]. An illustration of a wet flue gas desulfurizer is shown in Figure 2.13. Wet flue gas desulfurization systems remove 95 – 99% of flue gas SO_2. They also remove 40 – 60% of the mercury in the flue gas [2].

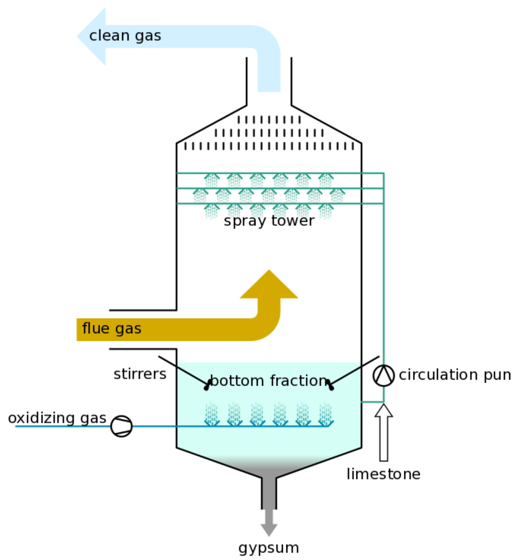

Figure 2.13: Illustration of a wet flue gas scrubber [Source: Wikimedia Commons]

Dry desulfurization systems, also called dry scrubbers or spray dryers, are also used to limit SO_2 emissions, primarily for plants burning low sulfur coal [7]. In contrast to wet scrubbers, dry scrubbers are generally located upstream of the a particulate control device such as an ESP or baghouse as shown in the Figure 2.14. Dry flue gas scrubbers can remove over 90% of flue gas SO_2 [26].

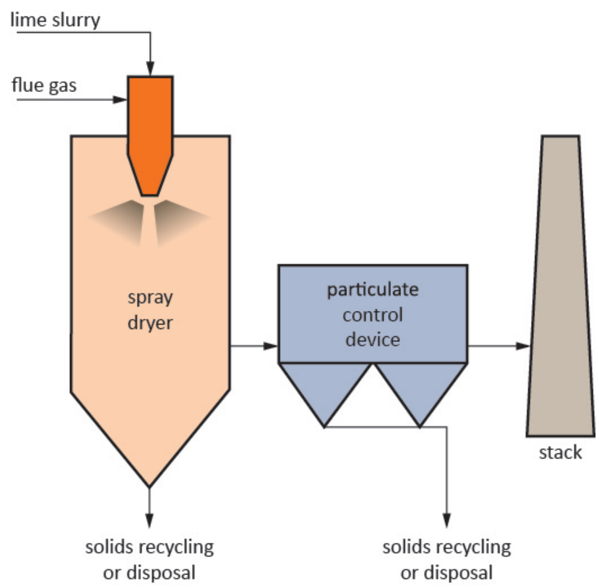

Figure 2.14: Illustration of a dry flue gas scrubber [Source: Cornerstone]

2.5.3 Selective Catalytic Reduction

About 10% of US coal generation capacity includes post combustion NO_x control [2]. Even so, there are a variety of technologies available for reducing NO_x emissions. One of the most efficient technologies, selective catalytic reduction, SCR, can achieve a removal rate over 90%. Selective catalytic reduction refers to processes by which a catalyst is used to facilitate the reaction of NO_x with a reductant and oxygen into nitrogen and water. An example of such a reaction with NO and anhydrous ammonia is shown in the equation below.

$$4NO + 4NH_3 + O_2 \rightarrow 4N_2 + 6H_2O \tag{2.6}$$

The downside of using anhydrous ammonia as a reductant is that it is toxic and difficult to transport and store. Thus, it is often transported and stored as aqueous ammonia and then hydrolyzed before it can be used. Urea, another reductant that is even easier to store, is also used, but must first be converted to ammonia through thermal decomposition. Another downside of using urea as a reductant is that it produces additional CO_2 emissions, as shown in the following chemical reaction.

$$4NO + 2(NH_2)_2CO + O_2 \rightarrow 4N_2 + 4H_2O + 2CO_2 \tag{2.7}$$

In addition to removing NO_x, SCR in conjunction with an FGD system increases the mercury removal rate to around 95%.

2.6 Low-Temperature Waste Heat Recovery

As mentioned in previous sections, heat is usually extracted from the flue gas in the air preheater until it has cooled to around 155 °C, just above the acid due point. Note that acid dew point, around 150 °C, is much higher than the dew point for water. Below the dew point, acids within the flue gas start to condense, causing corrosion and scale buildup. The scale can obstruct the flow of the flue gas, increasing the back pressure on the fan and decreasing plant performance. However, a flue gas cooler, also known as a low temperature economizer, can be made with corrosion resistant material such as a fluoropolymer or phenolic coating and used to extract additional heat from the flue gas [27]. Implementation of flue gas coolers is most common in Europe, followed by Japan and a few pilot programs in the U.S. Flue gas coolers can be used to increase plant efficiency by preheating air or feedwater. Additionally, cooling the flue gas to a sufficiently low temperature introduces the possibility of recovering moisture from the flue gas, which this thesis will examine closely.

2.6.1 Preheating Air or Feedwater

A comprehensive study on the potential for low temperature waste heat recovery from boiler flue gas to improve power plant performance was published by Sarunac [12]. For that study, ASPEN, a chemical process optimization package, was used to build a baseline model of a supercritical coal plant that included a steam-air heater and a series of seven feedwater heaters.

The model was adjusted by replacing the steam-air heater, the two lowest temperature feedwater heaters, or all three with a flue gas cooler. According to his results, preheating the combustion air with a flue gas cooler yields an increase in efficiency of approximately 0.5% regardless of coal type. Preheating the feedwater achieves a 0.5 – 0.8% increase in efficiency, with lignite burning plants experiencing the greatest improvement. Preheating both the feedwater and the combustion air achieves a 0.6 – 0.9% increase in efficiency, with lignite burning plants again experiencing the greatest improvement.

In addition to improving efficiency, cooling the flue gas closer to or below its dew point reduces the rate at which water is evaporated in a flue gas scrubber. Cooling the flue gas to around 70 °C before entering the flue gas desulfurizer can result in as much as an 80% reduction in the evaporation rate. More accurate calculations for how cooling the flue gas would reduce the rate of water consumption by the flue gas scrubber is beyond the scope of this analysis.

By recovering waste heat from the flue gas, some of that heat is converted to work, but most of it is diverted to the condenser, thus increasing the load on the cooling tower and the rate at which heat is rejected to the environment through evaporation. One of the goals of my analysis is to quantify the increase in water consumption rate as the result of waste heat recovery from the flue gas.

2.6.2 Moisture Recovery

This report investigates the use of a condensing heat exchanger, referred to as a flue gas dryer, with ambient air as the heat sink as a means of recovering moisture from the flue gas. Some of the moisture is also recovered from the flue gas cooler, in the case where the flue gas is cooled below its dew point and transfers latent heat to feedwater. This concept is based on the arrangement described by Levy et. al. in a series of technical reports on the recovery of water from coal boiler flue gas [13]. The results of these studies indicate that several factors influence the fraction of water in the flue gas that can be recovered by the flue gas dryer. The higher the initial moisture content of the flue gas, the higher its dew point. A higher dew point enables more condensation for the same energy transfer from the flue gas to the combustion air or feedwater. Thus, the recovery fraction is higher for flue gas produced by high moisture coal.

The temperature of the heat sink also affects the fraction of water that can be recovered. With boiler feedwater at an inlet temperature of 30 °C as a heat sink, capture efficiency is 0 – 10%. With air as the heat sink, there is a seasonal difference in the rate of heat rejection and condensation based on the temperature difference between the flue gas and ambient air in the summer or winter. With pre-combustion air[2] as a heat sink, capture efficiency is 20 – 25% in the winter and 10 – 18% in the summer. With ambient air[3] as a heat sink, capture efficiency is 80 – 87% in the winter and 48 – 72% in the winter [13].

The extent to which the water recovered by the flue gas dryer has to be treated depends on its acidity. For a coal plant with a wet flue gas desulfurizer before the flue

[2]The discharge from the forced draft fan.

[3]That is, ambient air that is not being pre-heated for combustion, but is instead just used as a heat sink for the condensing flue gas.

gas dryer, most of the sulfuric acid will already be removed from the flue gas. The acid dew point, around 150 °C, is 100 °C or more above the dew point for the water in the flue gas. Thus, a significant percentage of the acid will condense out of the flue gas and can be removed before any water condenses, thus reducing the acidity of the water recovered from the flue gas. Even without any low temperature waste heat recovery, the flue gas dryer could be designed such that any acid that condenses out before the water dew point is separated and removed. The recovered water would then have to go through a pH control process, wherein a caustic is mixed with the water and reacts with the acid to form salt and water.

2.7 Summary of Background

The background chapter provides an overview of coal plant energy flows, operation, water use, and pollution control. It also includes a descriptions of various compositions of coal and how coal composition affects the composition of the flue gas and the cost power plant. It concludes with a discussion of low-temperature waste heat and water recovery from boiler flue gas.

Using the information included in this section, I constructed an original thermodynamic model in MATLAB of a supercritical coal plant with a cooling tower. This model estimates the potential improvements in efficiency that can be achieved by recovering waste heat from boiler flue gas to preheat air or feedwater and the associated increase in water consumption. This model also estimates the rate at which water can be recovered from the flue gas. The inputs and equations used in the model are described in detail in the following chapter.

Chapter 3

Methodology

3.1 Overview

For this analysis, I constructed a thermodynamic model of a supercritical coal plant with a cooling tower based on the models used by Sarunac and Levy in their respective reports [12], [13]. The arrangement of components and temperatures are largely the same for their models and mine. Instead of building the model in ASPEN, I wrote all of the relevant thermodynamic equations in MATLAB. Many of these equations include outputs from XSteam, an open source function for looking up and solving for the thermodynamic properties of water. The scenarios for waste heat recovery that I considered were also based on Sarunac and Levy's reports, with a few alterations. The heat input was calculated based on a first law analysis of the boiler (often referred to as boiler\economizer in this report). The net power output was calculated based on a first law analysis of the steam turbine, pumps, and generator. The evaporation rate of cooling water is determined from a first law analysis of the condenser and the cooling tower. With the power output and evaporation rate, the water consumption factor in $\frac{gal}{MWh}$ was determined for the variations of the model.

For scenario A, the steam-air heater was replaced with a flue gas cooler. For scenario B, the lowest temperature feedwater heater was replaced with a flue gas cooler. For scenario C, the two lowest temperature feedwater heaters were replaced with a flue gas cooler and, in this scenario, the flue gas cooler also recovers some of the moisture from the flue gas. The feasibility of each of these scenarios is considered in

Table 3.1: Summary of scenarios for low temperature waste heat recovery and components replaced with an FGC

Scenario	Components replaced with FGC
A	Steam-air heater
B	Feedwater heater G replaced
C	Feedwater heaters F & G

terms of the minimum heat exchanger effectiveness required for the flue gas cooler and the improvement in efficiency as the result of low temperature waste heat recovery. A summary of scenarios for low temperature waste heat recovery and the components replaced with an FGC is given in Table 3.1.

Upon exiting the flue gas cooler, FGC, the flue gas goes through a flue gas dryer. The outlet temperature of the flue gas, and thus the rate at which water is recovered by the flue gas dryer, is determined as a function of the heat exchanger effectiveness of the flue gas dryer for the variations of the model. The water recovery factor in $\frac{gal}{MWh}$ can be determined from the power output and the water recovery rate from the flue gas dryer. The ratio of the water recovery factor and water consumption factor are considered an indication of how much of the water used in cooling a supercritical coal plant can be recovered from the flue gas. Including a flue gas desulfurizer in the model adds additional complexity to calculation of the water consumption and recovery factors. Thus, for computational simplicity, a flue gas desulfurizer was not included in the model.

3.2 Coal and Flue Gas Properties

To model the waste heat and water recovery from flue gas, the chemical composition of the flue gas has to be determined based on the composition of the fuel and combustion air. The composition of dry air on a mass basis is shown Table 3.2, and the wet-mass composition of coals used in the power plant model are shown in Table 3.3.

Table 3.2: Mass composition of dry air

N_2	76.28
O_2	23.29
CO_2	0.30
Ar	0.13

Table 3.3: Composition of coals on a wet mass basis ($\frac{kg}{kg_{coal}}$)

	Lignite	**Subbituminous**	**Bituminous**
Carbon	0.34	0.49	0.60
Oxygen	0.11	0.12	0.08
Hydrogen	0.03	0.04	0.04
Nitrogen	0.01	0.01	0.01
Sulphur	0.01	0.00	0.04
Chlorine	0.00	0.00	0.00
Ash	0.12	0.06	0.14
Moisture	0.39	0.28	0.08

The amount of oxygen required to completely combust a kilogram of coal can be determined from the stoichiometric coefficient of oxygen in each of the combustion reactions, the mass fractions of carbon, hydrogen, sulfur, and oxygen for each coal type, x_i, and the molecular weight of each of the reactants, M_i . The stoichiometric combustion reactions for carbon, hydrogen, and sulfur are shown in the following equations:

$$C + O_2 \rightarrow CO_2 \tag{3.1}$$

$$H_2 + \frac{1}{2}O_2 \rightarrow H_2O \tag{3.2}$$

$$S + O_2 \rightarrow SO_2 \tag{3.3}$$

$$O_{2,required,C} = \frac{M_{O_2}}{M_C} x_C \tag{3.4}$$

$$O_{2,required,H_2} = \frac{1}{2} \frac{M_{O_2}}{M_{H_2}} x_{H_2} \tag{3.5}$$

$$O_{2,required,S} = \frac{M_{O_2}}{M_{SO_2}} x_S \tag{3.6}$$

$$O_{2,required} = \sum_i O_{2,required,i} - x_{O_2}, \text{ for } i = \{C, H_2, S\} \tag{3.7}$$

The mass of the resultant combustion products, CO_2, H_2O, and SO_2, per kg of coal, θ_i, can be determined from the following equations:

$$\theta_{CO_2} = x_C + O_{2,required,C} \tag{3.8}$$

$$\theta_{H_2O} = x_{H_2} + O_{2,required,H_2} \tag{3.9}$$

$$\theta_{SO_2} = x_S + O_{2,required,S} \tag{3.10}$$

The required oxygen can be divided by the mass percentage of oxygen in air, $\nu_{O_2,air}$, to determine the stoichiometric air-fuel ratio, as shown in the following equation.

$$AFR_{stoich} = \frac{O_{2,required}}{\nu_{O_2,air}} \tag{3.11}$$

The excess air, E, can then be used to determine the air-fuel ratio for the combustor. This model assumes 21% excess air.

$$AFR = (1 + E) AFR_{stoich} \tag{3.12}$$

After combustion, the resultant flue gas is a mixture of the unreacted gases brought in with air and coal, mostly N_2, H_2O, and excess O_2, and the combustion products, CO_2, H_2O, and SO_2. The mass of the unreacted N_2, Ar, and $C\ell$ can be determined from the following equation:

$$\phi_i = AFR\ \nu_{i,air} + x_i,\ \text{for } i = \{N_2, Ar, C\ell\} \tag{3.13}$$

The mass of the combustion products, CO_2 and SO_2, in the flue gas per kilogram of coal combusted, ϕ_i, can be determined from the following equation:

$$\phi_i = AFR\ \nu_{i,air} + \theta_i,\ \text{for } i = \{CO_2, SO_2\} \tag{3.14}$$

The mass of oxygen in the flue gas per mass of coal combusted depends on the difference between the actual and stoichiometric air-fuel ratios.

$$\phi_{O_2} = (AFR - AFR_{stoich})\nu_{O_2,air} \tag{3.15}$$

The water in the exhaust comes from three sources: the moisture of the coal, water produced from the combustion of hydrogen, and water vapor in the air, as shown in the following equation:

$$\phi_{H_2O} = AFR\ \nu_{H_2O,air} + \theta_{H_2O} + x_{H_2O} \tag{3.16}$$

The exhaust-fuel ratio, EFR, is the sum of the constituents in the flue gas, and the flue gas-fuel ratio, EFR_{dry}, excludes the mass of water in the exhaust.

$$EFR = \sum_i \phi_i \tag{3.17}$$

$$EFR_{dry} = EFR - \phi_{H_2O} \tag{3.18}$$

The wet and dry mass compositions of the flue gas are given by the following equations:

$$\nu_{i,fg,wet} = \frac{\phi_i}{EFR}\ \text{for } i = \{Constituent \in Wet\ Flue\ Gas\} \tag{3.19}$$

$$\nu_{i,fg,dry} = \frac{\phi_i}{EFR_{dry}} \text{ for } i = \{Constituent \in Dry\ Flue\ Gas\} \tag{3.20}$$

The mass averaged specific heat of the flue gas is the sum of the products of the dry mass composition and specific heat for each of the constituents, $c_{p,i}$. The values used for the gas specific heats are included in the appendix.

$$c_{p,fg} = \sum_i \nu_i\ c_{p,i} \text{ for } i = \{Constituent \in Dry\ Flue\ Gas\} \tag{3.21}$$

The volumetric composition of the flue gas is required to determine its molecular weight, water vapor pressure, and dew point.

$$y_{i,wet} = \frac{\nu_i M_i}{\sum_i \nu_i M_i} \text{ for } i = \{Constituent \in Wet\ Flue\ Gas\} \tag{3.22}$$

$$y_{i,dry} = \frac{\nu_i M_i}{\sum_i \nu_i M_i} \text{ for } i = \{Constituent \in Dry\ Flue\ Gas\} \tag{3.23}$$

The molecular weight of the flue gas can be determined as a molar average of the constituents in the dry flue gas.

$$M_{fg,dry} = \sum_i y_i M_i \text{ for } i = \{Constituent \in Dry\ Flue\ Gas\} \tag{3.24}$$

The vapor pressure of water in the flue gas can then be determined by multiplying the molar composition of water in the flue gas by the atmospheric pressure. The dew point temperature is the saturation temperature corresponding to the vapor pressure.

$$P_{vap} = y_{H_2O} P_{atm} \tag{3.25}$$

$$T_{dew} = T_{sat,\ P=P_{vap}} \tag{3.26}$$

The partial pressures experienced by the dry flue gas and water vapor can be related to each other by the ratio of their molecular weights.

$$c = \frac{M_{H_2O}}{M_{fg,dry}} \tag{3.27}$$

For flue gas at a temperature below the dew point, the relative humidity can be determined with the following equation.

$$\omega = \frac{P_{vap,\ T<T_{dew}}}{(P_{atm} - P_{vap})} \tag{3.28}$$

3.3 Thermodynamic Model of Supercritical Coal Plant

A schematic for the supercritical coal plant simulated in the baseline model is shown in Figure 3.1. Energy rate balances for each of the components are given roughly in the order they were solved in the model.

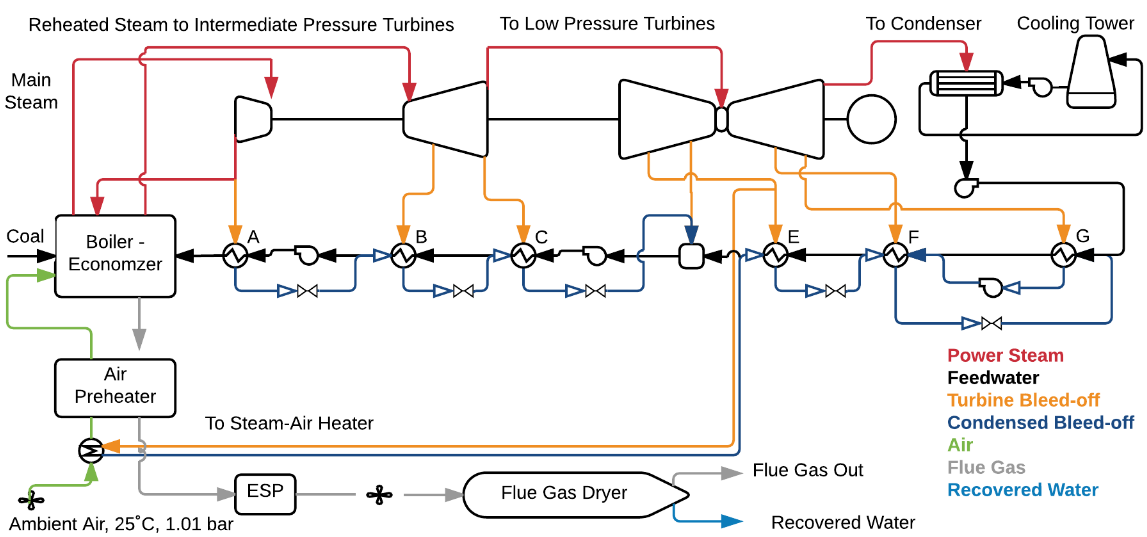

Figure 3.1: Schematic of baseline model for supercritical coal plant

Steam Turbines

In this model of a super-critical coal plant, mechanical power is generated by numerous turbines at high, intermediate, and low pressures. The calculations for each of these turbines are essentially the same except for differences in the turbine inlet and outlet pressures and the flow rate of steam through the turbines. In between each turbine, some amount of steam is bled off to supply heat to one of seven feedwater heaters. Thus, each subsequent turbine has a lower mass flow rate, $\dot{m}_i$. For any of the turbines in this model, the mechanical power output, $\dot{W}_i$, can be determined from the mass flow rate, isentropic efficiency, $\eta_{s,i}$, inlet enthalpy, h_{in}, and isentropic outlet enthalpy, $h_{out,s}$, as shown in the following equation.

$$\dot{W}_i = \eta_{s,i} \; \dot{m}_i (h_{in} - h_{out,s})_i \tag{3.29}$$

The isentropic efficiencies for each of the turbines are given in Table 3.4. Note that HPT, IPT, and LPT stand for high, intermediate, and low turbines respectively. Similarly, BFP, BP, and DP stand for boiler feedwater, booster, and drain pumps respectively.

Table 3.4: Isentropic efficiencies of steam turbines in model of supercritical coal plant

Component	η_s[%]
HPT	85.08
IPT - 1	83.54
IPT - 2	86.48
LPT - 1	87.50
LPT - 2	89.69
LPT - 3	89.87
LPT - 4	89.73
LPT - 5,6	67.25
BFP	82.88
BP	92.52
DP	85.00

The inlet steam enthalpy and entropy can be determined from the inlet temperature and pressure.

$$h_{i,in} = h_{(T,P)_{in}} \tag{3.30}$$

$$s_{i,in} = s_{(T,P)_{in}} \tag{3.31}$$

However, for most of the intermediate and low pressure turbines, the inlet enthalpy is equal to the outlet enthalpy of the previous turbine. In that case, the inlet entropy can be determined from the inlet enthalpy and pressure as shown in the following equation.

$$s_{i,in} = s_{(h,P)_{in}} \tag{3.32}$$

The isentropic outlet enthalpy can determined from the inlet entropy and outlet pressure.

$$h_{i,out,s} = h_{(P_{out},\ s_{in})} \tag{3.33}$$

Once the inlet enthalpy and isentropic outlet enthalpy have been determined, the power output of the turbine can be calculating using equation 3.29. The turbine power output can then be used in the following equation to determine the actual outlet enthalpy of the turbine.

$$h_{i,out} = h_{i,in} - \frac{\dot{W}_i}{\dot{m}_i} \tag{3.34}$$

For clarity, the calculations for each of the turbines will be written out in full detail.

Feedwater Heaters

Most of the feedwater heaters in this model are closed, meaning that while heat is extracted from turbine bleed-off in a heat exchanger, the mass of the steam being preheated is constant. The change in enthalpy of the water being pre-heated in a closed feedwater heater will be labeled as Δh_{fwh}. The bleed-off exits the feedwater heaters as a saturated fluid at the bleed-off pressure. Thus, the enthalpy of the bleed-off exiting the feedwater heater, $h_{bo,out}$, can be determined as follows:

$$h_{bo,out} = h_{(x=0,P)} \tag{3.35}$$

The mass flow rate of bleed-off from the turbines is determined from energy rate balances of each of the feedwater heaters.

Air Preheater

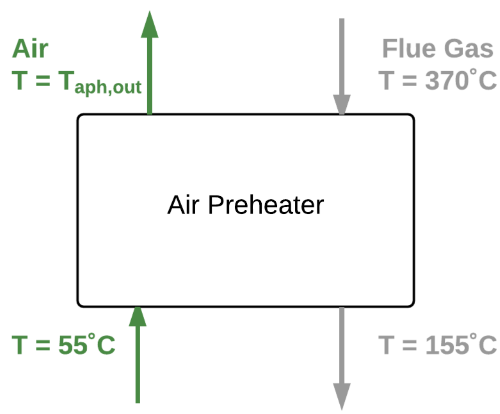

Figure 3.2: Detailed schematic of air preheater as defined in all scenarios

Flue gas typically exits the economizer around 375 °C [11], [12]. It can then be used to preheat combustion air, which cools it to 155 °C, slightly above the acid dew point. The combustion air enters the air preheater after exiting a steam-air heater at 55 °C. The temperature at which the air exits the air preheater and enters the boiler can be determined from the following equation:

$$T_{air,aph,out} = \frac{EFR_{dry}(c_p(T_{be,out} - T_{aph,out}))_{fg} - \phi_{H_2}\Delta h_{v,aph}}{AFR\ c_{p,air}} + T_{air,aph,in} \tag{3.36}$$

The subscripts *aph* and *be* stand for 'air pre-heater' and 'boiler/economizer' respectively. The change in enthalpy of the water vapor through the air preheater can be determined from the inlet and outlet temperatures of the flue gas in the air preheater:

$$\Delta h_{v,aph} = (h_{(T=370°C)} - h_{(T=155°C)})_{P=1.01bar} \tag{3.37}$$

Where the specific heat of ambient air, $c_{p,air}$, is assumed to be $1.01\frac{kJ}{kgK}$.

High Pressure Turbine

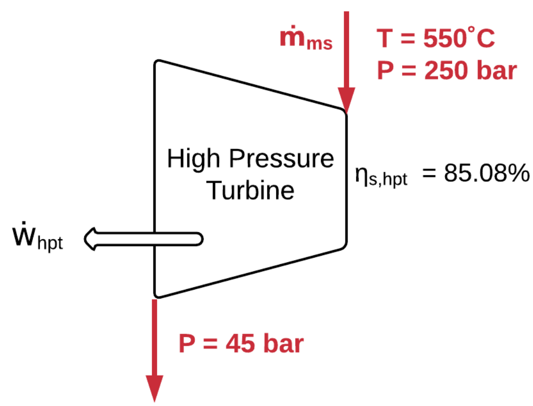

Figure 3.3: Detailed schematic of high pressure turbine as defined in all scenarios

The flow rate of the steam passing through the high pressure turbine, $\dot{m}_{ms}$, was set to $500\frac{kg}{s}$. An energy rate balance of the high pressure turbine is depicted by the following equations:

$$\dot{W}_{hpt} = \eta_{s,hpt}\ \dot{m}_{ms}(h_{in} - h_{out,s})_{hpt} \tag{3.38}$$

$$h_{hpt,in} = h_{(T=550°C,\ P=250bar)} \tag{3.39}$$

$$s_{hpt,in} = s_{(T=550°C,\ P=250bar)} \tag{3.40}$$

$$h_{htp,out,s} = h_{(P=45bar,\ s=s_{hpt,in})} \tag{3.41}$$

$$h_{hpt,out} = h_{hpt,in} - \frac{\dot{W}_{hpt}}{\dot{m}_{ms}} \tag{3.42}$$

Boiler Feedwater Pump

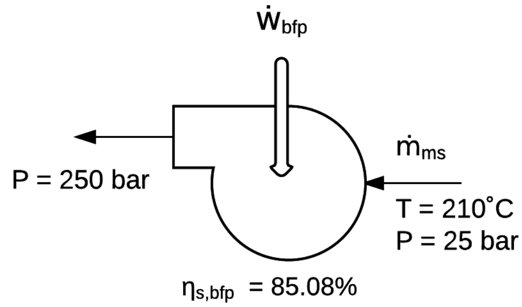

Figure 3.4: Detailed schematic of boiler feedwater pump as defined in all scenarios

Before feedwater enters Feedwater Heater A, it is pressurized to 250 bar by a boiler feedwater pump. The power input required for the boiler feedwater pump is given by the following equation.

$$\dot{W}_{bfp} = \frac{\dot{m}_{ms}}{\eta_{s,bfp}}(h_{out,s} - h_{in})_{bfp} \tag{3.43}$$

Where the enthalpy and entropy and the pump inlet can be determined from the temperature and pressure.

$$h_{bfp,in} = h_{(T=210°C,\ P=25bar)} \tag{3.44}$$

$$s_{bfp,in} = s_{(T=210°C,\ P=25bar)} \tag{3.45}$$

The isentropic outlet entropy can then be determined from the inlet entropy and outlet pressure.

$$h_{bfp,out,s} = h_{(P=250bar,\ s=s_{bfp,in})} \tag{3.46}$$

The steam enthalpy at the outlet of the boiler feedwater pump can be determined from the required power input.

$$h_{bfp,out} = h_{bfp,in} + \frac{\dot{W}_{bfp}}{\dot{m}_{ms}} \tag{3.47}$$

Feedwater Heater A

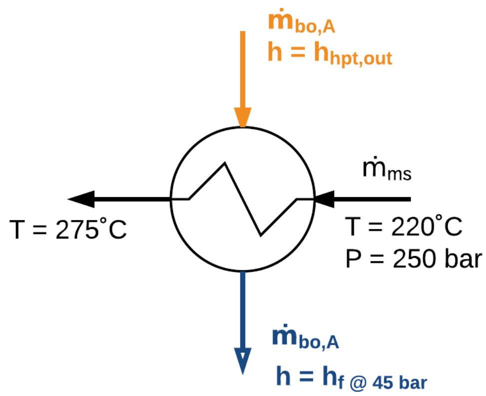

Figure 3.5: Detailed schematic of boiler feedwater heater A as defined in all scenarios

From the high pressure turbine, most of the main steam is reheated in the boiler and economizer. A small amount, $\dot{m}_{bo,A}$, is bled off and used to to preheat steam feedwater heater A and exits as a saturated liqid at 45 bar.

$$\dot{m}_{bo,A}(h_{hpt,out} - h_{bo,A,out}) - \dot{m}_{ms}\Delta h_{fwh,A} = 0 \tag{3.48}$$

$$\Delta h_{fwh,A} = h_{(T=275°C,\ P=250bar)} - h_{bfp,out} \tag{3.49}$$

$$h_{bo,A,out} = h_{(x=0,\ P=45bar)} \tag{3.50}$$

Rearranging equation 3.48 results in the following equation for $\dot{m}_{bo,A}$.

$$\dot{m}_{bo,A} = \frac{\dot{m}_{ms}\Delta h_{fwh,A}}{h_{hpt,out} - h_{bo,A,out}} \tag{3.51}$$

Boiler\Economizer

By convention, the rate and efficiency for coal plants in the U.S. is based on the higher heating value, HHV [28]. Thus, the rate at which heat is produced by the

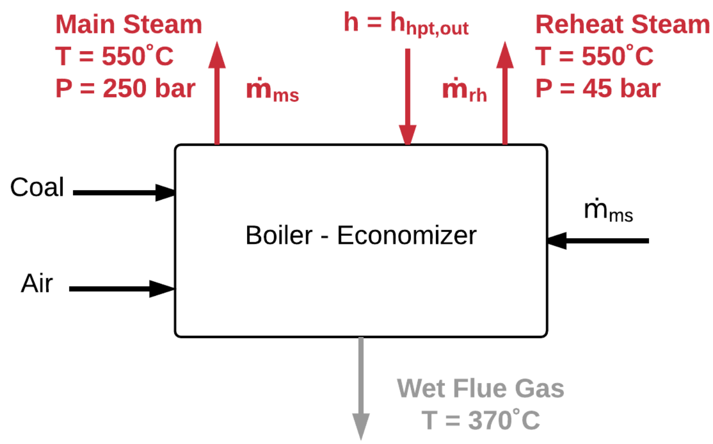

Figure 3.6: Detailed schematic of boiler and economizer as defined in all scenarios

combustion of coal is defined as the product of the mass flow rate and higher heating value of the coal.

$$\dot{Q}_{in} = \dot{m}_{coal} HHV \tag{3.52}$$

Including the expression for the heat input from equation 3.52, equation 3.53 represents an energy rate balance for the boiler/economizer as shown in Figure 3.6.

$$\dot{m}_{coal} HHV + (\dot{m} c_p T_{aph,out})_{air} - (\dot{m} c_p T_{be,out})_{FG} - (\dot{m} h_{be,out})_v - (\dot{m} \Delta h_{be})_{ms} - (\dot{m} \Delta h_{be})_{rh} = 0 \tag{3.53}$$

Where the subscript rh stands for 'reheater'. The changes in enthalpy for the main steam and reheat steam are described in the following equations:

$$\Delta h_{ms,be} = (h_{T=550^\circ C} - h_{T=275^\circ C})_{P=250bar} \tag{3.54}$$

$$\Delta h_{rh,be} = (h_{T=550^\circ C} - h_{hpt,out})_{P=45bar} \tag{3.55}$$

The mass flow rate of the reheat steam, $\dot{m}_{rh}$, is described by the following equation:

$$\dot{m}_{rh} = \dot{m}_{ms} - \dot{m}_{bo,A} \tag{3.56}$$

Based on these equations, the mass flow rate of coal can be determined by rearranging equation 3.53.

$$\dot{m}_{coal} = \frac{(\dot{m}\Delta h_{be})_{ms} + (\dot{m}\Delta h_{be})_{rh}}{HHV + AFR\ (c_p T_{aph,out})_{air} + EFR_{dry}\ (c_p T_{be,out})_{fg} - \phi_{H_2O} h_{v,be,out}} \tag{3.57}$$

Once the mass flow rate of the coal, $\dot{m}_{coal}$, is determined from equation 3.57, the mass flow rates of air, flue gas, and water vapor entering and exiting the boiler can also be calculated with the following equations.

$$\dot{m}_{air} = AFR\ \times \dot{m}_{coal} \tag{3.58}$$

$$\dot{m}_{fg} = EFR_{dry}\ \times \dot{m}_{coal} \tag{3.59}$$

$$\dot{m}_{v} = \phi_{H_2O}\ \times \dot{m}_{coal} \tag{3.60}$$

The mass flow rate of the coal can be used to determine the required power input for the coal pulverizer, $\dot{W}_p$. According to Babcock and Wilcox's *Steam: Its Generation and Use*, the specific power requirement for a coal pulverizer is approximately 15 kWh per ton of coal, including the power required to drive the primary air fans [7]. Thus, the required power input for the coal pulverizer can be determined with the following equation.

$$\dot{W}_p = 15\ \frac{[kWh]}{[ton\ coal]} \times \frac{1}{1000}\ \frac{[ton]}{[kg]} \times \dot{m}_{coal} \tag{3.61}$$

The mass flow rates of air and wet flue gas through the system can be used to determine the required power inputs for the forced and induced draft fans, $\dot{W}_{fd}$ and $\dot{W}_{id}$ respectively. The pressure drop for the forced draft fan is the sum of the pressure drops corresponding to the steam-air heater, the air preheater, the connecting ductwork and the burner, as listed in the Table 3.5. The pressure drop for the induced draft fan is the sum of the pressure drops corresponding to the economizer, the air

Table 3.5: Pressure drops of components related to coal plant air intake and forced draft fan in inches of water

	Δ P
Steam-air Heater	1 [29]
Air Preheater	5 [7]
Burner	5 [29]

Table 3.6: Pressure drops of components related to flue gas exhaust and induced draft fan in inches of water

	Δ P
Economizer	5 [29]
Air Preheater	5 [7]
Electrostatic Precipitator	1 [30]
Flue Gas Cooler	2 [31]
Flue Gas Dryer	2 [29]

preheater, and the electrostatic precipitator as listed in Table 3.6. Note that the flue gas cooler and flue gas dryer discussed in the waste heat and moisture recovery sections will have an effect on the pressure drop associated with the induced draft fan, and so their associated pressure drops are also included in Table 3.6.

Once the total pressure drops associated with the forced and induced draft fans have been determined, the required power inputs can be determined from the following equations.

$$\dot{W}_{fd} = \eta_{fd} \Delta P_{fd} \dot{m}_{air} \tag{3.62}$$

$$\dot{W}_{id} = \eta_{id} \Delta P_{id} (\dot{m}_{fg} + \dot{m}_v) \tag{3.63}$$

Where η_{fd} and η_{id} are both assumed to be 0.9.

Once the mass flow rates of water vapor, $\dot{m}_v$, and dry flue gas, $\dot{m}_{fg}$, through the boiler have been determined, the relative humidity, ω_{be}, of the flue gas leaving the boiler can be determined from the following equation.

$$\omega_{be} = \frac{\dot{m}_v}{\dot{m}_{fg}} \tag{3.64}$$

Intermediate Pressure Turbines

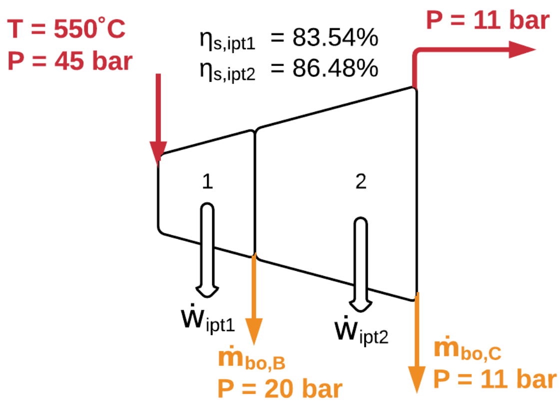

Figure 3.7: Detailed schematic of the intermediate pressure turbines as defined in all scenarios

After being reheated in the boiler/economizer, steam is taken to a series of intermediate pressure turbines. From these turbines, steam is bled off to supply heat to Feedwater Heaters B and C. The calculations for the intermediate pressure turbines are described in the following equations, starting with the power output from the first one, $\dot{W}_{ipt1}$.

$$\dot{W}_{ipt1} = \eta_{s,ipt1}\ \dot{m}_{rh}(h_{in} - h_{out,s})_{ipt1} \tag{3.65}$$

$$h_{ipt1,in} = h_{(T=550°C,\ P=45bar)} \tag{3.66}$$

$$s_{ipt1,in} = s_{(T=550°C,\ P=45bar)} \tag{3.67}$$

$$h_{ipt1,out,s} = h_{(P=20bar,\ s=s_{ipt1,in})} \tag{3.68}$$

$$h_{ipt1,out} = h_{ipt1,in} - \frac{\dot{W}_{ipt1}}{\dot{m}_{ms}} \tag{3.69}$$

The power output for the second intermediate pressure turbine, $\dot{W}_{ipt2}$, is given by the following equation:

$$\dot{W}_{ipt2} = \eta_{s,ipt2}\ \dot{m}_{ip2}(h_{in} - h_{out,s})_{ipt2} \tag{3.70}$$

Where the mass flow rate through the second intermediate pressure turbine, $\dot{m}_{ipt2}$, is equal to the mass flow rate through the first minus the bleed-off to feedwater heater B, $\dot{m}_{bo,B}$.

$$\dot{m}_{ipt2} = \dot{m}_{rh} - \dot{m}_{bo,B} \tag{3.71}$$

$$h_{ipt2,in} = h_{ipt1,out} \tag{3.72}$$

$$s_{ipt2,in} = s_{(h_{ipt2,in},\ P=20bar)} \tag{3.73}$$

$$h_{ipt2,out,s} = h_{(P=11bar,\ s=s_{ipt2,in})} \tag{3.74}$$

$$h_{ipt2,out} = h_{ipt2,in} - \frac{\dot{W}_{ipt2}}{\dot{m}_{ipt2}} \tag{3.75}$$

Feedwater Heater B

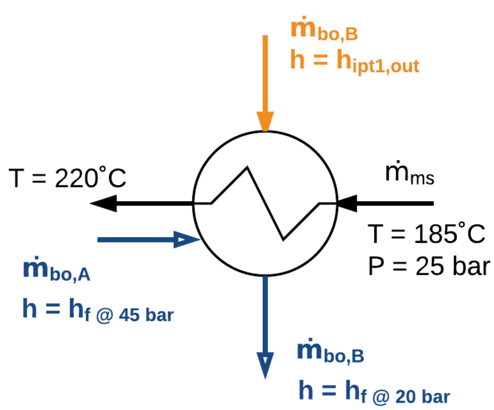

Figure 3.8: Detailed schematic of boiler feedwater heater B as defined in all scenarios

The bleed-off from the first intermediate pressure turbine, $\dot{m}_{bo,B}$, is used to preheat steam feedwater heater B and exits as a saturated liquid at 20 bar.

$$\dot{m}_{bo,B}(h_{ipt1,out} - h_{bo,B,out}) + \dot{m}_{bo,A}(h_{bo,A,out} - h_{bo,B,out}) - \dot{m}_{ms}\Delta h_{fwh,B} = 0 \tag{3.76}$$

$$\Delta h_{fwh,B} = (h_{(T=210°C)} - h_{(T=185°C)})_{P=25bar} \tag{3.77}$$

$$h_{bo,B,out} = h_{(x=0,\ P=20bar)} \tag{3.78}$$

Rearranging equation 3.76 results in the following equation for $\dot{m}_{bo,B}$.

$$\dot{m}_{bo,B} = \frac{\dot{m}_{bo,A}(h_{bo,B,out} - h_{bo,A,out}) + \dot{m}_{ms}\Delta h_{fwh,B}}{h_{ipt1,out} - h_{bo,B,out}} \tag{3.79}$$

Booster Pump

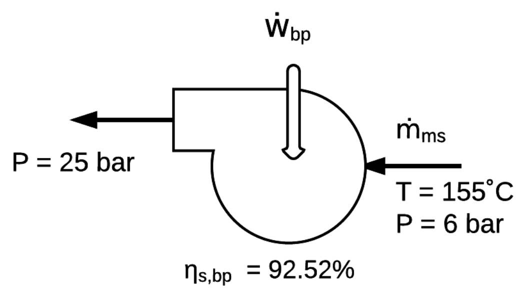

Figure 3.9: Detailed schematic of booster pump as defined in all scenarios

In between the open feedwater heater and feedwater heater C is a booster pump that increases the pressure of the feedwater from 6 to 25 bar. The power input required for the booster pump, $\dot{W}_{bp}$, can be determined from the following equation:

$$\dot{W}_{bp} = \frac{\dot{m}_{ms}}{\eta_{s,bp}}(h_{out,s} - h_{in})_{bp} \tag{3.80}$$

Where the enthalpy and entropy at the pump inlet can be determined from the temperature and pressure.

$$h_{bp,in} = h_{(T=155°C,\ P=6bar)} \tag{3.81}$$

$$s_{bp,in} = s_{(T=155°C,\ P=6bar)} \tag{3.82}$$

The isentropic outlet entropy can then be determined from the inlet entropy and outlet pressure.

$$h_{bp,out,s} = h_{(P=25bar,\ s=s_{bfp,in})} \tag{3.83}$$

The steam enthalpy at the outlet of the booster pump can then be determined from the required power input.

$$h_{bp,out} = h_{bp,in} + \frac{\dot{W}_{bp}}{\dot{m}_{ms}} \tag{3.84}$$

Feedwater Heater C

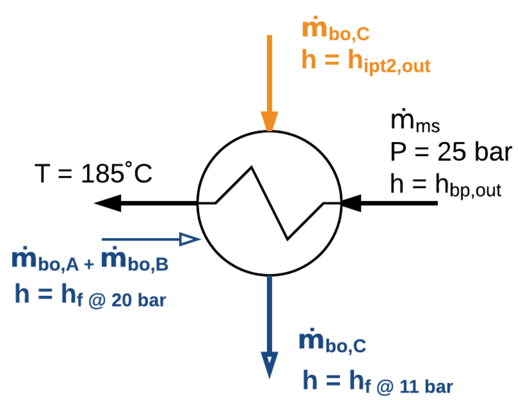

Figure 3.10: Detailed schematic of boiler feedwater heater C as defined in all scenarios

The bleed-off from the second intermediate pressure turbine, $\dot{m}_{bo,C}$, is used to preheat steam feedwater heater C and exits as a saturated liquid at 11 bar.

$$\dot{m}_{bo,C}(h_{ipt2,out} - h_{bo,C,out}) + (\dot{m}_{bo,A} + \dot{m}_{bo,B})(h_{bo,B,out} - h_{bo,C,out}) - \dot{m}_{ms}\Delta h_{fwh,C} = 0 \tag{3.85}$$

$$\Delta h_{fwh,C} = (h_{(T=185°C)} - h_{bp,out}) \tag{3.86}$$

$$h_{bo,C,out} = h_{(x=0,\ P=11bar)} \tag{3.87}$$

Where $h_{bp,out}$ is the enthalpy at the outlet of the booster pump that precedes feedwater heater C. Rearranging equation 3.85 results in the following equation for $\dot{m}_{bo,C}$.

$$\dot{m}_{bo,C} = \frac{(\dot{m}_{bo,A} + \dot{m}_{bo,B})(h_{bo,C,out} - h_{bo,B,out}) + \dot{m}_{ms}\Delta h_{fwh,C}}{h_{ipt2,out} - h_{bo,C,out}} \tag{3.88}$$

Low Pressure Turbines

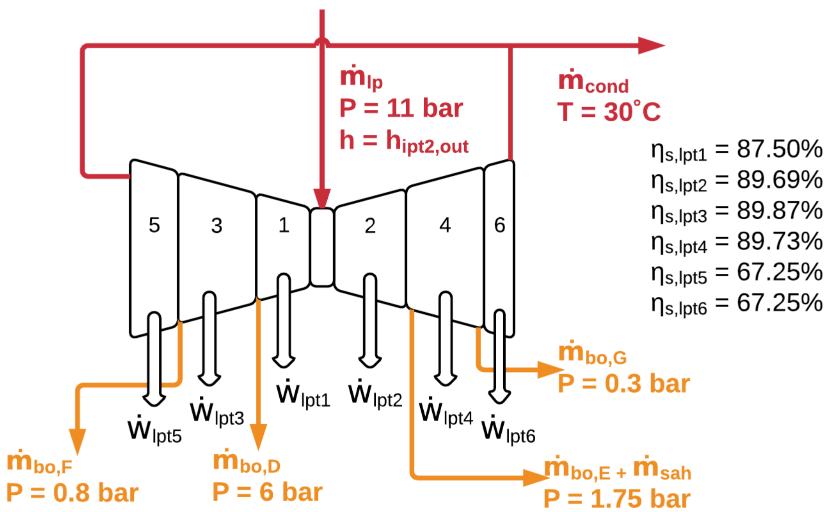

Figure 3.11: Detailed schematic of low pressure turbines as defined in all scenarios

Without being reheated, the fluid exiting the second intermediate turbine, minus the amount bled-off for feedwater heater C is, pumped through a series of low pressure turbines as shown in Figure 3.11. That flow is then split in half between the odd and even numbered low pressure turbines, and after each turbine, some steam is bled-off to a feedwater heater. The remaining flow of steam, $\dot{m}_{cond}$, is then sent to the condenser

$$\dot{m}_{lpt} = \dot{m}_{ipt2} - \dot{m}_{bo,C} \tag{3.89}$$

$$\dot{m}_{lpt} = \dot{m}_{bo,D} + \dot{m}_{bo,E} + \dot{m}_{bo,F} + \dot{m}_{bo,G} + \dot{m}_{cond} \tag{3.90}$$

The calculations for each of the low pressure turbines are described in the following equations, starting with the power output for the first one, $\dot{W}_{lpt1}$.

$$\dot{W}_{lpt1} = \eta_{s,lpt1}\ \dot{m}_{lpt1}(h_{in} - h_{out,s})_{lpt1} \tag{3.91}$$

Where the mass flow rate through the first low pressure turbine, $\dot{m}_{lpt1}$, is half of the flow rate entering the series of low pressure turbines.

$$\dot{m}_{lpt1} = \frac{1}{2}\dot{m}_{lpt} \tag{3.92}$$

$$h_{lpt1,in} = h_{ipt2,out} \tag{3.93}$$

$$s_{lpt1,in} = s_{(h_{lpt1,in},\ P=11bar)} \tag{3.94}$$

$$h_{lpt1,out,s} = h_{(P=6bar,\ s=s_{lpt1,in})} \tag{3.95}$$

$$h_{lpt1,out} = h_{lpt1,in} - \frac{\dot{W}_{lpt1}}{\dot{m}_{lpt1}} \tag{3.96}$$

The power output for the second low pressure turbine, $\dot{W}_{lpt2}$, is given by the following equation.

$$\dot{W}_{lpt2} = \eta_{s,lpt2}\ \dot{m}_{lpt2}(h_{in} - h_{out,s})_{lpt2} \tag{3.97}$$

Where the mass flow rate through the second low pressure turbine, $\dot{m}_{lpt2}$, is half of the flow rate entering the series of low pressure turbines.

$$\dot{m}_{lpt2} = \frac{1}{2}\dot{m}_{lpt} \tag{3.98}$$

$$h_{lpt2,in} = h_{ipt2,out} \tag{3.99}$$

$$s_{lpt2,in} = s_{(h_{lpt2,in},\ P=11bar)} \tag{3.100}$$

$$h_{lpt2,out,s} = h_{(P=1.75bar,\ s=s_{lpt2,in})} \tag{3.101}$$

$$h_{lpt2,out} = h_{lpt2,in} - \frac{\dot{W}_{lpt2}}{\dot{m}_{lpt2}} \tag{3.102}$$

The power output for the third low pressure turbine, $\dot{W}_{lpt3}$, is given by the following equation.

$$\dot{W}_{lpt3} = \eta_{s,lpt3}\ \dot{m}_{lpt3}(h_{in} - h_{out,s})_{lpt3} \tag{3.103}$$

Where the mass flow rate through the third low pressure turbine, $\dot{m}_{lpt3}$, is equal to the flow rate through the first turbine minus the bleed-off to the open feedwater heater, $\dot{m}_{bo,D}$.

$$\dot{m}_{lpt3} = \dot{m}_{lpt1} - \dot{m}_{bo,D} \tag{3.104}$$

$$h_{lpt3,in} = h_{ipt1,out} \tag{3.105}$$

$$s_{lpt3,in} = s_{(h_{lpt3,in},\ P=6bar)} \tag{3.106}$$

$$h_{lpt3,out,s} = h_{(P=0.8bar,\ s=s_{lpt3,in})} \tag{3.107}$$

$$h_{lpt3,out} = h_{lpt3,in} - \frac{\dot{W}_{lpt3}}{\dot{m}_{lpt3}} \tag{3.108}$$

The power output for the fourth low pressure turbine, $\dot{W}_{lpt4}$, is given by the following equation.

$$\dot{W}_{lpt4} = \eta_{s,lpt4}\ \dot{m}_{lpt4}(h_{in} - h_{out,s})_{lpt4} \tag{3.109}$$

Where the mass flow rate through the fourth low pressure turbine, $\dot{m}_{lpt4}$, is equal to the flow rate through the second turbine minus the bleed-off to feedwater heater E, $\dot{m}_{bo,E}$, and the steam-air heater, $\dot{m}_{sah}$.

$$\dot{m}_{lpt4} = \dot{m}_{lpt2} - \dot{m}_{bo,E} - \dot{m}_{sah} \tag{3.110}$$

$$h_{lpt4,in} = h_{ipt2,out} \tag{3.111}$$

$$s_{lpt4,in} = s_{(h_{lpt4,in},\ P=1.75bar)} \tag{3.112}$$

$$h_{lpt2,out,s} = h_{(P=0.3bar,\ s=s_{lpt2,in})} \tag{3.113}$$

$$h_{lpt4,out} = h_{lpt2,in} - \frac{\dot{W}_{lpt4}}{\dot{m}_{lpt4}} \tag{3.114}$$

The power output for the fifth low pressure turbine, $\dot{W}_{lpt5}$, is given by the following equation.

$$\dot{W}_{lpt5} = \eta_{s,lpt5}\ \dot{m}_{lpt5}(h_{in} - h_{out,s})_{lpt5} \tag{3.115}$$

Where the mass flow rate through the fifth low pressure turbine, $\dot{m}_{lpt5}$, is equal to the flow rate through the third turbine minus the bleed-off to feedwater heater F, $\dot{m}_{bo,F}$.

$$\dot{m}_{lpt5} = \dot{m}_{lpt3} - \dot{m}_{bo,E} \tag{3.116}$$

$$h_{lpt5,in} = h_{lpt3,out} \tag{3.117}$$

$$s_{lpt5,in} = s_{(h_{lpt5,in},\ P=0.8bar)} \tag{3.118}$$

The outlet pressure for the fifth low pressure turbine is equal to the saturation pressure corresponding to the condenser temperature, which according to this model is 30°C.

$$h_{lpt5,out,s} = h_{(P_{sat\ @T=30°C},\ s=s_{lpt3,in})} \tag{3.119}$$

$$h_{lpt5,out} = h_{lpt5,in} - \frac{\dot{W}_{lpt5}}{\dot{m}_{lpt5}} \tag{3.120}$$

The power output for the sixth low pressure turbine, $\dot{W}_{lpt6}$, is given by the following equation.

$$\dot{W}_{lpt6} = \eta_{s,lpt6}\ \dot{m}_{lpt6}(h_{in} - h_{out,s})_{lpt6} \tag{3.121}$$

Where the mass flow rate through the sixth low pressure turbine, $\dot{m}_{lpt6}$, is equal to the flow rate through the fourth turbine minus the bleed-off to feedwater heater G, $\dot{m}_{bo,G}$.

$$\dot{m}_{lpt6} = \dot{m}_{lpt4} - \dot{m}_{bo,G} \tag{3.122}$$

$$h_{lpt6,in} = h_{lpt4,out} \tag{3.123}$$

$$s_{lpt6,in} = s_{(h_{lpt4,in},\ P=0.3bar)} \tag{3.124}$$

The outlet pressure for the sixth low pressure turbine is equal to the saturation pressure corresponding to the condenser temperature, which according to this model is 30°C.

$$h_{lpt68,out,s} = h_{(P_{sat\ @T=30°C},\ s=s_{lpt4,in})} \tag{3.125}$$

$$h_{lpt6,out} = h_{lpt6,in} - \frac{\dot{W}_{lpt6}}{\dot{m}_{lpt6}} \tag{3.126}$$

Open Feedwater Heater

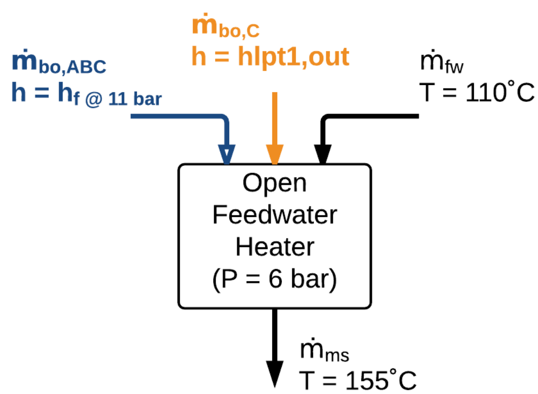

Figure 3.12: Detailed schematic of the open feedwater heater as defined in all scenarios

The bleed-off from the first low pressure turbine, $\dot{m}_D$, is injected into an open feedwater heater along with the bleed-off from feedwater heaters A, B, and C, to pre-heat feedwater.

$$\dot{m}_{bo,D}(h_{lpt1,out} - h_{ofwh,out}) + \dot{m}_{bo,ABC}(h_{bo,C,out} - h_{ofwh,out}) + \dot{m}_{fw}(h_{fwhE,out} - h_{ofwh,out}) = 0 \tag{3.127}$$

Where $\dot{m}_{ob,ABC}$ is the sum of the bleed-off from feedwater heaters A, B, and C, and the feedwater flow-rate, $\dot{m}_{fw}$, is equal to the flow rate of steam at the inlet to the low pressure turbines.

$$\dot{m}_{bo,ABC} = \dot{m}_{bo,A} + \dot{m}_{bo,B} + \dot{m}_{bo,C} \tag{3.128}$$

$$\dot{m}_{fw} = \dot{m}_{lpt} - \dot{m}_{bo,D} \tag{3.129}$$

The outlet enthalpy of the open feedwater heater and of feedwater heater E can be determined from their temperatures, 155 °C and 110 °C respectively, and the feedwater pressure, 6 bar.

$$h_{ofwh,out} = h_{(T=155°C,\ P=6bar)} \tag{3.130}$$

$$h_{fwhE,out} = h_{(T=110°C,\ P=6bar)} \tag{3.131}$$

Rearranging equation 3.127 results in the following equation for $\dot{m}_{bo,D}$.

$$\dot{m}_{bo,D} = \frac{\dot{m}_{bo,ABC}(h_{ofwh,out} - h_{bo,C,out}) + \dot{m}_{lpt}(h_{ofwh,out} - h_{fwhE,out})}{h_{lpt1,out} - h_{fwhE,out}} \tag{3.132}$$

Steam-air Heater

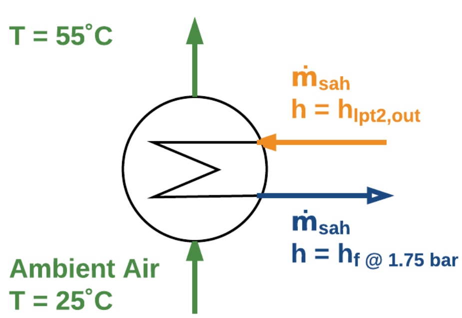

Figure 3.13: Detailed schematic of the steam-air heater as defined in the baseline model and scenarios B and C

Some of the bleed-off from the second low pressure turbine, $\dot{m}_{sah}$, where 'sah' stands for steam-air heater, is used to preheat combustion air from the ambient temperature, assumed to be 25 °C, to 55 °C and exits as a saturated liquid at 1.75 bar.

$$\dot{m}_{sah}(h_{lpt2,out} - h_{sah,out}) - (\dot{m}c_p\Delta T)_{air,sah} = 0 \tag{3.133}$$

$$h_{sah,out} = h_{(x=0,\ P=1.75bar)} \tag{3.134}$$

Rearranging equation 3.133 results in the following equation for $\dot{m}_{sah}$.

$$\dot{m}_{sah} = \frac{(\dot{m}c_p\Delta T)_{air,sah}}{h_{lpt2,out} - h_{sah,out}} \tag{3.135}$$

Feedwater Heater E

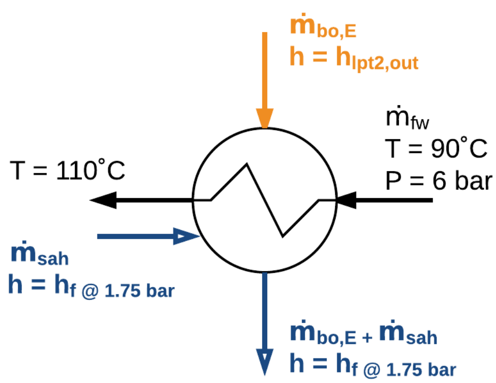

Figure 3.14: Detailed schematic of feedwater heater E as defined for all scenarios

Some of the bleed-off from the second low pressure turbine, $\dot{m}_{bo,E}$, is used to preheat feedwater in feedwater heater E and exits as a saturated liquid at 1.75 bar.

$$\dot{m}_{bo,E}(h_{lpt2,out} - h_{bo,E,out}) + \dot{m}_{sah}(h_{sah,out} - h_{bo,E,out}) - \dot{m}_{fw}\Delta h_{fwhE} = 0 \tag{3.136}$$

$$\Delta h_{fwhE} = (h_{(T=110^\circ C)} - h_{(T=90^\circ C)})_{P=6bar} \tag{3.137}$$

$$h_{bo,E,out} = h_{(x=0,\ P=1.75bar)} \tag{3.138}$$

Note that $h_{sah,out} = h_{bo,E,out}$. Rearranging equation 3.136 results in the following equation for $\dot{m}_{bo,E}$.

$$\dot{m}_{bo,E} = \frac{\dot{m}_{fw}\Delta h_{fwhE}}{h_{lpt2,out} - h_{bo,E,out}} \tag{3.139}$$

Feedwater Heater F

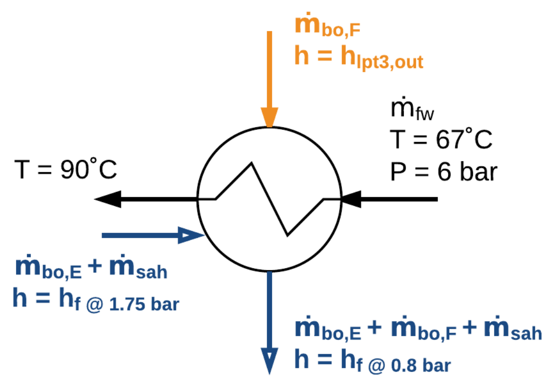

Figure 3.15: Detailed schematic of feedwater heater F as defined for the baseline model and scenarios A and B

The bleed-off from the third low pressure turbine, $\dot{m}_{bo,F}$, is used to preheeat feedwater in feedwater heater F and exits as a saturated liquid at 0.8 bar.

$$\dot{m}_{bo,F}(h_{lpt3,out} - h_{bo,F,out}) + \dot{m}_{E,sah}(h_{bo,E,out} - h_{bo,F,out}) - \dot{m}_{fw}\Delta h_{fwhF} = 0 \tag{3.140}$$

$$\Delta h_{fwhF} = (h_{(T=90°C)} - h_{(T=67°C)})_{P=6bar} \tag{3.141}$$

$$\dot{m}_{E,sah} = \dot{m}_{sah} + \dot{m}_{bo,E} \tag{3.142}$$

$$h_{bo,F,out} = h_{(x=0,\ P=0.8bar)} \tag{3.143}$$

Rearranging equation 3.140 results in the following equation for $\dot{m}_{bo,F}$.

$$\dot{m}_{bo,F} = \frac{\dot{m}_{E,sah}(h_{bo,F,out} - h_{bo,E,out}) + \dot{m}_{fw}\Delta h_{fwhF}}{h_{lpt3,out} - h_{bo,F,out}} \tag{3.144}$$

Feedwater Heater G and Drain Pump

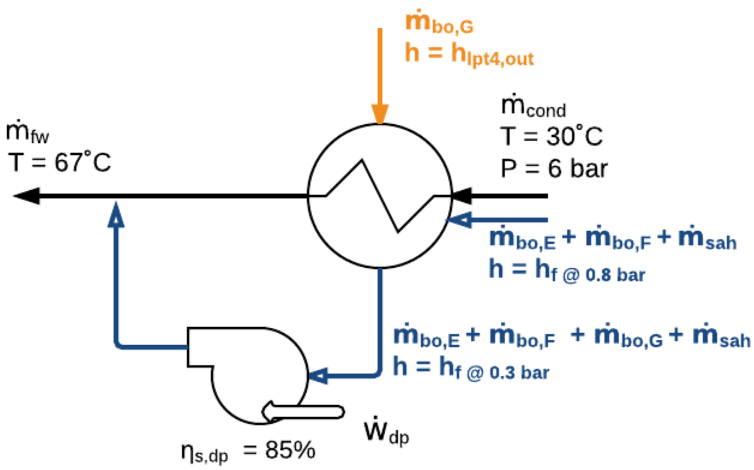

Figure 3.16: Detailed schematic of feedwater heater G as defined for the baseline model and scenario A

The bleed-off from the fourth low pressure turbine, $\dot{m}_{bo,G}$ is used to preheat feedwater in feedwater heater G. The drain pump used to pump the bleed-off exiting feedwater G into the main feedwater line complicates the energy rate balance. Thus, to calculate $\dot{m}_{bo,G}$, the control volume must include both feedwater heater G and the drain pump.

$$\dot{m}_{bo,G}h_{lpt4,in} + \dot{m}_{EF,sah}h_{bo,F,out} + \dot{m}_{cond}h_{fwhG,in} - \dot{m}_{fw}h_{fwhF,in} + \dot{W}_{dp} = 0 \tag{3.145}$$

Where $\dot{m}_{EF,sah}$ is the sum of $\dot{m}_{E,sah}$ and $\dot{m}_{bo,F}$, and $\dot{m}_{cond}$ is equal to $\dot{m}_{fw}$ minus $\dot{m}_{EF,sah}$ and $\dot{m}_{bo,G}$

$$\dot{m}_{EF,sah} = \dot{m}_{E,sah} + \dot{m}_{bo,F} \tag{3.146}$$

$$\dot{m}_{cond} = \dot{m}_{fw} - \dot{m}_{EF,sah} - \dot{m}_{bo,G} \tag{3.147}$$

The enthalpies and entropies can be determine from the inlet and outlet conditions of the feedwater and bleed-off.

$$h_{fwh,G,in} = h_{(T=30.06°C,\ P=6bar)} \tag{3.148}$$

$$h_{bo,G,out} = h_{(x=0\ \ P=0.3bar)} \tag{3.149}$$

$$s_{dp,in} = s_{(h_{bo,G,out},\ P=0.3bar)} \tag{3.150}$$

$$h_{dp,out,s} = h_{(s_{dp,in},\ P=0.3bar)} \tag{3.151}$$

The required power input for the drain pump depend on the mass flow rate through the drain pump and its isentropic efficiency.

$$\dot{W}_{dp} = \frac{\dot{m}_{EF,sah} + \dot{m}_{bo,G}}{\eta_{s,dp}}(h_{out,s} - h_{bo,G,out}) \tag{3.152}$$

Rearranging equation 3.145 results in the following equation for $\dot{m}_{bo,G}$.

$$\dot{m}_{bo,G} = \frac{\dot{m}_{EF,sah}[h_{fwh,G,in} - h_{bo,F,out} + \frac{1}{\eta_{s,dp}}(h_{bo,G,out} - h_{dp,out,s})] + \dot{m}_{fw}(h_{fwh,F,in} - h_{fwh,G,in})}{h_{bo,G,in} - h_{fwh,G,in} + \frac{1}{\eta_{dp}}(h_{dp,out,s} - h_{bo,G,out})} \tag{3.153}$$

Condensate Pump

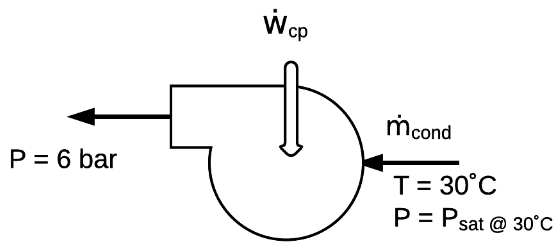

Figure 3.17: Detailed schematic of the condensate pump as defined for all scenarios

Between feedwater heater G and the condenser, water is pumped to the saturation pressure corresponding to the condenser temperature to 6 bar. The required power input, $\dot{W}_{cp}$, can be determined from the following equations.

$$\dot{W}_{cp} = \dot{m}_{cond}(h_{fwh,G,in} - h_{cond,out}) \tag{3.154}$$

$$h_{cond,out} = h_{(x=0,\ P_{sat@T=30^{\circ}C})} \tag{3.155}$$

Condenser and Cooling Tower

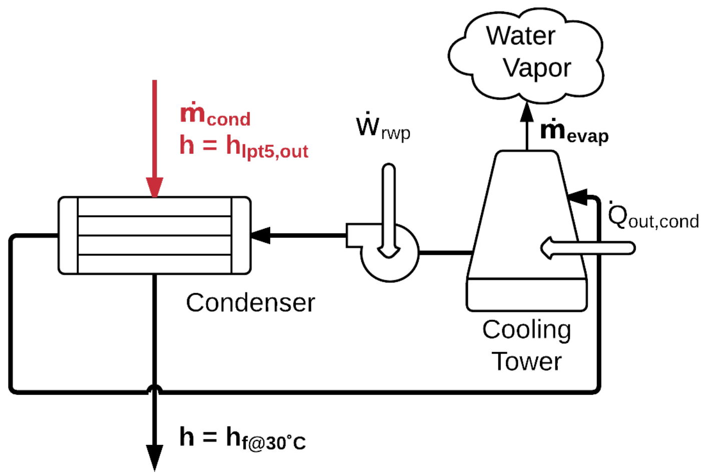

Figure 3.18: Detailed schematic of the cooling as defined for all scenarios

Once the remaining steam has exited the last low pressure turbines, it is sent through the condenser. The rate of heat rejection to the condenser, $\dot{Q}_{out,cond}$, can be determined by the following equation.

$$\dot{Q}_{out,cond} = \dot{m}_{cond}(h_{lp6,out} - h_{cond,out}) \tag{3.156}$$

That heat is then rejected to the recirculating water from the cooling tower. The power input required for the recirculating water pump, shown as $\dot{W}_{rwp}$ on Figure 3.18, is based on the pressure drop through the cooling loop, ΔP_{cl}, the flow rate of the recirculating water, $\dot{V}_{rw}$, and the pump efficiency, η_{rwp}, as shown in the following equation.

$$\dot{W}_{rwp} = \eta_{rwp}\dot{V}_{rw}\Delta P_{cl} \tag{3.157}$$

The recirculating water pump is assumed to be 90% efficient. The flow rate can be determined based on a simple rule of thumb — three gallons per minute of recirculating water per ton of cooling load for the condenser [32]. One ton of cooling is approximately equivalent to 3.515 kW, and there are approximately 264.172 gallons in a cubic meter. Including those conversion factors, the flow rate of the recirculating water can be determined from the following equation.

$$\dot{V}_{rw} = 3\frac{[\frac{gal}{min}]}{[ton\ cooling]} \times \frac{1}{3.517}\frac{[ton\ cooling]}{[kW]} \times \frac{1}{261.172}\frac{[m^3]}{[gal]} \times \frac{1}{60}\frac{[min]}{[sec]} \times \dot{Q}_{out,cond} \tag{3.158}$$

The pressure drop in the cooling loop is based on the sum of the pressure drops in the lines going to and from the cooling tower and condenser, the pressure drop of the recirculating water through the condenser itself, and the elevation difference between the top and bottom of the cooling tower, each of which are shown in Figure 3.19.

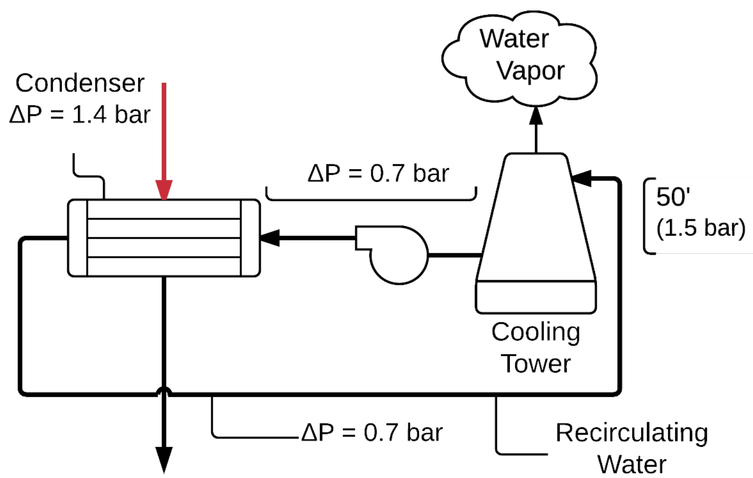

Figure 3.19: Schematic of condenser as defined for all scenarios

The total pressure drop, approximately 4 bar, along with the flow rate, can be plugged into equation 3.157 to determine the necessary power input for the recirculating water pump. The heated up recirculating water goes through the cooling tower and rejects the waste heat from the condenser to the environment through evaporation. That process is covered in detail in the water consumption and recovery section.

3.3.1 Scenarios for Low Temperature Waste Heat Recovery

The majority of the components and operating parameters remain the same for each of the scenarios for low temperature waste heat recovery. This section includes the schematics for power plant models similar to the baseline model except that a flue gas cooler is used in place of one or more of the components. For each scenario, the schematic of the modified power plant model is followed by a detailed schematic of the flue gas cooler and associated fluid flows.

Scenario A

A schematic of the supercritical coal plant with where the steam-air heater has been replaced by a flue gas cooler is shown in Figure 3.20, and a detailed schematic showing the inlet and outlet temperatures of the flue gas cooler is shown in Figure 3.21.

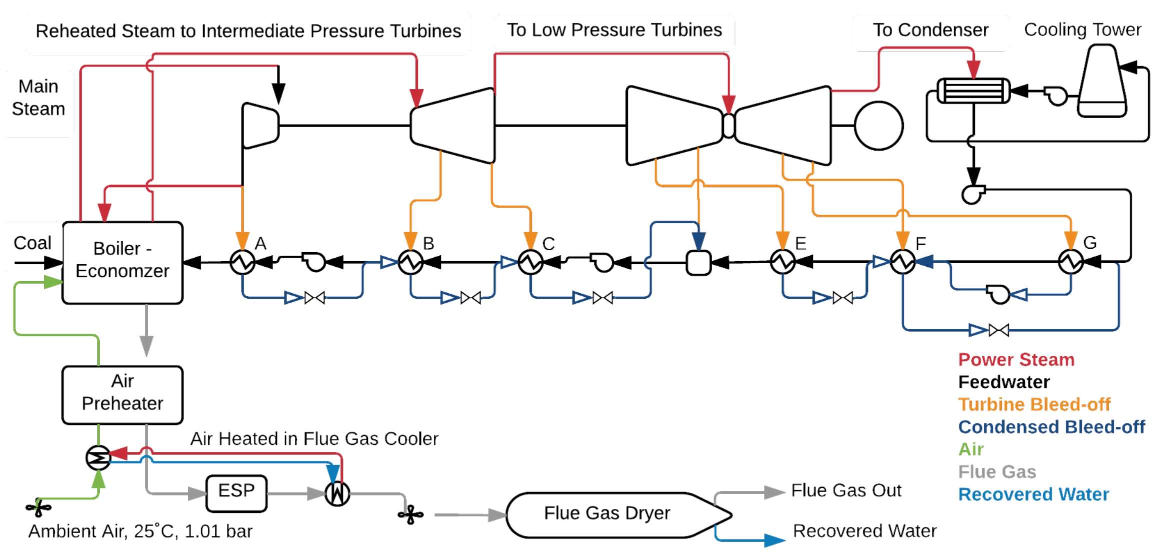

Figure 3.20: Schematic for supercritical coal plant, Scenario A, wherein the steam-air heater is replaced with a flue gas cooler

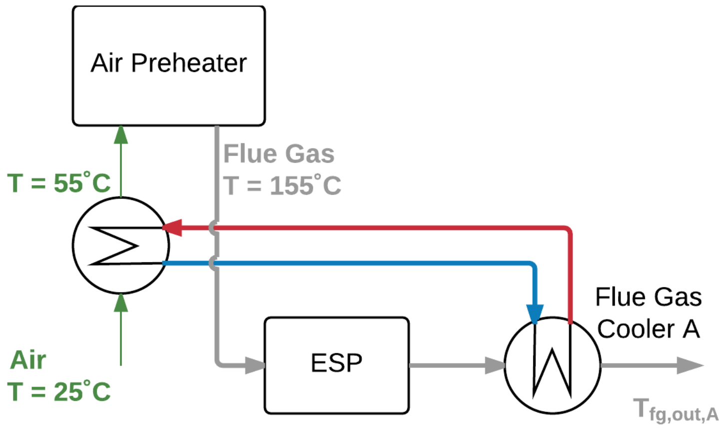

Figure 3.21: Detailed schematic of the flue gas cooler in scenario A

For scenario A, the steam-air heater is replaced with a flue gas cooler. Thus, $\dot{m}_{sah} = 0$. The rate at which waste heat is removed from the flue gas can be calculated from the mass flow rate, specific heat, and temperature change of the air through the flue gas cooler.

$$\dot{Q}_{whr,A} = (\dot{m}c_p\Delta T)_{air,fgc,A} \tag{3.159}$$

At the same time, the rate of waste heat recovery can be defined in terms of the change in temperature of the flue gas and associated water vapor.

$$\dot{Q}_{whr,A} = (\dot{m}c_p\Delta T)_{fg,fgc,A} + \dot{m}_{H_2O}(h_{v,T=155^\circ C} - h_{v,T_{fg,out,A}}) \tag{3.160}$$

Thus, $T_{fg,out,A}$ can be found by setting equations 3.159 and 3.160 equal to each other. To determine the minimum effectiveness of the flue gas cooler, the actual rate of waste heat recovery can be compared to the maximum rate of heat recovery. The maximum rate of heat recovery would result in the flue gas exiting the flue gas cooler at 25 °C, the inlet temperature of the air being preheated.

$$\dot{Q}_{whr,A,max} = \dot{m}_{fg}[c_{p,fg}(155^\circ C - 25^\circ C) + \omega_{be}h_{v,155^\circ C} - (\omega_{be} - \omega_{min,A})h_{w,25^\circ C} - \omega_{min,A}h_{v,25^\circ C}] \tag{3.161}$$

Where h_w refers to the enthalpy of the water that has condensed out of the flue gas, ω_{be} is the relative humidity of the flue gas leaving the boiler, and $\omega_{min,A}$ is the relative humidity of flue gas at 25 °C as determined from equations 3.27 and 3.28. Once the maximum rate of waste heat recovery is known, the minimum effectiveness of the flue gas cooler can be determined from the following equation.

$$\epsilon_{fgc,A} = \frac{\dot{Q}_{whr,A}}{\dot{Q}_{whr,A,max}} \tag{3.162}$$

Scenario B

A schematic of the supercritical coal plant where feedwater heater G has been replaced by a flue gas cooler is shown in Figure 3.22, and a detailed schematic showing the inlet and outlet temperatures of the flue gas cooler is shown in Figure 3.23.

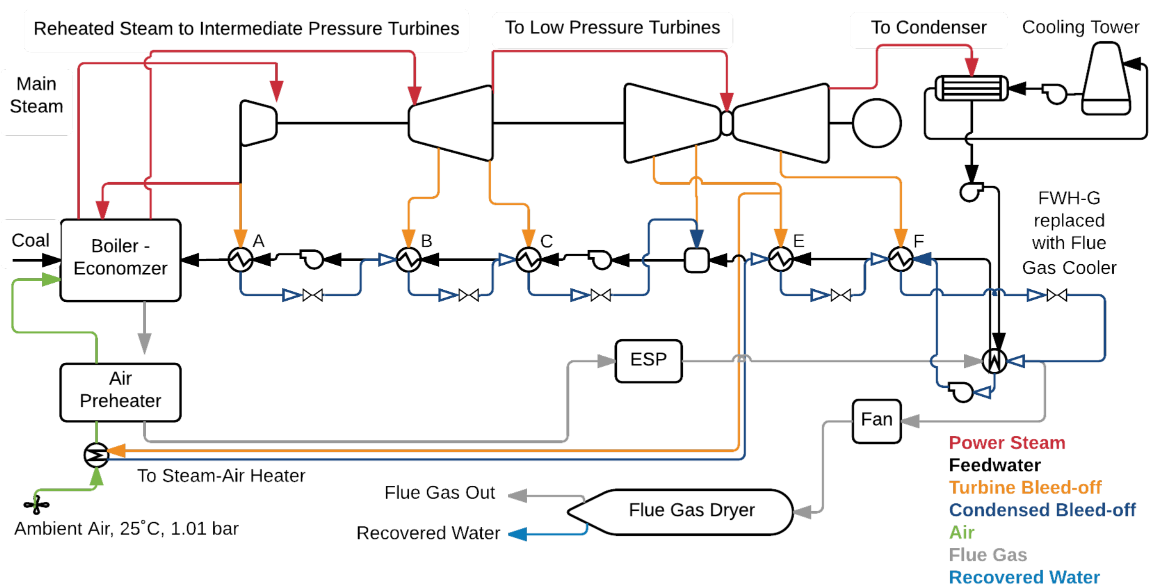

Figure 3.22: Schematic for supercritical coal plant, Scenario B, wherein FWH-G has been replaced with a flue gas cooler

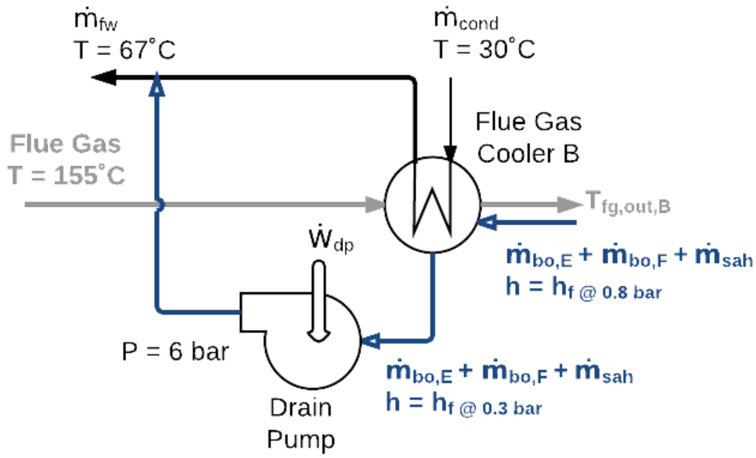

Figure 3.23: Detailed schematic of the flue gas cooler in scenario B

For scenario B, feedwater heater G is replaced with a flue gas cooler. Thus, $\dot{m}_{bo,G} = 0$. The rate at which waste heat is recovered from the flue gas can be calculated from the mass flow rate and temperature change of the feedwater through the flue gas cooler, taking into account the work input to the drain pump.

$$\dot{Q}_{whr,B} = \dot{m}_{fw}(h_{fwh,F,in} - h_{fwh,G,in}) + \dot{m}_{EF,sah}(h_{fwh,G,in} - h_{bo,F,out} - \frac{1}{\eta_{dp}}(h_{dp,in} - h_{dp,out,s})) \tag{3.163}$$

At the same time, the rate of waste heat recovery can be defined in terms of the change in temperature of the flue gas and associated water vapor.

$$\dot{Q}_{whr,B} = (\dot{m}c_p\Delta T)_{fg,fgc,B} + \dot{m}_{H_2O}(h_{v,T=155^\circ C} - h_{v,T_{fg,out,b}}) \tag{3.164}$$

Thus, $T_{fg,out,B}$ can be found by setting equations 3.163 and 3.164 equal to each other. To determine the minimum effectiveness of the flue gas cooler, the actual rate of waste heat recovery can be compared to the maximum rate of heat recovery. The maximum rate of heat recovery would result in the flue gas exiting the flue gas cooler at 30 °C, the inlet temperature of the feedwater being preheated.

$$\dot{Q}_{whr,B,max} = \dot{m}_{fg}[c_{p,fg}(155°C - 30°C) + \omega_{be}h_{v,155°C} - (\omega_{be} - \omega_{min,B})h_{w,30°C} - \omega_{min,B}h_{v,30°C}] \tag{3.165}$$

Where $\omega_{min,B}$ is the relative humidity of flue gas at 30 °C as determined from equations 3.27 and 3.28. Once the maximum rate of waste heat recovery is known, the minimum effectiveness of the flue gas cooler can be determined from the following equation.

$$\epsilon_{fgc,B} = \frac{\dot{Q}_{whr,B}}{\dot{Q}_{whr,B,max}} \tag{3.166}$$

Scenario C

A schematic of the supercritical coal plant where feedwater heaters F and G have been replaced by a flue gas cooler is shown in Figure 3.24, and a detailed schematic showing the inlet and outlet temperatures of the flue gas cooler is shown in Figure 3.25.

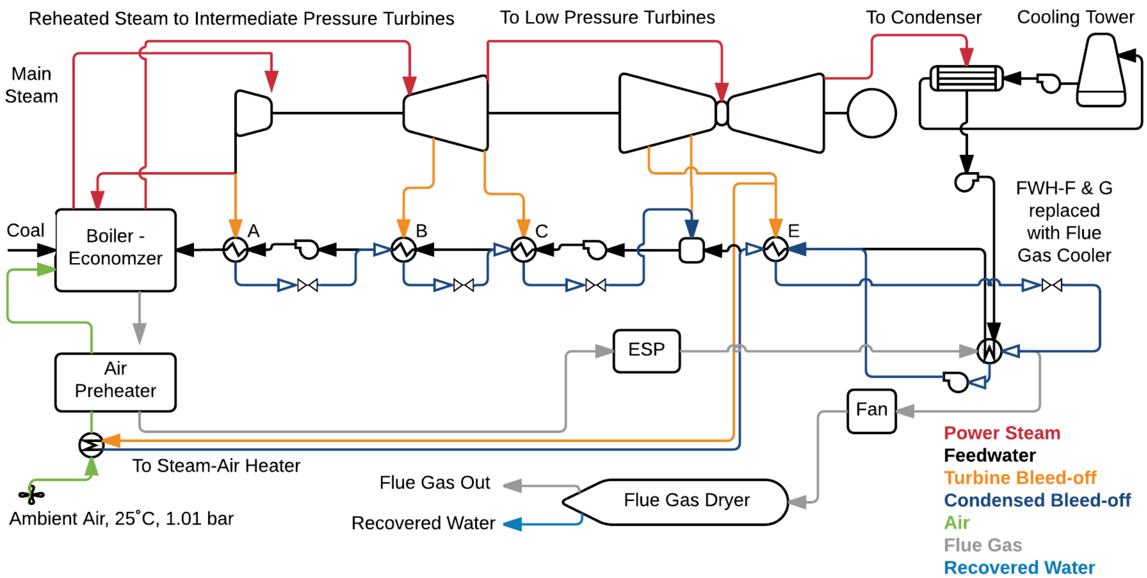

Figure 3.24: Schematic for supercritical coal plant, Scenario C, wherein FWH-F and G has been replaced with a flue gas cooler

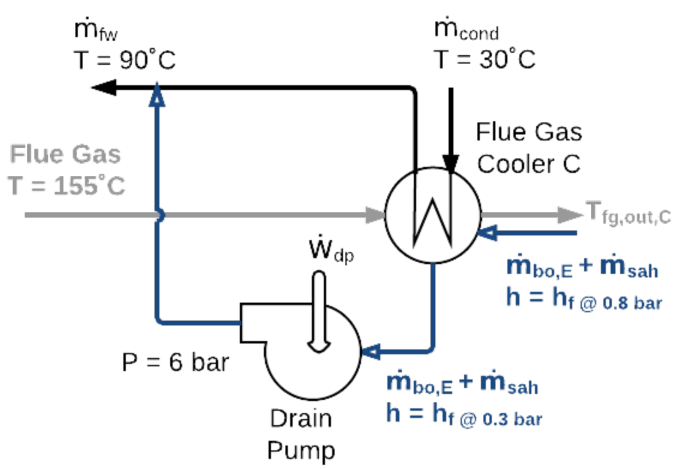

Figure 3.25: Detailed schematic of the flue gas cooler in scenario C

For scenario C, feedwater heaters F and G are replaced with a flue gas cooler. Thus, $\dot{m}_{bo,G} = \dot{m}_{bo,F} = 0$. The rate at which waste heat has to be recovered to replace feedwater heater F, $\dot{Q}_{whr,fwhF}$, can be calculated by modifying equation 3.140 as follows:

$$\dot{Q}_{whr,fwhF} = \dot{m}_{fw}\Delta h_{fwhF} + \dot{m}_{E,sah}(h_{bo,F,out} - h_{bo,E,out}) \tag{3.167}$$

The rate at which waste heat has to be recovered to replace feedwater heater G, $\dot{Q}_{whr,fwhG}$, can be calculated from equation 3.163. The total rate of waste heat recovery for scenario C, $\dot{Q}_{whr,C}$, is the sum of $\dot{Q}_{whr,fwhF}$ and $\dot{Q}_{whr,fwhG}$.

$$\dot{Q}_{whr,C} = \dot{Q}_{whr,fwhF} + \dot{Q}_{whr,fwhG} \tag{3.168}$$

At the same time, the rate of waste heat recovery can be defined in terms of the change in temperature of the flue gas and associated water vapor. However, unlike scenarios A and B, scenario C requires the extraction of both latent and sensible heat from the flue gas. It is simpler to split the waste heat recovery rate into two equations, $\dot{Q}_{whr,C,1}$, the heat recovery rate required to cool the flue gas to its dew point and $\dot{Q}_{whr,C,2}$, the heat recovery rate when water vapor starts condensing out of the flue gas.

$$\dot{Q}_{whr,C,1} = \dot{m}_{fg}[c_{p,fg}(155°C - T_{dew}) + \dot{m}_{H_2O}(h_{v,T=155°C} - h_{v,T_{dew}})] \tag{3.169}$$

$$\dot{Q}_{whr,C,2} = \dot{m}_{fg}[c_{p,fg}(T_{dew} - T_{fg,out,C}) + \omega_{be}h_{v,T_{dew}} - (\omega_{be} - \omega_{T_{fg,out,C}})h_{w,T_{dew}} - \omega_{T_{fg,out,C}}h_{v,T_{fg,out,C}}] \tag{3.170}$$

$$\dot{Q}_{whr,C} = \dot{Q}_{whr,C,1} + \dot{Q}_{whr,C,2} \tag{3.171}$$

Thus, $T_{fg,out,C}$ can be determined by setting equations 3.168 and 3.171 equal to each other. The condensed water from the flue gas cooler is the difference between the relative humidities at the inlet and exit of the cooler.

$$\dot{m}_{cond,fgc} = \dot{m}_{fg}(\omega_{be} - \omega_{T_{fg,out,C}}) \tag{3.172}$$

To determine the minimum effectiveness of the flue gas cooler, the actual rate of waste heat recovery can be compared to the maximum rate of heat recovery, which is the same for scenarios B and C.

$$\dot{Q}_{whr,C,max} = \dot{Q}_{whr,B,max} \tag{3.173}$$

Once the maximum rate of waste heat recovery is known, the minimum effectiveness of the flue gas cooler can be determined from the following equation.

$$\epsilon_{fgc,C} = \frac{\dot{Q}_{whr,C}}{\dot{Q}_{whr,C,max}} \tag{3.174}$$

3.3.2 Flue Gas Dryer

Upon exiting the ESP in the baseline case or the flue gas cooler for scenarios A, B, and C, the flue gas goes through a condensing heat exchanger called a flue gas dryer[1]. The flue gas dryer is a heat exchanger between the flue gas and ambient air used to cool the flue gas for the purpose of recovering some its moisture as illustrated in Figure 3.26.

[1]For a newly built coal plant, the flue gas would likely go through a flue gas scrubber before going through the flue gas dryer [29]. Use of a flue gas cooler before the scrubber would lower the temperature of the flue gas which would lower the rate at which water evaporates in the scrubber. The flue gas would then exit the scrubber at or near the adiabatic saturation temperature with 100% relative humidity [7] before entering the flue gas dryer. A scrubber would have a significant effect on the quality of the water recovered from the flue gas dryer.

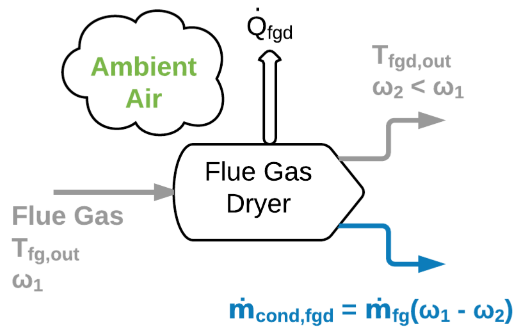

Figure 3.26: Detailed schematic of the flue gas dryer generalized for all scenarios

The rate at which the the flue gas rejects heat to the environment, $\dot{Q}_{fgd}$, is given by the following equation:

$$\dot{Q}_{fgd} = \dot{m}_{fg}[c_{p,fg}(T_{fg,out,i} - T_{fgd,out}) + \omega_1 h_{v,T_{fg,out}} - (\omega_1 - \omega_2)h_{w,T_{fgd,out}} - \omega_2 h_{v,T_{fgd,out}}] \tag{3.175}$$

Where ω_1, the relative humidity at the inlet of the flue gas dryer, is equal to ω_{be} for all of the scenarios except for scenario C. For scenario C, $\omega_1 = \omega_{T_{fg,out,C}}$. The temperature and relative humidity at the outlet of the flue gas dryer, $T_{fgd,out}$ and ω_2, depend on its effectiveness, ϵ_{fgd}. The effectiveness of the flue gas dryer is the ratio of the actual rate of heat rejection from the flue gas to the environment and the maximum rate of heat rejection, by which the flue gas is cooled to the ambient air temperature, 25 °C.

$$\dot{Q}_{fgd,max} = \dot{m}_{fg}[c_{p,fg}(T_{fg,out} - 25°C) + \omega_1 h_{v,T_{fg,out}} - (\omega_{be} - \omega_{min,fgd})h_{w,25°C} - \omega_{min,fgd} h_{v,25°C}] \tag{3.176}$$

$$\epsilon_{fgd} = \frac{\dot{Q}_{fgd}}{\dot{Q}_{fgd,max}} \tag{3.177}$$

The amount of water recovered in the flue gas dryer is based on the change in relative humidity.

$$\dot{m}_{cond,fgd} = \dot{m}_{fg}(\omega_1 - \omega_2) \tag{3.178}$$

The total rate of water recovery from the flue gas is the sum of the water recovery rates in the flue gas cooler and flue gas dryer, the former of which is zero except for scenario C.

$$\dot{m}_{cond} = \dot{m}_{cond,fgc} + \dot{m}_{cond,fgd} \tag{3.179}$$

The mass condensation rate can then be converted into a volumetric flow rate in $\frac{gal}{hr}$ by dividing by the density of water and multiplying by the necessary conversion factors.

$$\dot{V}_{cond} = 264.17\ [\frac{gal}{m^3}] \times 3600\ [\frac{s}{hr}] \times \frac{\dot{m}_{cond}}{\rho}\ [\frac{m^3}{s}] \tag{3.180}$$

3.3.3 Power Output and Efficiency

The mechanical power output, $\dot{W}_{shaft}$, from the plant is the sum of the power generated by the high, intermediate, and low pressure turbines. The electrical power output, $\dot{W}_{out}$, is then the product of the mechanical power output and the efficiency of the generator, assumed to be 99%. The power input, $\dot{W}_{in}$, is the sum of the power required for the boiler feedwater pump, booster pump, drain pump, condensate pump, recirculating water pump, pulverizer, and forced and induced draft fans. The net power output, $\dot{W}_{net}$, is the difference between the power output and power input.

$$\dot{W}_{shaft} = \dot{W}_{hpt} + \sum_{i=1}^{2} \dot{W}_{ipt,i} + \sum_{j=1}^{6} \dot{W}_{lpt,j} \tag{3.181}$$

$$\dot{W}_{out} = \eta_{gen} \dot{W}_{shaft} \tag{3.182}$$

$$\dot{W}_{in} = \dot{W}_{bfp} + \dot{W}_{bp} + \dot{W}_{dp} + \dot{W}_{cp} + \dot{W}_{rwp} + \dot{W}_{p} + \dot{W}_{fd} + \dot{W}_{id} \tag{3.183}$$

$$\dot{W}_{net} = \dot{W}_{out} - \dot{W}_{in} \tag{3.184}$$

The overall efficiency of the coal plant is then the ratio of the net electrical power output to the heat input.

$$\eta = \frac{\dot{W}_{net}}{\dot{Q}_{in}} \tag{3.185}$$

3.3.4 Water Consumption and Recovery

The cooling tower must reject the heat from the condenser to the environment through evaporation as shown in Figure 3.18. The mass evaporation rate, $\dot{m}_{evap}$, can be determined from the load on the cooling tower and water's enthalpy of vaporization at standard conditions, $h_{evap} = 2257\frac{kJ}{kg}$.

$$\dot{m}_{evap} = \frac{\dot{Q}_{out,cond}}{h_{evap}} \tag{3.186}$$

The mass evaporation rate can then be converted into a volumetric flow rate as shown previously with the mass condensation rate in equation 3.180.

$$\dot{V}_{evap} = 264.17\ [\frac{gal}{m^3}] \times 3600\ [\frac{s}{hr}] \times \frac{\dot{m}_{evap}}{\rho}\ [\frac{m^3}{s}] \tag{3.187}$$

The water consumption factor, WCF, is the volume of water lost from the cooling tower per MWh of electricity produced by the plant.

$$WCF = \frac{\dot{V}_{evap}}{\dot{W}_{net}}\ [\frac{gal}{MWh}] \tag{3.188}$$

Similarly, the water recovery factor, WRF, is the volume of water recovered from the flue gas dryer per MWh of electricity produced by the plant.

$$WRF = \frac{\dot{V}_{cond}}{\dot{W}_{net}}\ [\frac{gal}{MWh}] \tag{3.189}$$

Where $\dot{V}_{cond}$ can found by subsitituting $\dot{m}_{cond}$ from equation 3.179 into equation 3.187 in place of $\dot{m}_{evap}$. The production-consumption ratio, PCR, i.e., the fraction of the water consumed in cooling the plant that can be recovered from the flue gas, can be determined by dividing the water recovery factor by the water consumption factor.

$$PCR = \frac{WRF}{WCF} \tag{3.190}$$

The net consumption factor, NCF, is the difference between the water consumption and recovery factors. The net consumption factor is the amount of water that has to be replaced as the result of evaporation.

$$NCF = WCF - WRF \tag{3.191}$$

3.4 Summary of Methodology

The methodology chapter provides a step-by-step overview of the inputs and equations I used in my thermodynamic model of a supercritical coal plant. It includes a detailed explanation of the various compositions of coal and how coal composition affects flue gas composition, particularly moisture. The majority of the methodology focuses on the equations related to each of the individual components included in the coal plant model. The components of particular interest for this analysis are the flue gas coolers and flue gas dryer. The flue gas coolers are used to preheat either air or feedwater according to three separate arrangements, called scenario A, B, or C. Upon exiting the flue gas cooler, the flue gas enters the flue gas dryer where water from the flue gas can be recovered. After defining the operating parameters and equations for each of the components, factors such as the rate of power production, overall efficiency, water consumption factor and water recover factor are determined for each of the waste heat recovery scenarios. These values are presented and discussed in the results.

Chapter 4

Results

4.1 Overview

The results begin with a summary of the mass flows — air, flue gas, water, and coal — through the power plant. That summary is followed by a presentation of the efficiency improvements gained from the scenarios for low temperature waste heat recovery. The remainder of the results focus on the feasibility and benefits to be gained by a coal plant using a flue gas dryer to reduce its water consumption factor.

4.2 Mass Flows

The ratios of air, dry flue gas, and water vapor entering and exiting the boiler per kilogram of coal — called AFR, EFR_{dry} and ϕ_{H_2O} in equations 3.12, 3.20 and 3.16 respectively — are shown in Table 4.1. Note that the higher carbon coals require a higher AFR for complete combustion.

Table 4.1: Ratios of air, flue gas, and water for coals used in this model ($\frac{kg}{kg_{coal}}$)

	AFR	EFR_{dry}	ϕ_{H_2O}
Lignite	5.43	5.63	0.65
Subbituminous	7.73	8.03	0.60
Bituminous	9.85	10.24	0.43

Based on the ratios in 4.1, the mass composition of the flue gas was determined to be essentially the same for all of the coal types except for the fraction of SO_2, that makes up less than 0.2% of the flue gas produced by lignite and subbituminous coals but makes up 0.8% of flue gas produced by highly sulfuric bituminous coals. This difference had a negligible effect on the overall mass composition and resulting flue gas specific heat. The average composition of dry flue gas for all of the coal types is shown in Table 4.2.

Table 4.2: Mass composition of dry flue gas

N_2	73.49
CO_2	22.12
O_2	3.87
SO_2	0.39
Ar	0.12

Based on this mass composition, the specific heat of the flue gas, $c_{p,fg}$, was determined to be 0.99 $\frac{kJ}{kg}$. With $c_{p,fg}$ and the values from Table 4.1, the mass flow rates of coal, air, flue gas, and water vapor through the boiler were determined and are shown in Table 4.3.

Table 4.3: Mass flow rates of coal, air, flue gas, and water through the boiler ($\frac{kg}{s}$)

	Coal	Air	Flue Gas	Water
Lignite	109	592	614	71
Subbituminous	72	558	580	43
Bituminous	51	501	521	22

4.3 Efficiency and Water Consumption Factor

For each scenario, subbituminous coal plants are approximately 2% more efficient than lignite plants, and bituminous coal plants are approximately 2% more efficient than subbituminous coal plants.The overall efficiency of the plant increases

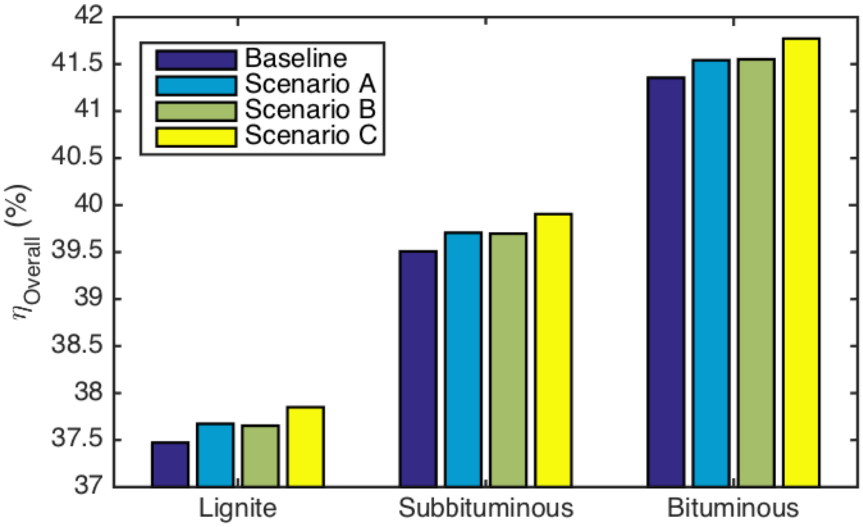

Figure 4.1: Overall efficiency based on coal type for each scenario

by about 0.2% from the baseline for scenarios A and B and about 0.4% from the baseline for scenario C. These results are somewhat less optimistic than those reported by Sarunac [12]. In that study, replacing the steam-air heater with a flue gas cooler achieved a 0.5% increase in efficiency, more than twice the efficiency improvement calculated by this model [12]. However, replacing the two lowest temperature heat exchangers achieved an efficiency improvement of 0.5 – 0.8% according to the Sarunac model, very close to the 0.4% calculated by this model. However, these increases in efficiency comes at a cost of additional water use per unit of electricity produced as shown in Figure 4.2.

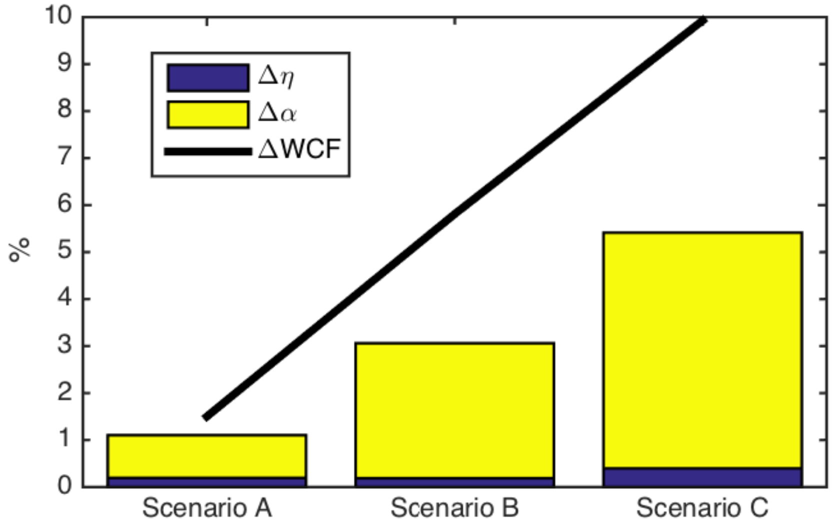

Figure 4.2: As plant efficiency, η, increases, the fraction of waste heat rejected to the condenser, α, increases as well, driving up the plant's WCF.

There is about a 0.2% increase in efficiency for either scenario A or B and a 0.4% increase in efficiency for scenario C. At the same time, the fraction of plant waste heat that is rejected to the condenser, labeled as α on Figure 4.2, increases by about 1%, 3% or 5% for scenarios A, B and C respectively. As a result, the water consumption factor increases by about 1.5%, 5.7%, and 9.7% for scenarios A, B, and C respectively.

The minimum effectivenesses for the flue gas cooler as defined for scenarios A, B, and C as defined in equations 3.162, 3.166, and 3.174 is shown in Figure 4.3. The required effectiveness increases substantially for higher quality coal and for each successive scenario for waste heat recovery.

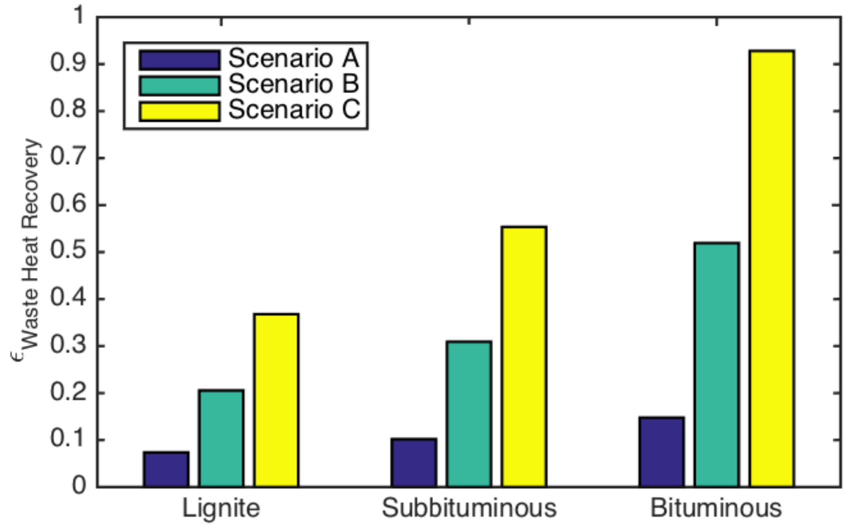

Figure 4.3: Minimum effectiveness for waste heat recovery unit for each scenario and coal type

4.4 Water Recovery Factor

The water recovery factor as defined by equations 3.177, 3.180, and 3.189 is directly related to the effectiveness of the flue gas dryer as shown in Figure 4.4. It is higher for lower quality coals because of the increased moisture content of the flue gas and the higher dew point, as shown in Table 4.4. It is higher for each successive waste heat scenario because the flue gas is cooled to a lower temperature, as shown in Table 4.5. Note that the temperature at the inlet of the flue gas dryer is 155 °C regardless of fuel type in the baseline case, and that the inlet temperature is below the dew point regardless of fuel type for scenario C.

Table 4.4: Flue gas dew point based on coal type [°C]

Lignite	56
Subbituminous	48
Bituminous	38

Table 4.5: Inlet temperature of flue gas dryer depending on fuel type and scenario for waste heat recovery [°C]

Scenario	A	B	C
Lignite	126	78	54
Subbituminous	126	74	45
Bituminous	126	64	31

Based on these results, the maximum water recovery factor is about 91, 50 or 19 $\frac{gal}{MWh}$ for coal plants burning lignite, subbitumous or bituminous coal respectively. For a given heat exchanger effectiveness for the flue gas dryer, more water is recovered for each successive scenario for low temperature waste heat recovery.

The production-consumption ratio as defined in equation 3.190 as shown in Figure 4.5 is higher for lower quality coals. That is because power plants burning lower quality coals reject a greater percentage of waste heat through the flue gas than power plants burning higher quality coal, which reduces the percentage of waste heat rejected to the condenser and, thus, the water consumption factor. Additionally, the water recovery factor is higher for lower quality coals, as shown in Figure 4.4. The result is that a more significant fraction of the water consumed in the cooling tower is recovered in the flue gas cooler for the lower quality coals and with each successive waste heat recovery scenario as shown in Figure 4.5.

Based on these results, the maximum production-consumption ratio is about 0.16, 0.10 or 0.38 for coal plants burning lignite, subbituminous or bituminous coal

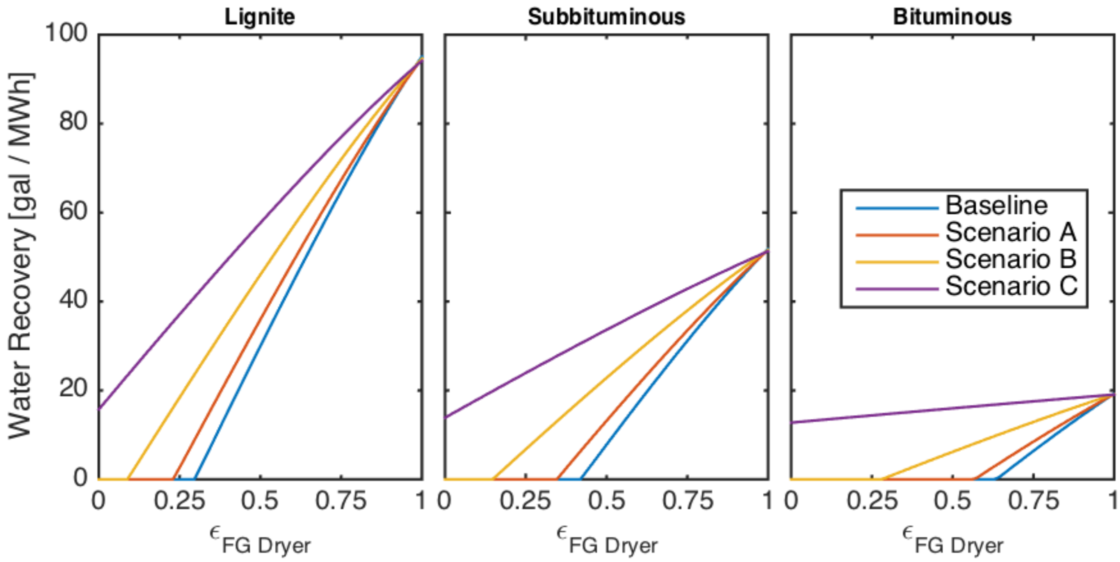

Figure 4.4: Volume of water recovered per MWh of electricity produced by the plant

respectively. For a flue gas dryer with heat exchanger effectiveness below about 0.85, the production-consumption ratio is higher for each successive scenario for waste heat recovery.

The effectiveness of the flue gas dryer must be sufficient to cool the flue gas below its dew point before there is any reduction in the net consumption factor as defined by equation 3.191. Beyond that point, an increase in the effectiveness of the flue gas dryer can reduce the water consumption factor below that of the baseline model without any waste heat or moisture recovery as shown in Figure 4.6.

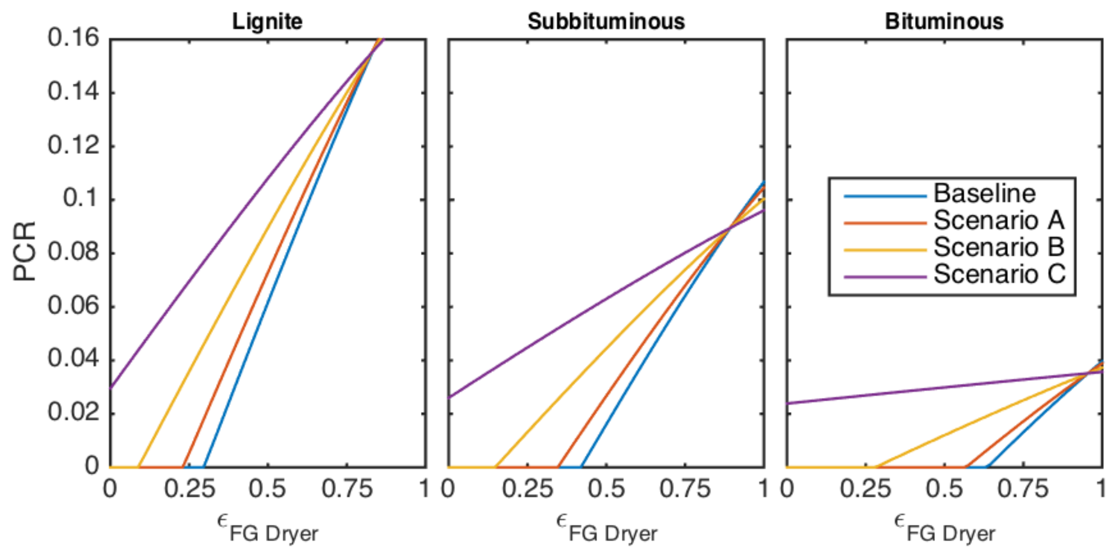

Figure 4.5: Ratio of water production and consumption for each scenario and coal type based on the effectiveness of the FG Dryer

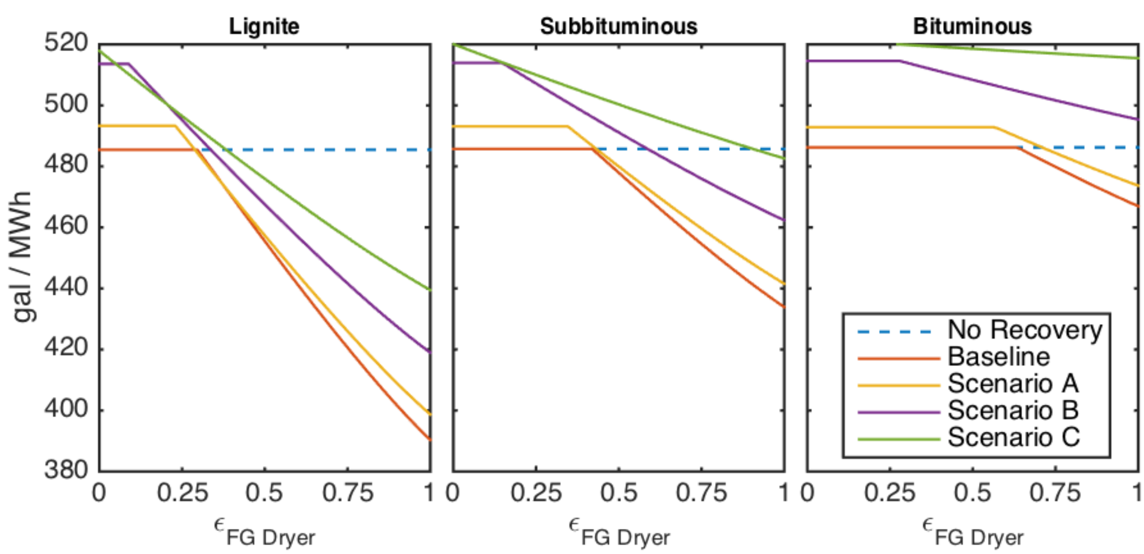

Figure 4.6: Net water consumption factor for each scenario and coal type based on the effectiveness of the FG dryer

The flat lines on Figure 4.6 indicate that the effectiveness of the flue gas dryer is not sufficient to reduce the water consumption factor for a given waste heat recovery scenario. According to these results, a flue gas cooler with an effectiveness of 0.5 would be sufficient to reduce the net water consumption factor for any of the waste heat recovery scenarios considered for a coal plant burning lignite. For a coal plant burning subbituminous coal, a flue gas cooler with an effectiveness of 0.5 would be sufficient to reduce the net water consumption factor for the baseline arrangement or for waste heat recovery scenario A. For a coal plant burning Bituminous coal, the flue gas cooler would have to have an effectiveness greater than 0.5 to achieve any reduction in the net water consumption factor.

Chapter 5

Conclusions

The analysis performed in this thesis contributes to ongoing efforts to improve the fuel efficiency and reduce the rate of water consumption of coal and other fossil fuel power plants. Results indicate that as much as a 0.4% increase in efficiency can be gained from low temperature waste heat recovery of boiler flue gas and that recovering water from boiler flue gas could reduce coal plant water consumption. Overall, the results indicate that such measures would be most feasible and beneficial for coal plants burning lower quality coals.

5.1 Summary of Results

All of the scenarios for low temperature waste heat recovery considered in this analysis achieved an increase in overall efficiency from 0.2 – 0.4%. The increase in efficiency for scenarios A and B was about 0.2% regardless of coal type. However, the resulting increase in water consumption rate was much higher for scenario B than for scenario A. Similarly, the required heat exchanger effectiveness for the flue gas cooler for scenario B is higher than for scenario A. Based on these factors, scenario A would be preferable to scenario B. The increase in efficiency for scenario C is about 0.4%, regardless of coal type. The water consumption factor and minimum heat exchanger effectiveness is higher for scenario C than for either scenario A or B. Thus, whether scenario C is preferable to scenario A depends on the relative value of efficiency and water consumption and the additional cost for a more effective heat exchanger. For all of the scenarios considered, the overall efficiency, water consumption factor, and minimum heat exchanger effectiveness for the flue gas cooler are higher for a coal plant burning high quality coal.

For a flue gas dryer with a given heat exchanger effectiveness, the water recovery factor will be higher for each successive waste heat recovery scenario. The same is true for the production-consumption ratio for a flue gas dryer with an effectiveness of 0.85 or lower. The maximum water recovery factor and production-consumption ratio is much higher for low quality coals. The required heat exchanger effectiveness for the flue gas dryer for the net water consumption factor to be reduced compared to the baseline without any low temperature waste heat recovery is higher for high quality coal.

In general, the results indicate that low temperature waste heat and water recover from boiler flue gas would be more feasible and beneficial for coal plants burning lignite as opposed to higher quality coal. Because these plants already have a lower efficiency, the relative increase in efficiency is somewhat higher. Similarly, the relative increase in water consumption factor is somewhat lower for a lignite plant. The high moisture content and dew point of the flue gas produced from lignite combustion makes it easier to recover water with a flue gas dryer. The higher water recovery factor along with the lower water consumption factor means that a greater percentage of the water evaporated in the cooling tower can be recovered in the flue gas dryer of a lignite plant than for a plant burning higher quality coal.

5.2 Future Work

This analysis is intended to simulate the operation and thermodynamic performance of a supercritical coal plant with the intent of identifying pathways to improving efficiency and reducing water consumption. However, for the sake of computational effort, there are many details that were simplified or left out entirely. The most significant simplification in this analysis is the lack of a flue gas scrubber in the power plant model. If a flue gas scrubber was included downstream of the flue gas cooler, the reduced temperature of the flue gas would reduce the rate at which water evaporates in the scrubber. The flue gas would then leave the scrubber at or near the adiabatic saturation temperature before it entered the flue gas dryer. By entering the flue gas dryer fully saturated, the rate at which water could be recovered from the flue gas would increase. Additionally, with over 90% of the SO_2 removed from the flue gas, the treatment costs for the recovered water would be substantially lower [29]. Thus, a flue gas scrubber would have a significant impact on water recovery from coal plant flue gas. Further analysis should incorporate a flue gas scrubber into the coal plant model.

The background section includes some comments on how the water recovered from the flue gas would be treated. The extent to which the water has to be treated depends both on its quality and how it is going to be used. The water quality depends on the extent to which pollutants are removed from the flue gas before it enters the flue gas dryer, either because they condensed out in the flue gas cooler or were intentionally removed in a flue gas scrubber. Stringent treatment would be needed for the recovered water to be used as cooling water or feedwater, but other uses such as scrubber makeup water or water for bottom ash sluicing would require less treatment. Future work should include more detail regarding these processes and their associated costs and waste streams.

In this analysis, effects of the condensation of acid in the flue gas cooler are mentioned but not quantified. The background includes a description of how acid condensation can foul the flue gas cooler and increase the back pressure on the exhaust fan, decreasing plant efficiency. An effort to quantify these reductions in efficiency is necessary for a comprehensive understanding of the effects of low temperature waste heat and water recovery from boiler flue gas. Additionally, the latent heat of the condensing acid and the subsequent changes in the thermal properties of the flue gas should also be taken into consideration.

Flue gas reheat is another relevant topic that was not included in this analysis. For pollution control purposes, flue gas above a certain moisture content has to be rejected to the atmosphere at a high enough temperature to ensure proper dispersal of pollutants. Future analysis should incorporate this process into the coal plant model and scenarios for waste heat recovery.

This analysis is generally applicable to new coal plants, none of which are expected to be built in the U.S. in the foreseeable future. It would be extremely difficult for an existing coal plant to have equipment such as feedwater heaters removed and replaced with flue gas coolers. Thus, further work should consider the potential for retrofits which add flue gas coolers and dryers to existing coal plants.

While this analysis covers the feasibility and potential benefits of low temperature waste heat and water recovery from a thermodynamic perspective, it does not address them from an economic perspective. Improving plant efficiency increases the amount of electricity that can be produced by burning a certain amount of coal, the economic impact of which depends on a variety of factors, including the price of electricity, the price of coal, and the capital cost associated with waste heat recovery equipment. Similarly, the economic impact of reducing water consumption depends on the availability and price of water and the capital cost associated with water recovery equipment. Further analysis should include estimates of the economic impact of low temperature waste heat and water recovery from boiler flue gas.

5.3 Recommendations

This analysis should be considered in the context of efforts to improve efficiency and reduce water consumption in the electric power sector. As addressed in this analysis, efficiency improvements sometimes come at the expense of increased water consumption. Thus, the relative values of efficiency and water consumption have to be taken into consideration.

As mentioned in the previous section, adding low temperature waste heat and water recovery from boiler flue gas to a coal plant increases capital expenditures. However, many states and cities dealing with ever increasing demand for water have established various grants and funds for the purpose of investing in water production or conservation infrastructure. An example of such a fund in Texas is SWIFT, State Water Implementation Fund for Texas. Water conservation or recovery equipment for power plants are not currently on the list of approved projects for SWIFT, but it may make sense for them to be included in the future based on the significant amount of water consumed in the electric power sector.

More generally, the results of this study indicate that improvements in one area of a power plant's operation can affect other aspects of its operation for better or worse. Recovering waste heat from the flue gas increases the rate of water consumption in the cooling tower. At the same time, it is easier to recover moisture from the flue gas after it has been cooled in an flue gas cooler. Thus, incorporating both solutions allows for an improvement in both efficiency and a reduction in water consumption. Other challenges related to the Energy-Water Nexus might involve similar interactions between efficiency and water use. The complexity of these challenges limits the extent to which any individual technology or policy can be a "silver bullet." Rather, a variety of solutions working in concert with each other is needed to achieve more sustainable energy and water systems.

Appendix

Table 1: Specific heat of constituents of dry flue gas [$\frac{kJ}{kgK}$]

N_2	1.04
CO_2	0.84
O_2	0.92
SO_2	0.64
Ar	0.52

Table 2: Molar weight of constituents of dry flue gas [$\frac{kg}{kmol}$]

N_2	28.01
CO_2	44.01
O_2	32.00
SO_2	64.06
Ar	39.95

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Vita

Andrew Samuel Reimers was born in Orange, TX, December 18, 1989, the son of Joseph and Laurie Reimers. He received a Bachelor of Science degree in mechanical engineering from Texas A&M University in 2013. He is currently a member of the Webber Energy Group at the University of Texas where his research focuses on energy and water use at thermoelectric power plants and applications for waste heat recovery.

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