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Comparing the Effectiveness of Heat Rate Improvements in a Coal-Fired Power Plant Utilizing a Chilled-Ammonia CO2 Capture System

Elaine Aiken
Lehigh University

COMPARING THE EFFECTIVENESS OF HEAT RATE IMPROVEMENTS IN A COAL-FIRED POWER PLANT UTILIZING A CHILLED-AMMONIA CO_2 CAPTURE SYSTEM

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Elaine Aiken

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Elaine Aiken

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Dr. Edward K. Levy

Thesis Advisor

Dr. D. Gary Harlow

Chairperson of the Department

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Abstract

Coal-fired power plants emit carbon dioxide (CO_2), which is likely to be regulated in the near future. In order to lower CO_2 emissions, research is being conducted on technologies to separate CO_2 from other components in flue gas. CO_2 separation can be achieved with an ammonia post-combustion capture system, which chemically scrubs CO_2 from flue gas by absorbing it into the ammonia. The absorbed CO_2 is regenerated from the ammonia so that a relatively pure CO_2 stream is achieved, which is then compressed to high pressures.

Utilizing a capture system requires additional equipment, power, and heat requirements, which can be met by extracting steam from the steam turbine cycle, which would reduce the power output of the plant. The CO_2 capture and compressor system generates heat which can be utilized to offset some of the heat requirements, thereby improving plant performance and reducing the impact of the capture system.

1 Introduction

As emission standards in the United States are continually becoming stricter, new technologies are being researched and developed in an attempt to keep up with the increasing number of environmental regulations. In a coal-fired power plant, the coal is burned which emits a flue gas that is released into the environment. This flue gas is a combination of carbon dioxide (CO_2), water (H_2O), nitrogen (N_2), and oxygen (O_2). In an attempt to reduce the CO_2 emissions from coal-fired power plants, research is being conducted to separate CO_2 from the other components of the flue gas. The CO_2, which makes up about 11% of the flue gas, was targeted because environmental regulations are starting to limit levels of CO_2 emissions from power plants.

Once the CO_2 is separated, it can be utilized for enhanced oil recovery or it can be sequestered. Enhanced oil recovery is a process that uses pressurized CO_2 to retrieve more oil from oil reservoirs. The CO_2 helps push oil from pockets that ordinary oil recovery methods may miss. The CO_2 then remains in the reservoir rather than being released into the air. Sequestration is a process of injecting the CO_2 deep into the ground for storage in geological formations such as saline aquifers. These storage options reduce the amount of CO_2 that is released into the atmosphere.

There are two primary methods being developed for CO_2 capture from coal burning power plants. One method is oxy-combustion which combusts coal in pure oxygen as opposed to air, which contains nitrogen. This combustion in oxygen leads to a flue gas that is almost purely CO_2. Since the flue gas from oxy-combustion is almost pure CO_2, no additional work needs to be done in order to separate the CO_2 from any other components. This process requires the addition of an air separation unit which provides the pure oxygen, and some alterations must be made to the existing boiler to allow for the coal to combust in pure oxygen in the boiler. Oxy-combustion does not require changes to the rest of the existing power plant.

The other method being researched is post-combustion carbon capture systems. This process uses an amine and water mixture that reacts with the flue gas and absorbs most of the CO_2 in an absorber column. The CO_2 is then separated back out from the amine and water solution in a stripper vessel at an elevated temperature and pressure. This is a back-end CO_2 capture system that is easily added to an existing coal-fired power plant. These systems do not require any changes to the existing power plant, but it does require additional equipment to be added to the plant.

Both capture methods have their advantages and disadvantages. Each process requires additional equipment, additional energy requirements, and reduces the overall power output of a coal-fired power plant. Even so, these types of capture systems will become necessary as CO_2 emission limits become stricter in the future. The method of post-combustion carbon capture is modeled in this thesis, and the amine used for the capture system is chilled ammonia (NH_3).

In order to sequester the captured CO_2 or to use it for enhanced oil recovery, it must be highly pressurized. To reach this high pressure, the compression process requires a significant power input which decreases the plant's overall performance. However, this compression cycle creates a large amount of waste heat which can be used to offset heat requirements in other parts of the plant. Integrating this heat from compression can increase the net power of the plant and improve the unit heat rate and efficiency.

When coal is burned in a power plant, the heat from the combustion process is used to heat water and create steam. This steam is sent through a series of turbines which produce the plant's power. Some equipment in a power plant requires heat input which is met by extracting steam from this steam turbine cycle, and this reduction in steam through the turbine cycle reduces the net power output of the plant. Using the heat generated from the CO_2 compression cycle for some of the plant's heat requirements can decrease the amount of steam being extracted from the steam turbine cycle and therefore increase the net power output of the plant. There are three main methods of integrating the compression heat that are researched in this thesis.

The first method of thermal integration researched is to offset the heat requirements of some of the feedwater heaters which pre-heat the water used for the steam turbine cycle. Without thermal integration from the compression cycle, the feedwater is heated through a series of feedwater heaters that require steam extractions from the steam turbine cycle. With thermal integration, some of the extractions can be either replaced or reduced in order to increase the steam flow through the turbine cycle and therefore increase the plant's net power output.

Another method of thermal integration is to offset the heat requirements for the stripper reboiler. In the post-combustion carbon capture system, the stripper vessel, which separates the NH_3 and water from the CO_2, contains a reboiler that requires a large amount of heat to operate. Without thermal integration from the compression cycle, the reboiler heat duty requirement is met with steam extracted from the steam turbine cycle. With thermal integration from the compression cycle to the reboiler, the steam extraction needed from the steam turbine cycle can be reduced which leads to an increase in the net power output of the plant.

The last method of thermal integration researched is coal-drying. When the moisture of coal is reduced, less coal must be burned in order to produce the same amount of power. The heat from compression can be used to achieve this reduction in moisture and therefore reduced the amount of coal that needs to be burned.

Three different types of compressors were analyzed for the CO_2 compression. These compressors were compared for their required power and the usable heat that they produce. The three methods of thermal integration were then analyzed for each type of compressor. The effect on plant performance was compared for each type of compressor and each method of thermal integration in order to find the CO_2 compression process that would have the least impact on the plant's overall performance.

ASPEN Plus software was used to model the chilled ammonia post-combustion carbon capture system along with the three different compressors and the thermal integration. The boiler and steam turbine cycle have been modeled in ASPEN Plus by previous graduate students, and the capture system and compressors were added to these previously created models. These models were used to simulate the power plant with the addition of the chilled ammonia carbon capture system and the different compressors. Within the model, heat integration can be done in order to analyze the effect on the plant's overall performance.

2 Coal Plant

A coal-fired power plant operates by burning coal in a boiler which generates heat that is used to create high-pressure steam. The steam is then sent through a steam turbine cycle which produces the plant's power. The plant analyzed in this thesis used a steam turbine cycle based on a steam turbine kit that was already developed, and the boiler was designed in order to generate the steam required for the steam turbine kit.

The focus of the research in this thesis was on the development of the chilled ammonia carbon capture system, its effect on plant performance, and the effect of heat integration from the capture system. Therefore, a coal-fired power plant that was modeled in ASPEN Plus by fellow Lehigh University graduate student Erony Martin was utilized, and the capture system was attached to this plant model (Martin, 2011). The development and results generated from this ASPEN Plus model of the coal plant is described in detail in Martin's thesis.

The following sections describe the basic operation of this plant so that the addition of a CO_2 capture system to the coal plant can be understood. For more detail on the components of a coal-fired power plant and how it operates, please refer to the theses of Lehigh graduate students Martin, Walsh, and Szatkowski (2011, 2009, 2009).

The plant layout is illustrated below in Figure 2.1, and the boiler and steam turbine cycle are discussed in Sections 2.1 and 2.2, respectively. The CO_2 capture system is discussed in Section 3, and the CO_2 compression is discussed in Section 4.

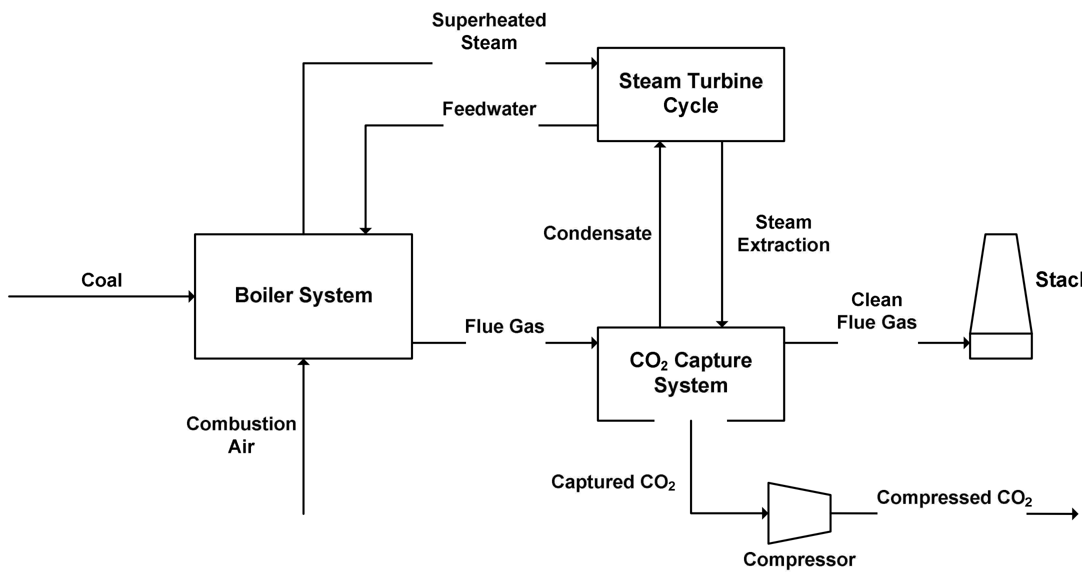

Figure 2.1 - Plant Layout.

2.1 Boiler

The boiler utilized in this thesis was designed to burn a sub-bituminous, Powder River Basin (PRB) coal which is pulverized before it enters the boiler. This coal contains 28.09% moisture, and the effect of drying the coal before combustion is discussed in Section 6.2.4. The required amount of coal fed to the boiler was based on generating the steam required for the steam turbine kit, and this resulted in the use of 648,177 lb/hr of coal.

The composition of the coal used is detailed below in Table 2.1. The proximate analysis is a measurement of the relative amounts of carbon, ash, volatiles and moisture, and the ultimate analysis is the measurement of the amount of each component by weight. The sulfur analysis is the measurement of the different types of sulfur present in the coal. The higher heating value (HHV) is the amount of energy released from burning one pound of coal.

Coal combustion generates heat which is used to produce high-pressure steam. Burning coal creates flue gas which gets treated to reduce the pollutants within it, and it is then released into the atmosphere. With the addition of a post-combustion carbon capture system, this flue gas is instead fed to the capture system where the CO_2 is separated, and the remaining flue gas is then released.

Table 2.1 - Composition of PRB Coal.

Sub-bitumious Coal (PRB) As-Received Analyses	
Proximate Analysis (wt%)	
Moisture	28.09
Ash	6.31
Volatile Matter	32.17
Fixed Carbon	32.98
Sulfur	0.45
HHV (Btu/lb)	8,426
Ultimate Analysis (wt%)	
Carbon	49.21
Hydrogen	3.51
Sulfur	0.45
Oxygen	11.67
Nitrogen	0.73
Chlorine	0.02
Moisture	28.09
Ash	6.31
Sulfur Analysis (wt%)	
Pyritic	0.17
Sulfate	0.03
Organic	0.43

Figure 2.2 below illustrates the basic boiler design where the feedwater is converted to steam by coal combustion and sent to the steam turbine cycle.

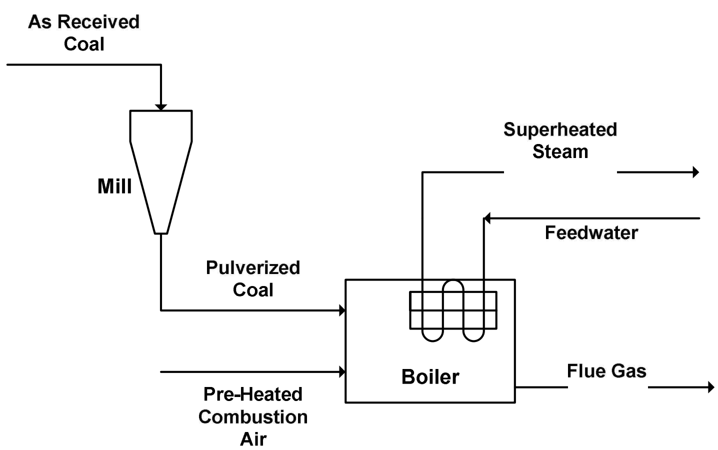

Figure 2.2 - Boiler Diagram.

There are other pieces of equipment required for the boiler system that are included in the analyses in this thesis. There are fans to maintain the proper pressure of air circulating in the system as well as a heat exchanger to pre-heat the combustion air with the hot flue gas. There is also additional equipment used treat the flue gas before it is sent to the capture system to remove components such as ash and sulfur.

2.2 Steam Turbine Cycle

In the coal-fired power plant analyzed in this thesis, a supercritical steam turbine kit was used. This pre-designed turbine kit gave the parameters that were used in the steam turbine cycle (STC) modeled in ASPEN Plus, and this allowed for verification that the ASPEN Plus model was working accurately. Figure 2.3 illustrates the supercritical steam turbine cycle which is described in this section. A detailed diagram of the steam turbine cycle, including temperatures, pressures, and flowrates of each stream can be found in Appendix A.

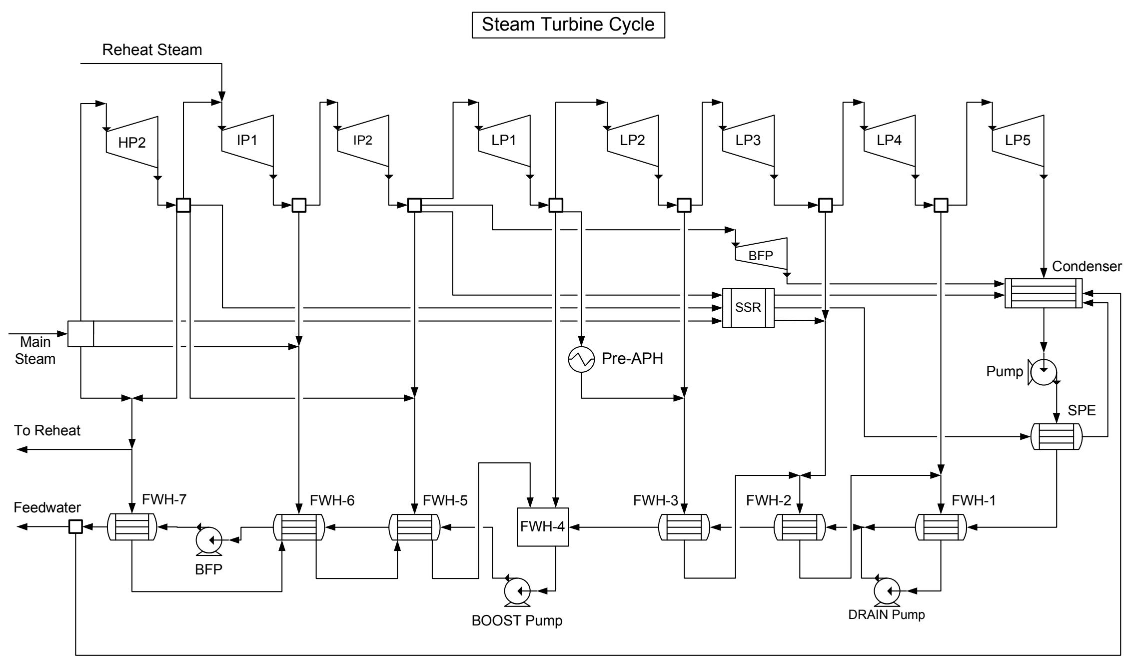

Figure 2.3 - Steam Turbine Cycle.

Feedwater is pre-heated by a series of feedwater heaters (FWHs), and it is then sent to the boiler where it is heated by the combustion process and becomes high-pressure steam. This high-pressure steam travels through a series of turbines, which generate the plant's power. After the steam passes through all of the turbines, it becomes the feedwater and travels through the feedwater heaters again.

The steam from the boiler first travels through a high-pressure (HP) turbine which is followed by a two-stage intermediate pressure (IP) turbine. This is then followed by a five-stage low pressure (LP) turbine. The steam is expanded in each stage and the temperature and pressure of the steam are reduced. When the steam is expanded in the turbine, it exerts a force on the turbine blades causing it to produce work in the form of shaft work. The temperature and pressure of the steam entering and exiting the turbine, as well as the flowrate of the steam, determines the power produced by the turbine. This series of turbines can be seen in Figure 2.3 where the turbines are labeled HP1, IP1 and IP2, and LP1 through LP5. The booster feed pump (BFP) shown in Figure 2.3 uses a steam extraction to power the booster feed pump, therefore the power produced by this turbine is not included in the total. The turbine stage work is tabulated in Table 2.2 below.

Table 2.2- Turbine Stage Data.

Turbine	Inlet Pressure (psia)	Outlet Pressure (psia)	Steam Flow (lb/hr)	Isentropic Efficiency (%)	Work Out (kW)
HP 1	3,689.7	740	4,122,221	85.08%	183,314
IP 1	666	295	3,730,357	83.54%	106,167
IP 2	295	165.1	3,545,824	86.48%	65,889
LP 1	165.1	87.4	3,145,724	87.50%	57,586
LP 2	87.4	24.9	2,797,689	89.69%	85,016
LP 3	24.9	11.96	2,688,685	89.87%	39,200
LP 4	11.96	4.68	2,574,150	89.73%	42,927
LP 5	4.68	0.61	2,395,203	67.45%	55,798
Total					635,897

The FWHs are heat exchangers that heat the feedwater with high temperature steam extractions from the turbine cycle. Extracting this steam from the turbine cycle reduces the amount of steam that travels through the turbines which reduces the power produced by the turbines. There are a series of seven FWHs and each one uses steam extractions from different stages of the turbines. The higher temperature FWHs require steam extractions from earlier in the turbine cycle, while the lower temperature FWHs require steam extractions from the LP turbines. These FWHs are all closed FWHs with the exception of the fourth FWH (FWH-4) which is an open FWH. In a closed FWH, the feedwater and the steam extraction do not mix and remain separate streams exiting the FWH. In an open FWH, the feedwater and the steam extraction are mixed together and both streams exit the FWH as the feedwater stream. It can be seen in Figure 2.3 that FWH-4 has three streams entering but only one exit stream once the streams are all mixed.

After a steam extraction is used in a FWH, it is then sent to the previous, lower temperature FWH and mixed with the steam extraction for that FWH. This allows the heat that still remains in that steam extraction to be utilized in the previous FWH. This cascading of the steam extractions is done for each FWH. The exception to this is FWH-4, which collects the recycled steam extractions from FWH-5 through FWH-7 and mixes them together and sends them to FWH-5. Figure 2.4 below illustrates the design of a closed FWH including the recycled steam extraction from the following FWH.

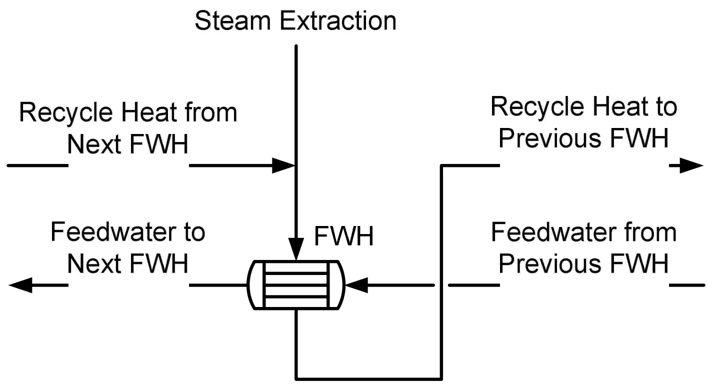

Figure 2.4 - Feedwater Heater Diagram.

The inlet and outlet conditions of the feedwater in each FWH, as well as the steam extractions required are shown in Table 2.3 below.

Table 2.3 - Feedwater Heat Data.

	Feedwater		Steam Extraction
Feedwater Heater	**Inlet Temp. (°F)**	**Outlet Temp. (°F)**	**Flowrate (lb/hr)**
FWH-1	87.1	151.9	178,947
FWH-2	152.7	193.8	114,535
FWH-3	193.8	231.4	109,004
FWH-4	231.4	313.8	257,172
FWH-5	314.5	363.6	163,004
FWH-6	363.6	414.2	184,533
FWH-7	426.2	506.9	432,374

2.3 Plant Performance

There are multiple pieces of auxiliary equipment within the power plant that take away from the power produced by the plant. These extra power requirements make up what is termed the station service power. This station service power includes the power requirements of equipment such as the pulverizer, fans, and pumps. When all of this is considered, the plant produces a net power of 587.7 MW and has a net unit heat rate (HR) of 9,362 Btu/kWh. Table 2.4 below contains the plant statistics for the ASPEN Plus model used in the research in this thesis.

Table 2.4 - Plant Statistics with no CO_2 Capture Case.

Boiler Performance	
Q Steam (Btu/hr)	4,776,219,255
Coal Flow Rate (lb/hr)	648,177
HHV Coal (Btu/lb)	8,426
Q Coal (Btu/hr)	5,501,978,582
Turbine Power	
HP Power (kW)	183,314
IP 1 Power (kW)	106,165
IP 2 Power (kW)	65,888
LP 1 Power (kW)	57,560
LP 2 Power (kW)	84,978
LP 3 Power (kW)	39,182
LP 4 Power (kW)	42,908
LP 5 Power (kW)	55,774
Gross Power (kW)	635,768
Station Service Power	
Mill Power (kW)	3,430
Fan Power (kW)	18,340
Pump Power (kW)	1,658
Auxiliary Power (kW)	15,000
Total Pss (kW)	38,428
Plant Performance	
Net Power (kW)	587,804
Turbine Cycle HR (Btu/kWh)	7,627
Net Unit HR (Btu/kWh)	9,291
Unit Efficiency (%)	36.72%

The “Q Steam” in this table refers to the Btu’s of energy available in the steam, and “Q Coal” refers to the Btu’s of energy available in the coal. The higher heating value (HHV) is a measurement of how much heat is released from combustion and is dependent on the type and moisture content of coal.

This base model of the coal plant was used and the CO_2 capture system that was designed for this thesis was added to this coal plant. The effects that the capture system has on these plant statistics and the effects of heat integration are detailed in Section 5.

3 CO_2 Capture

The post-combustion carbon capture systems are designed to separate CO_2 from the flue gas using an amine that absorbs CO_2. An amine is an organic compound, and amines used for CO_2 capture absorb CO_2 and have superior absorption performance. In order to separate the CO_2, the amine and flue gas interact in a vessel called an absorber. The amine and CO_2 mixture exit the bottom of the absorber as a liquid while the rest of the flue gas exits the top of the absorber as a gas. The amine and CO_2 mixture is termed the rich amine due to the fact that the amine is now rich in CO_2. This rich amine solution is then heated and pressurized and sent to the stripper vessel where the amine and CO_2 separate. The amine exits the bottom of the stripper as a liquid, and at this point it is referred to as the lean amine. This lean amine is then returned to the absorber and continues to cycle through the capture system. The CO_2 exits the top of the stripper as a gas and is sent to a compression cycle where it is pressurized for further use or for storage.

This carbon capture process can operate with different amines, and the different amines being researched each have their own advantages and disadvantages. For this CO_2 scrubber unit, the amine used was ammonia, which is mixed with water and some CO_2 which aids in the absorption of CO_2 from the flue gas. This ammonia stream, the lean amine solution, is chilled before entering the absorber in order to increase the CO_2 absorption into the ammonia solution.

One of the advantages of using ammonia is that the separation of the CO_2 from the aqueous ammonia solution can be achieved with less energy input than with other amines. Another advantage is that ammonia is less corrosive to the equipment in the carbon capture process than other amines. One of the disadvantages of using chilled ammonia is the addition of refrigeration cycles that aren't needed with other amines. Another disadvantage is ammonia slipping out of the top of the absorber with the vent gas after the CO_2 has been absorbed, and this slip leads to the requirement of an ammonia abatement process to minimize ammonia emissions into the atmosphere.

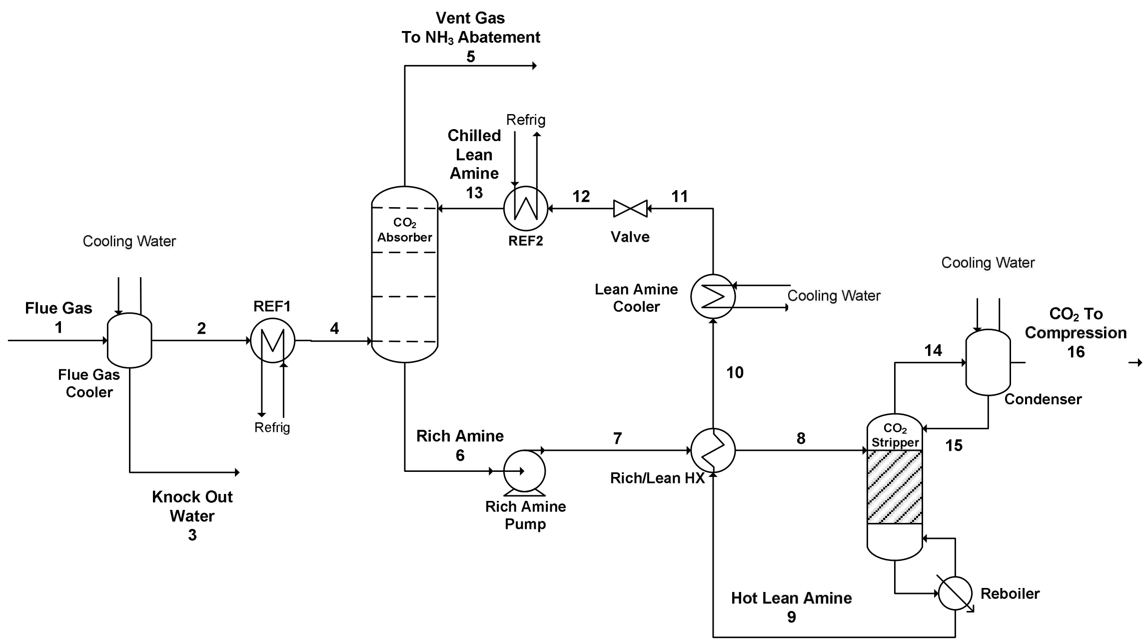

Figure 3.1 – Post-Combustion Chilled Ammonia Carbon Capture System.

Figure 3.1 illustrates the post-combustion chilled ammonia carbon capture process that is outlined in the following sections.

3.1 CO_2 Absorption

The first stage of the carbon capture process involves the chilled lean ammonia solution (stream 13 in Figure 3.1) reacting with the chilled flue gas (stream 4) in order to separate the CO_2 from the flue gas. The flue gas and the lean ammonia solution are both chilled before the absorption process in order to improve the CO_2 absorption. The streams can be chilled with cooling water first, and then a refrigeration cycle must be used to further chill the streams.

The absorption of the CO_2 into the lean ammonia is achieved in a vessel in which the two streams come into contact and the chemical reactions between the two streams allow for the CO_2 to be absorbed into the lean ammonia stream resulting in the rich ammonia stream.

The lean ammonia solution enters the absorber at the top stage and the flue gas enters the absorber at the bottom stage. The streams come into contact and the lean ammonia stream absorbs the CO_2 from the flue gas. The vent gas exits the top of the absorber as a vapor with minimal amounts of CO_2 (stream 5). Some of the ammonia escapes with the vent gas out of the top of the absorber and this is discussed further in Section 3.3. The rich ammonia exits the bottom of the absorber with the majority of the original CO_2 that was present in the flue gas (stream 6).

3.2 CO_2 Separation

The CO_2 must then be separated back out from the ammonia solution. In order to enhance this stripping process, the rich ammonia stream is pressurized (stream 7) and heated (stream 8) before the separation process. It then enters a vessel called a stripper where the CO_2 separates from the ammonia due to the elevated temperature and pressure. The CO_2 stream exits the top of the stripper as a vapor (stream 14) and the lean ammonia stream exits the bottom of the stripper as a hot liquid (stream 9). The stripper contains a condenser at the top of the vessel and a reboiler at the base of the vessel.

The condenser cools the exiting CO_2 stream (stream 14) in order to condense excess water out of the stream. Condensing water from the stream provides a near pure CO_2 vapor stream exiting the condenser (stream 16). This condensed water (stream 15) is then returned to the stripper vessel and ultimately ends up in the lean ammonia stream. The exiting CO_2 stream is sent to the compression cycle discussed in Section 4. The condenser uses a cooling water heat exchanger to cool the CO_2 stream, and the heat from the CO_2 stream is transferred to the cooling water. This heat can be used elsewhere in the plant to offset heat requirements, and this is discussed further in Section 5.

The reboiler is a heat exchanger that heats the liquid exiting the bottom of the stripper (the stripper bottoms) in order to separate more CO_2 and return it to the stripper vessel. When heated, the stripper bottoms create vapor which is primarily the CO_2 and a liquid that is primarily the ammonia. The vapor is sent back to the stripper, and the liquid (hot lean amine, stream 9) exits the reboiler. This is illustrated in Figure 3.2 below. This process increases the separation of the CO_2 from the ammonia solution and is necessary to achieve the level of CO_2 separation that is desired. The reboiler requires a large amount of heat input to operate, and this heat requirement is met using steam extracted from the steam turbine cycle. This additional steam extraction has a large impact on the plant. This impact is discussed further in Section 6.

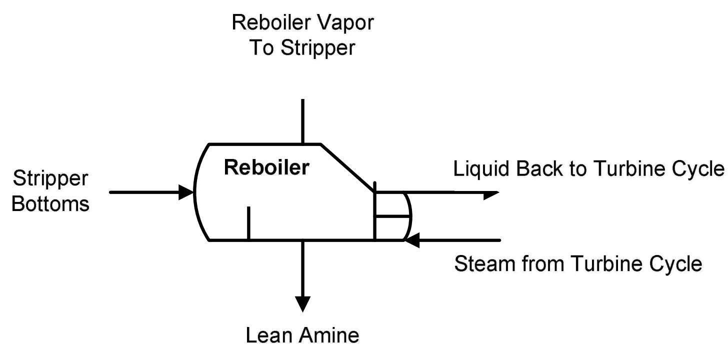

Figure 3.2 - Stripper Reboiler.

The lean ammonia solution exits the stripper reboiler as a liquid. The lean ammonia stream exiting the stripper is at an elevated temperature and is therefore termed the hot lean ammonia stream. This excess heat is used in a heat exchanger to heat the rich ammonia stream before it enters the stripper. The cool lean stream (stream 10) must then be further cooled before it is returned to the absorber at the beginning stage of the carbon capture system.

3.3 Ammonia Abatement

One of the disadvantages of using ammonia as the amine in the carbon capture process is ammonia slip. In the process of absorbing the CO_2, a portion of the ammonia exits the top of the absorber with the vent gas (stream 5). In order to prevent too much ammonia from being released into the atmosphere, an ammonia abatement process must be used to rid the vent gas of ammonia and recycle the ammonia back into the carbon capture process.

Ammonia abatement can be accomplished using a water wash process. This process is similar to the carbon capture system in that it involves absorbing the ammonia from the vent gas into water in an absorber, and then releasing the ammonia from the water in a stripper. The ammonia from the abatement process is sent back in to the carbon capture process and the water can be recirculated within the abatement process.

Similar to the carbon capture process, the absorption in the abatement process involves two streams entering an absorber. The vent gas containing the excess ammonia enters the bottom of the absorber and the water stream enters the top. The water absorbs the ammonia from the vent gas and exits the bottom of the absorber, and this stream is termed the rich abatement stream. The vent gas, which now contains minimal amounts of ammonia, exits the top of the absorber and is released into the atmosphere. The flow rate of the water stream entering the absorber is varied in order to achieve the desired level of ammonia exiting in the vent gas.

Also similar to the carbon capture process, the rich abatement stream is pressurized and heated before it enters the stripper. An ammonia and water stream exits the top of the stripper and the lean abatement stream, which is almost pure water, exits the bottom. The stripper in the abatement process operates similarly to the stripper in the carbon capture process in that there is a condenser at the top of the vessel and a reboiler at the bottom of the vessel.

The lean abatement stream exits the bottom of the stripper at a high temperature and is passed through a heat exchanger with the rich abatement stream in order to heat the rich stream before it enters the stripper. This cool lean stream is then further cooled with cooling water and then reenters the absorber at the beginning of the abatement process. The ammonia and water stream, which exits the top of the stripper at a high temperature, is cooled and then returned to the carbon capture process to the absorber.

Due to limited available data on ammonia abatement, the abatement process was not included in the analysis of the chilled ammonia carbon capture system in this thesis. In Mathias et al, the reboiler duty for the abatement process was found to be approximately 300 Btu/lb-CO_2 captured

(2009). In that paper, the ammonia slip out of the top of the absorber was 242 ppm, and in this thesis, the capture system analyzed had an ammonia slip of 257 ppm. Using a ratio based on Mathias et al results, a reboiler duty of 319.3 Btu/lb-CO_2 captured was assumed for the ammonia abatement process in the model analyzed in this thesis. This abatement reboiler duty was included in the analysis of the capture system's effect on the power plant performance.

Excluded from the analyses in this thesis are the refrigeration loads that would be required to chill the flue gas and the lean ammonia streams before they enter the absorber. These refrigeration cycles would have a significant impact on the plant performance so it is important to note that this is not included when comparing this capture system to other amine-based capture systems.

3.4 ASPEN Plus Model

In order to research and study the chilled ammonia post-combustion carbon capture system, a model of the system was created using ASPEN Plus software. This software allows the modeling of all the components necessary to the carbon capture process which includes the absorber, stripper, heat exchangers and pumps. With this model, various parameters can be varied and tested in order to find their effect on the system as a whole. ASPEN Plus has a sensitivity analysis function that allows the user to put in a range of values for a specific parameter, and the model will run with several values within that range. This allows the user to identify a trend in the data in order to determine if a data input should be increased or decreased to achieve better results. This provides a quick assessment of how an operating parameter can affect the capture system.

The system was designed for the final stream entering the compressor to contain 90% of the original CO_2 in the flue gas. Since some of the CO_2 remains in the lean ammonia stream and some is lost when water is removed from the CO_2 stream, the initial absorption of CO_2 in the absorber must be slightly higher than 90%.

The chemical reactions that occur between the carbon dioxide, ammonia, and water in the carbon capture system are tabulated in Table 3.1 below (Mathias et al, 2009).

Table 3.1 - Chemical Reactions for Chilled Ammonia Carbon Capture System.

$$2H_2O \leftrightarrow OH_3O^+ + OH^-$$
$$CO_2 + 2H_2O \leftrightarrow H_3O^+ + HCO_3^-$$
$$HCO_3^- + H_2O \leftrightarrow H_3O^+ + CO_3^{2-}$$
$$NH_3 + H_2O \leftrightarrow NH_4^+ + OH^-$$
$$NH_3 + HCO_3^- \leftrightarrow NH_2COO^- + H_2O$$
$$NH_4^+ + HCO_3^- \leftrightarrow NH_4HCO_{3(s)}$$

3.4.1 Stream Data

ASPEN Plus allows the user to input stream data for the streams entering the system. Aspen will then calculate the stream data for all intermediate streams and the streams exiting the system based on the equipment specifications. The stream data can be assigned a mass flowrate, a temperature, a pressure, and a molar composition. The two primary streams entering this model were the flue gas coming from the combustion process and the lean ammonia solution.

3.4.1.1 Flue Gas Stream

The data for the entering flue gas stream (Stream 1 in Figure 3.1) is determined by the ASPEN Plus model of the boiler cycle that was discussed in Section 2. This flue gas was at a pressure of 14.7 psia, a temperature of 135°F, and had a flowra te of 6,815,715 lb/hr. The composition (mole-fraction) of the incoming flue gas can be seen in Table 3.2 below.

Table 3.2 - Flue Gas Composition Before Flue Gas Cooler.

Component	**Composition (mole-fraction)**
H_2O	0.1773
CO_2	0.1119
N_2	0.6604
O_2	0.0504
Total	1.0000

The flue gas is cooled to 110°F (stream 2) using co oling water that is at 100°F, and in this cooling process, water is condensed from the flue gas (stream 3). The new flowrate of the 110°F flue gas after the flue gas cooler is 6,230,107 lb/hr, and the new composition of the flue gas can be seen in Table 3.3 below. The heat available from this flue gas cooler is discussed in Section 5.1.

Table 3.3 - Flue Gas Composition After Flue Gas Cooler.

Component	Composition (mole-fraction)
H_2O	0.0493
CO_2	0.1293
N_2	0.7632
O_2	0.0582
Total	1.0000

The only parameter of the flue gas that had to be determined was the temperature that the flue gas would be chilled to prior to entering the absorber (stream 4). A sensitivity analysis was conducted varying the temperature of the flue gas prior to entering the absorber. Varying this temperature had an effect on the amount of CO_2 that was separated from the flue gas and absorbed into the lean ammonia stream. Figure 3.3 below shows how the temperature of the flue gas affects the absorption of the CO_2 into the lean ammonia stream in the absorber.

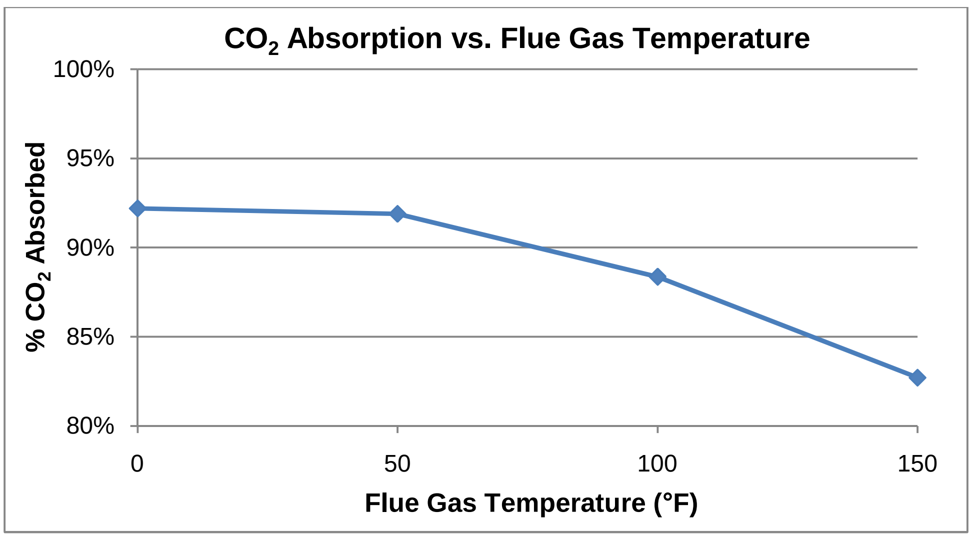

Figure 3.3 - Effect of Flue Gas Temperature on CO_2 Absorption.

The flue gas temperature had an impact on the amount of ammonia slip that occurred out of the top of the absorber, and lower flue gas temperatures resulted in lower amounts of ammonia slip. The flue gas temperature also had an impact on the moisture content of the flue gas before it entered the absorber, and lower flue gas temperatures resulted in lower moisture levels and better CO_2 absorption. A final temperature of 32°F was chose n for the flue gas entering the CO_2 absorber in order to achieve the desired amount of CO_2 absorption.

3.4.1.2 Lean Ammonia Stream

The chilled lean ammonia stream (Stream 13) entering the absorber is a mixture of ammonia, water, and CO_2. The optimal temperature of the lean ammonia stream, as well as the mass fraction of CO_2 in the stream, had to be determined in order to provide maximum CO_2 absorption into the lean ammonia in the absorber. The mass fraction of ammonia in the lean ammonia stream was set at 26% (Mathias et al, 2009). Atmospheric pressure (14.7 psia) was chosen for the lean ammonia stream since the flue gas was at this pressure (Mathias et al, 2009).

In order to determine the details of the lean ammonia stream, several studies were done by the author in order to determine the optimal conditions. In order to narrow down a range of temperatures and CO_2 mass fractions for the stream, published papers on chilled ammonia systems were researched. These papers led to narrowing the possible lean ammonia stream temperature to between 0℉ and 80℉ (Mathias et al, 2009; Darde et al, 2009). The mass fraction of CO_2 in the lean ammonia stream was narrowed to between 0.25 and 0.45 (Mathias et al, 2009; Darde et al, 2009).

These ranges were used as a starting point in determining the details of the lean ammonia stream entering the absorber. A sensitivity analysis was run in ASPEN Plus using various temperatures and weight percents of CO_2 for the lean ammonia stream. Varying these inputs led to changes in the amount of CO_2 that was absorbed.

It was found that lower temperatures for the lean ammonia stream led to higher levels of CO_2 absorption. Figure 3.4 below depicts the effect of the temperature of the stream on the amount of CO_2 that is absorbed.

When varying the mass fraction of CO_2 in the lean ammonia stream, the CO_2 absorption peaks around a mass fraction of 0.36. Figure 3.5 below depicts the affect of the CO_2 mass fraction of the lean ammonia stream on the CO_2 capture rate.

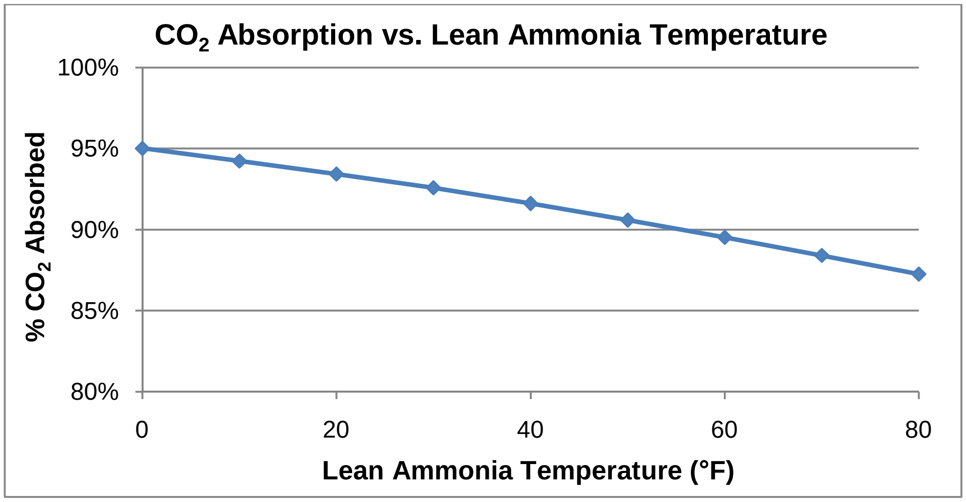

Figure 3.4- Effect of Lean Ammonia Temperature on CO_2 Absorption.

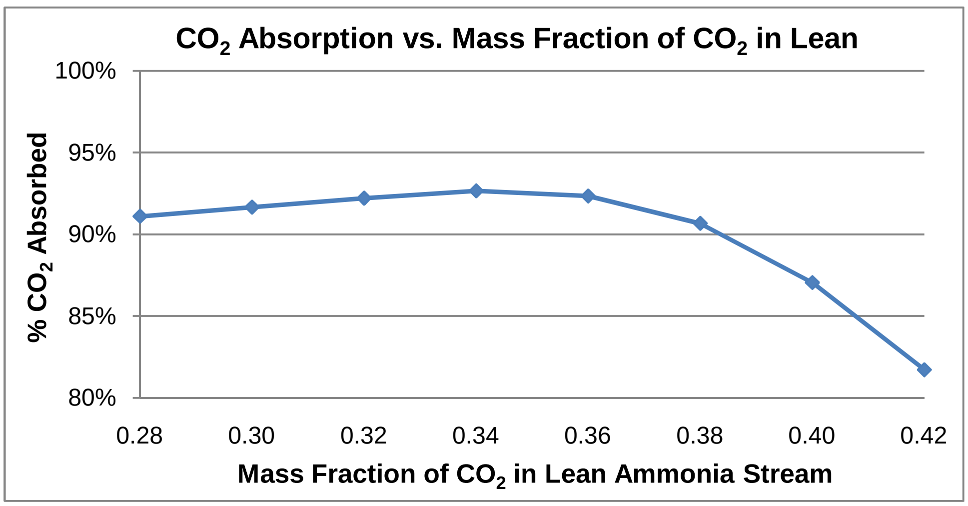

Figure 3.5 - Effect of mass fraction of CO_2 in Lean Ammonia Stream on CO_2 Absorption.

The final details of the lean ammonia stream were determined in order to maximize the CO_2 absorbed and minimize the ammonia slip, and a temperature of 32℉ and a CO_2 mass fraction of 0.37 were chosen.

3.4.1.3 Intermediate and Exiting Streams

Once the parameters were chosen for the two incoming streams, the flue gas and the lean ammonia, the remaining stream data was calculated by ASPEN Plus as it analyzed the capture system. This stream data is dependent on the equipment data that is discussed in the following sections. Table 3.4 below provides data for each stream illustrated previously in Figure 3.1.

Table 3.4 – Stream Data (see Figure 4.1 for Stream Numbers).

	Stream							
	1	**2**	**3**	**4**	**5**	**6**	**7**	**8**
Composition (Mole-Frac)								
CO_2	0.1119	0.1293	0.0000	0.1357	0.0139	0.0038	0.0029	0.0061
H_2O	0.1773	0.0493	1.0000	0.0061	0.0502	0.2216	0.2202	0.2252
NH_3	0.0000	0.0000	0.0000	0.0000	0.0003	0.0001	0.0000	0.0002
N_2	0.6604	0.7632	0.0000	0.7979	0.8731	0.0010	0.0010	0.0010
O_2	0.0504	0.0582	0.0000	0.0609	0.0624	0.0002	0.0002	0.0002
Electrolytes	0.0000	0.0000	0.0000	0.0000	0.0000	0.7734	0.7758	0.7674
Temperature (F)	135.0	110.0	110.0	32.0	99.6	43.6	39.3	150.0
Pressure (psia)	14.7	14.7	14.7	14.7	14.7	14.7	300.0	300.0
Mass Flow (lb/hr)	6,815,715	6,230,107	585,608	6,065,777	4,969,685	5,906,020	5,906,020	5,906,020
Vapor Frac	1.00	1.00	0.00	0.95	1.00	0.00	0.00	0.00

	9	**10**	**11**	**12**	**13**	**14**	**15**	**16**
Composition (Mole-Frac)								
CO_2	0.0084	0.0006	0.0000	0.0000	0.0000	0.8747	0.0009	0.9923
H_2O	0.4514	0.4664	0.4803	0.4802	0.4851	0.0720	0.8428	0.0002
NH_3	0.0594	0.0294	0.0104	0.0105	0.0016	0.0463	0.1564	0.0021
N_2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0059	0.0000	0.0044
O_2	0.0000	0.0000	0.0000	0.0000	0.0000	0.0012	0.0000	0.0009
Electrolytes	0.4808	0.5036	0.5092	0.5093	0.5133	0.0000	0.0000	0.0000
Temperature (F)	257.4	182.1	110.0	110.3	32.0	236.8	300.0	300.0
Pressure (psia)	300.0	300.0	300.0	14.7	14.7	300.0	110.0	110.0
Mass Flow (lb/hr)	4,809,928	4,809,928	4,809,928	4,809,928	4,809,928	1,136,890	231,862	1,096,091
Vapor Frac	0.00	0.00	0.00	0.00	0.00	1.00	0.00	1.00

3.4.2 Equipment Data

ASPEN Plus treats each piece of equipment in the model as a block. Certain operating parameters for each block are input by the user and ASPEN Plus then calculates block details such as heat duties, energy requirements, and stream outlet conditions. Various types of blocks were used in this model in order to simulate each piece of equipment in the chilled ammonia capture system.

3.4.2.1 CO_2 Absorber

Since the absorber is essentially a vessel with several stages that allow the chilled flue gas and lean ammonia streams (stream 4 and 13, respectively) to chemically react with each other, there is no heat or energy input into the vessel. Therefore the design of the vessel essentially affects the amount of CO_2 absorbed into the lean ammonia stream and the amount of ammonia that slips out of the top of the absorber.

The absorber was designed to operate at a constant 14.7 psia, as this is the pressure of both of the streams entering the absorber. The number of stages in the absorber that the streams pass through was the main operating parameter to be determined. A sensitivity analysis was used in order to determine the number of necessary stages in order to allow the streams enough time to react and thoroughly separate the CO_2 from the flue gas. Also, the number of stages affects the amount of ammonia slip, and more stages leads to more ammonia leaving the top of the absorber with the vent gas. The results of the sensitivity analysis led to the absorber being designed with 4 stages.

3.4.2.2 Rich Ammonia Pump

The rich ammonia stream exiting the absorber (stream 6) is pressurized before it is heated and enters the stripper. An optimal pressure for this stream (stream 7) had to be determined in order to maximize the separation of the CO_2 in the stripper. In ASPEN Plus, the exit pressure of the pump can be input as the operating parameter. A range of pressures was determined based on published papers, and then a sensitivity analysis was run in order to determine the optimal pressure in the stripper to achieve maximum CO_2 separation while maintaining a reasonable heat duty in the reboiler of the stripper (Mathias et al, 2009; Darde et al, 2009). The effect of the rich ammonia pressure on the CO_2 separation and the reboiler duty can be seen in Figure 3.6 below. It can be seen in this figure that as the pressure of the stripper is increased, more CO_2 is captured, but the reboiler duty increases as well.

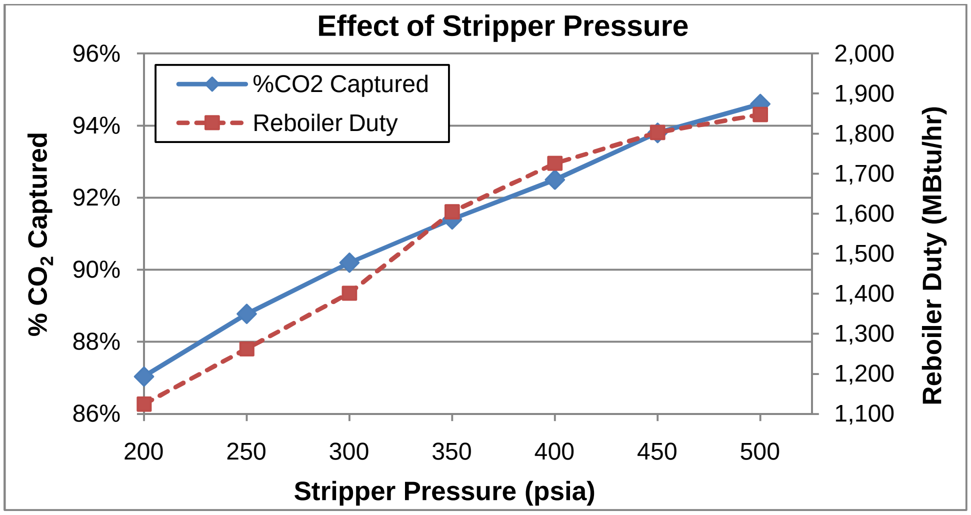

Figure 3.6 - Effect of Stripper Pressure on %CO_2 Captured and Reboiler Duty.

The optimal pressure chosen for the rich ammonia stream was determined to be 300 psia, and the resulting power required for the pump was 1,158.8 kW. The power for this pump, which is pumping liquid prior to the stripper, is much lower than the power required for compressing the CO_2 vapor exiting the stripper. This is one of the main advantages of the ammonia capture system over other amine capture systems that operate at lower pressures leading to more compression power requirements after the stripper.

3.4.2.3 Rich – Lean Heat Exchanger

The pressurized rich ammonia stream is heated before it enters the stripper. This is done by a heat exchanger between the rich and the lean ammonia streams which is modeled as a cross-flow heat exchanger. ASPEN Plus requires stream data for the streams entering the heat exchanger and at least one stream's exit temperature from the heat exchanger. ASPEN Plus determines the necessary size of the heat exchanger in order to achieve the desired temperature output as well as the heat duty of the heat exchanger.

The lean ammonia stream that exits the bottom of the stripper (see Section 3.4.2.4) is at an elevated temperature (stream 9). This hot lean stream has excess heat, so it is used to heat the rich ammonia stream (stream 7) to 150°F before it enters the stripper. The lean ammonia stream (stream 10) is then cooled with cooling water (stream 11), brought back to 14.7 psia (stream 12), chilled to 32°F (stream 13) and then returned to the absorber. The heat available from this lean ammonia cooler is discussed in Section 5.1.

3.4.2.4 CO_2 Stripper

The stripper is the vessel where CO_2 vapor is extracted from the rich ammonia stream, and it was designed to have 3 stages in order to provide enough time and surface area for the CO_2 to separate from the ammonia and water. The settings for the reboiler were designed in order to achieve the necessary CO_2 separation, and the resulting heat duty is 1,051.41 MBtu/hr. This value is positive because it is heat required from an external supply. This heat duty is achieved with 1,029,499lb/hr of 522 °F steam at 87 psia extracted from the steam turbine cycle between LP1 and LP2. This steam extraction results in a loss of 87.4 MW of power from a 587.8 MW power plant, which is nearly 15% of the plant's power output. When the abatement stripper is considered with an assumed heat duty of 350 MBtu/hr, the total steam extraction required for both reboilers is 1,372,207 lb/hr. This total steam extraction results in a loss of 116.5 MW of power, which is nearly 20% of the plant's power. The effects of the capture system on the power plant and ways to improve the plant performance are discussed further in Section 6.

The condenser in the stripper cools the exiting CO_2 stream (stream 14) to 110°F in order to condense water from this stream which is returned to the stripper (stream 15). The resulting CO_2 stream (stream 16) is a nearly pure CO_2 stream. The total heat duty of the condenser is -78.47 MBtu/hr. The negative value denotes heat that is transferred out of the condenser. This heat duty is achieved with 594,703 lb/hr of 100°F cooling water at 300 psia. The heat available from the condenser can be used elsewhere in the plant in order to offset heat requirements. The possible integration of this heat is discussed further in Section 5.

The CO_2 vapor that exits the top of the stripper contains 90% of the original CO_2 in the flue gas, and this vapor is sent to the compression cycle which is discussed in Section 4.

4 CO_2 Compression

After the CO_2 stream exits the top of the stripper, it must be compressed before it can be sequestered or used for enhanced oil recovery. Pressurizing the rich ammonia stream up to 300 psia from 14.7 psia before it enters the stripper greatly reduces the power requirements of the compression process since pressurizing the liquid rich ammonia stream to 300 psia requires less energy than it would take to pressurize the CO_2 vapor to 300 psia. The CO_2 stream entering compression is at 110℉. For the analysis in this thesis, the CO_2 stream was pressurized up to 2,215 psia because this is in the range of pressures required for CO_2 sequestration. Following compression, the CO_2 stream is cooled to 120℉ because this temperature is safely above the saturation temperature at this pressure.

The compression process can be completed with one stage of compression with a post-cooler or with multiple stages with inter-coolers between the stages in addition to the post-cooler. Inter-cooling is done in order to reduce the power needed for the compressors because a vapor is more easily pressurized when it is at a lower temperature. The inter-cooling and post-cooling is achieved using cooling water across a heat exchanger. Compressing in one stage requires a compressor that is designed for higher pressure ratios, which will require more power input. On the other hand, one stage compression results in more heat available from the exiting CO_2 stream, and this heat can be used elsewhere in the plant to offset heat requirements. Compressing with multiple stages requires less power input, but also produces less available heat to use elsewhere in the plant. The thermal integration of the heat produced during compression will be analyzed in Section 5.

Three different compressor designs were analyzed for their power requirements, the useful heat they generate, and their overall effect on the plant's performance. These compressors are designed for CO_2 streams that begin at lower pressures, so not all stages of the compressors were needed for the CO_2 stream from the chilled ammonia capture system that is already at 300 psia.

The compressors along with the inter-coolers and post-coolers were modeled using ASPEN Plus. With input on the compressor specifications such as the pressure ratio and efficiencies, ASPEN Plus is able to simulate the compression cycle and provide data such as the power requirements for the compressor and the CO_2 exit temperature from the compressor.

The coolers were modeled in ASPEN Plus as heat exchangers with 100℉ inlet cooling water. For the compressors with less stages, which had higher CO_2 exit temperatures, the cooling water used had to be at a higher pressure in order to maintain the cooling water as a liquid. A cooling water pressure was chosen based on the CO_2 exit temperature and the saturation pressure at this temperature. The coolers were designed to have the cooling water exit at a temperature 5℉ below the temperature of the hot CO_2 stream entering the heat exchanger. This temperature difference avoids any temperature crossover issues in the heat exchanger. A 15 psia pressure drop in the cooling water stream is assumed across any inter-coolers and post-coolers. Therefore, the cooling water exits the coolers at a lower pressure, and a pump is required to increase the pressure in order to recirculate the cooling water in the coolers. More stages of compression leads to more intercoolers and therefore more pump power for the multiple cooling water streams, however these pump powers are small relative to the overall compressor power. A 5 psia pressure drop in the CO_2 stream is assumed across the coolers. ASPEN Plus is able to determine the minimum flowrate of cooling water in order to cool the CO_2 stream to the desired temperature and to maintain the 5℉ temperature dif ference between the cooling water exiting and the CO_2 entering the heat exchanger. The heat duty of the cooler is also reported by ASPEN Plus.

These ASPEN Plus results were verified with first principle thermodynamic analyses using fluid properties of the streams entering and exiting the compressors and coolers. The fluid properties were found using values from the National Institute of Standards and Technology's Reference Thermodynamic Properties (REFPROP) which provides properties of various fluids depending on input such as the fluid's temperature and pressure. This program can give properties such as enthalpy, entropy, density, and quality. Two properties of a fluid must be known in order to look up the rest of the properties in REFPROP.

In order to verify the power requirements of each compressor, the compressor was treated as a control volume. There is one stream entering the compressor and one stream exiting as well as work being put into the compressor. The entrance temperature (T_1) and pressure (P_1) are known as well as the outlet pressure (P_2). Since the isentropic and mechanical efficiencies of the compressor are given by the manufacturer, the temperature at the outlet (T_2) can be found as well as the work required by the compressor. Figure 4.1 shows the control volume (CV) of the compressor.

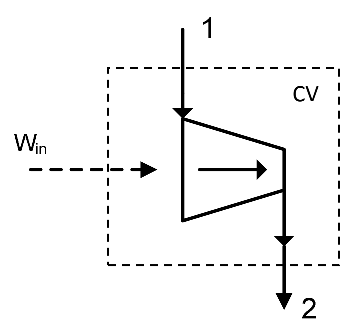

Figure 4.1 - Control Volume of Compressor.

Knowing the P_1 and T_1, the inlet enthalpy (h_1) and entropy (s_1) can be found using REFPROP. For the first step in the calculations, it is assumed that the compressor is isentropic, and therefore the entropy exiting the compressor is the same as the entropy entering the compressor. The outlet stream for the isentropic calculations is designated as 2s in order to not confuse it with the actual exit stream 2. Knowing the P_2 and the exit entropy (s_{2s}), the temperature and enthalpy of stream 2s (T_{2s} and h_{2s}, respectively) can be found using REFPROP. With these values, the ideal work ($W_{s,in}$) for an isentropic compressor can be found using equation 4.1.

$$W_{s,in} = h_{2s} - h_1 \tag{4.1}$$

The isentropic efficiency ($\eta_{isentropic}$) of the compressor is known, so the exit enthalpy (h_2) for the non-isentropic compressor can be found using Equation 4.5 which is developed using Equations

4.2 through 4.4. Once h_2 is found, the actual work ($W_{a,in}$) for the non-isentropic compressor can be found using equation 4.3.

$$\eta_{isentropic} = \frac{W_{ideal}}{W_{actual}} \tag{4.2}$$

$$W_{a,in} = h_2 - h_1 \tag{4.3}$$

$$\eta_{isentropic} = \frac{h_{2s}-h_1}{h_2-h_1} \tag{4.4}$$

$$h_2 = h_1 + \frac{h_{2s}-h_1}{\eta_{isentropic}} \tag{4.5}$$

The mechanical efficiency ($\eta_{mechanical}$) must then be taken into account for the CO_2 compressors. The work the compressor requires (W_{mech}) is found using equation 4.6 below.

$$W_{mech} = \frac{W_{a,in}}{\eta_{mech}} \tag{4.6}$$

To calculate the power requirement of the compressor ($\dot{W}_{in}$), the flowrate of the CO_2 stream entering the compressor must be used in Equation 4.7. This mass flowrate of CO_2 from the capture system that enters the compression cycle is 1,096,118 lb/hr.

$$\dot{W}_{mech} = \dot{m}_{CO2} W_{mech} \tag{4.7}$$

Once the exit enthalpy h_2 has been found with Equation 4.5, the exit temperature (T_2) can be found using REFPROP since P_2 is known. The CO_2 exit temperatures are then needed for the inter-cooler and post-cooler heat exchanger calculations.

The inter-coolers and post-coolers are designed as heat exchangers using cooling water for the cold stream. This is illustrated in Figure 4.2

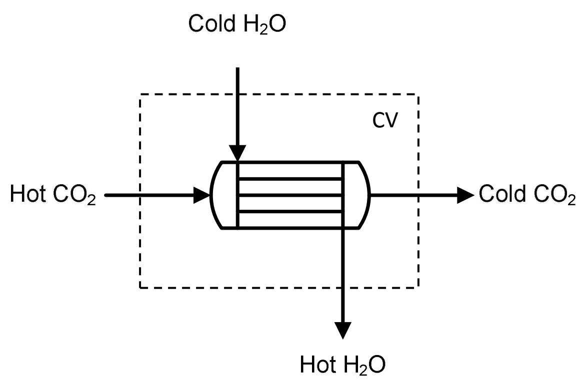

Figure 4.2 – Control Volume of Inter-Coolers and Post-Coolers.

The ASPEN Plus results for the heat exchangers can be verified using an energy balance. The inlet and outlet temperature and pressure of the CO_2 stream are known, as well as the inlet temperature and pressure of the cooling water stream. The outlet pressure of the cooling water is known and the outlet temperature is set to be 5℉ b elow the CO_2 inlet temperature. Knowing the temperature and pressure for the inlet and outlet for the two streams, the enthalpies of the streams entering and exiting can be found using REFPROP. These enthalpy values, along with the CO_2 stream flowrate, can be used in Equation 4.11 to find the necessary cooling water flowrate to meet these requirements. Equation 4.11 was developed by manipulating the energy balance equation, and this is shown in Equations 4.8 through 4.10. The heat duty of the heat exchanger can also be calculated using Equation 4.12.

$$\dot{E}_{in} = \dot{E}_{out} \tag{4.8}$$

$$(\dot{m}h)_{hot\ CO2} + (\dot{m}h)_{cold\ H2O} = (\dot{m}h)_{cold\ CO2} + (\dot{m}h)_{hot\ H2O} \tag{4.9}$$

$$\dot{m}_{CO2}(h_{hot\ CO2} - h_{cold\ CO2}) = \dot{m}_{H2O}(h_{cold\ H2O} - h_{hot\ H2O}) \tag{4.10}$$

$$\dot{m}_{H2O} = \dot{m}_{CO2}\frac{(h_{hot\ CO2} - h_{cold\ CO2})}{(h_{cold\ H2O} - h_{hot\ H2O})} \tag{4.11}$$

$$\dot{Q} = \dot{m}_{CO2}(h_{hot\ CO2} - h_{cold\ CO2}) \tag{4.12}$$

The equations outlined in this section, along with the data from REFPROP, were used to verify the ASPEN Tech calculations for the different compressors and coolers used for the CO_2 compression cycle. The results of the compression cycle and coolers given by ASPEN Tech and calculated using REFPROP and first principle analyses are compared in tables in Appendix B.2.

4.1 Ramgen

The first compressor design analyzed was Ramgen. Each stage of a Ramgen compressor is expected to be able to handle a pressure ratio of 10. With an inlet CO_2 pressure of 300 psia, only one stage of compression is needed for a chilled ammonia scrubber. This one stage of compression is then followed by a post-cooler to cool the CO_2 to 120°F. The CO_2 is compressed to 2,220 psia, because it is assumed that there will be a 5 psia pressure drop across the post-cooler. Compressing from 300 to 2,220 psia results in a compressor pressure ratio of 7.4, and as noted, the compressor is designed for pressure ratios up to 9.814. This compressor has an isentropic efficiency of 0.850 and a mechanical efficiency of 0.9701. Compressing the CO_2 stream in one stage with the Ramgen design requires a power input of 23,894 kW.

The CO_2 stream exits the compressor at 471.4°F before it enters the post-cooler. The CO_2 stream is then cooled to 120°F using cooling water that is at 100°F and 550 psia. This high pressure for the cooling water was necessary because the saturation pressure at 470°F is 514.1 psia. The cooler was designed to have the cooling water exit at a temperature of 466.4°F since this is 5°F below the hot CO_2 stream temperature. The amount of cooling water required is 459,225 lb/hr and the resulting heat duty of the post-cooler is 174.14 MBtu/hr.

Table 4.1 below summarizes the operating parameters and results of the Ramgen compressor. The manufacturer's specifications for this compressor and the comparison between the results from ASPEN Plus and the results from first principle calculations using REFPROP data can be found in Appendix B.1 and B.2, respectively.

Table 4.1 - Operating Parameters and Results Summary for Ramgen Compressor.

Compressor

	CO_2
T_{in} (℉)	100
T_{out} (℉)	471.4
P_{in} (psia)	300
P_{out} (psia)	2,220
Pressure Ratio	7.3
Isentropic Efficiency	0.8500
Mechanical Efficiency	0.9702
Power (kW)	23,894

Post-Cooler

	CO_2	Cooling Water
T_{in} (℉)	471.4	100.0
T_{out} (℉)	120.0	466.4
P_{in} (psia)	2,220	550
P_{out} (psia)	2,215	535
Flowrate (lb_m/hr)	1,096,118	459,225
Heat Duty (MBtu/hr)	174.14	

Figure 4.3 below shows the compression and cooling specific for the chilled ammonia capture system.

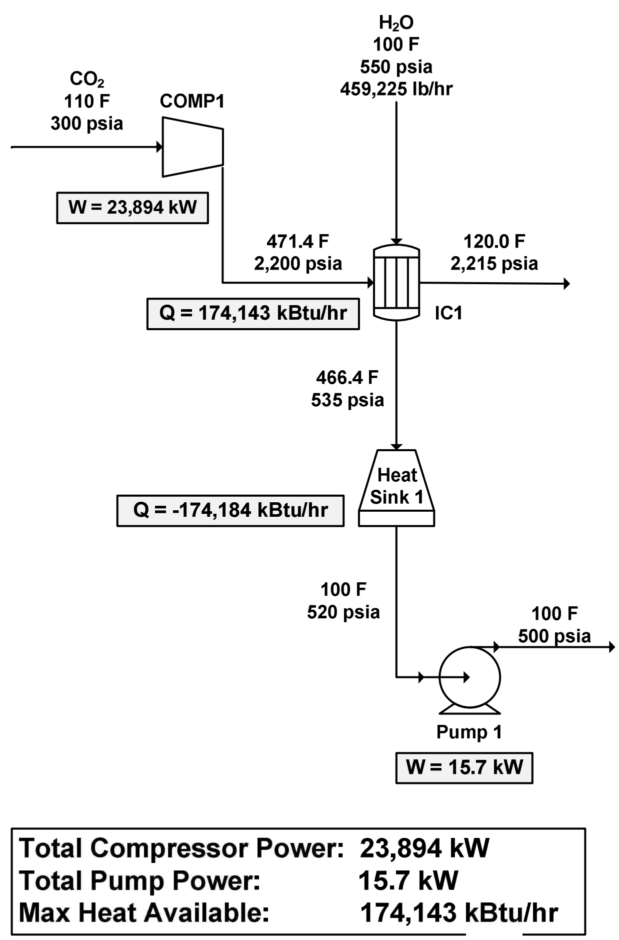

Figure 4.3 - Process Diagram for Ramgen Compressor with 1 Stage.

4.2 Inline 4

The second compressor design analyzed was an inline compressor (IL4). The number 4 signifies a specific manufacturer's compressor model, the identity of which is on file at the Energy Research Center. This compressor is designed for three stages of compression, but for the chilled ammonia system, only two of the stages are necessary. The manufacturer's second and third stage were used which provided pressure ratios of 6.05 and 1.22, respectively. The manufacturer's third stage is designed for a pressure ratio of 3.7, but only 1.22 is needed after the first stage's 6.05 pressure ratio. An inter-cooler is used between the two stages, but a post-cooler is not necessary because the CO_2 exiting the second compressor is only at 147°F due to the small pressure ratio in the second stage.

The first stage of compression raises the pressure of the CO_2 to 1,815 psia, and this stage has an isentropic efficiency of 0.8188 and a mechanical efficiency of 0.9920. Since there is no post-cooler and therefore no pressure drop in the post-cooler, the second stage only has to compress the CO_2 to 2,215 psia. This second stage has an isentropic efficiency of 0.8114 and a mechanical efficiency of 0.9980. The first stage of compression requires 21,225.8 kW of power and the second stage requires 785.4 kW of power. This results in a total power requirement of 22,011 kW for the inline compressor.

The intercooler was designed to cool the CO_2 to 110°F since this temperature is just above the saturation temperature for the CO_2. The CO_2 stream exits the first compressor at 438.2°F, and therefore the inter-cooler is designed to have the cooling water exit at 433.2°F. For this compressor design, the cooling water used was at 100°F and 400 psia for the inter-cooler. The saturation pressure of water at 430°F is 343.4 psia , so a pressure greater than this had to be used for the cooling water in order to maintain the cooling water as a liquid in the heat exchanger. The amount of cooling water required is 492,072 lb/hr and the resulting heat duty of the inter-cooler is 168.45 MBtu/hr. Table 4.2 below summarizes the operating parameters and results of the inline compressor.

Table 4.2 - Operating Parameters and Results Summary for Inline 4 Compressor.

Compressor

Stage 1	**CO_2**
T_{in} (F)	110
T_{out} (F)	438.2
P_{in} (psia)	300
P_{out} (psia)	1,815
Pressure Ratio	6.05
Isentropic Efficiency	0.8188
Mechanical Efficiency	0.9920
Power (kW)	21,226
Stage 2	**CO_2**
T_{in} (F)	110
T_{out} (F)	121.3
P_{in} (psia)	1,810
P_{out} (psia)	2,215
Pressure Ratio	1.22
Isentropic Efficiency	0.8114
Mechanical Efficiency	0.9980
Power (kW)	785.4
Total Power (kw)	**22,011**

Inter-Cooler

	CO_2	**Cooling Water**
T_{in} (F)	438.2	100.0
T_{out} (F)	110.0	433.2
P_{in} (psia)	1,815	400
P_{out} (psia)	1,810	385
Flowrate (lb_m/hr)	1,096,118	492,072
Heat Duty (MBtu/hr)	168.45	

Figure 4.4 shows the compression and cooling specific for the chilled ammonia capture system.

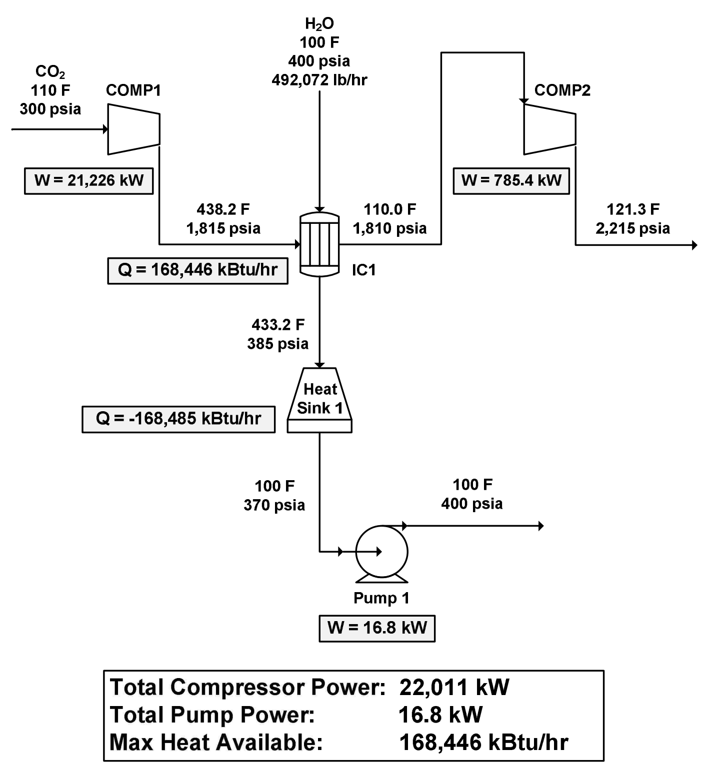

Figure 4.4 - Process Diagram for Inline 4 Compressor with 2 Stages.

The manufacturer's specifications for this compressor and the comparison between the results from ASPEN Plus and the results from first principle calculations using REFPROP data can be found in Appendix B.1 and B.2, respectively.

4.3 Integrally-Geared 1

The third compressor design analyzed was an integrally-geared (IG1) compressor. The number 1 signifies a specific manufacturer's compressor model, the identity of which is on file at the Energy Research Center. This compressor is designed for seven stages of compression, but for the chilled ammonia system, only four of the stages are necessary. The manufacturer's last four stages were used which provided pressure ratios of 2.112, 1.681, 1.622, and 1.542, respectively, to raise the pressure from 300 psia to 2,220 psia.

The first stage of compression raises the pressure of the CO_2 to 572.4 psia, and this stage has a polytropic efficiency of 0.8343. The second stage of compression raises the pressure of the CO_2 to 950.0 psia, and this stage has a polytropic efficiency of 0.8846. The third stage of compression raises the pressure of the CO_2 to 1,505.3 psia, and this stage has a polytropic efficiency of 0.8781. The final stage of compressions raises the pressure of the CO_2 to 2,220 psia, and this stage has a polytropic efficiency of 0.8585.

Two different designs for the inter-cooling and post-cooling were analyzed. One design followed the manufacturer's specifications for inter-cooling (IG1-A), and the other design inter-cooled to the lowest temperature possible (IG1-B). The pressures exiting each stage of compression are the same for both designs. For the cooling that follows the manufacturer's specifications, their pressure drops across the coolers were used, and these drops varied between the different coolers. For the lowest temperature cooling, a pressure drop of 5 psia was assumed across all the coolers as it was for the Ramgen and inline designs. Figures 4.5 and 4.6 illustrate the IG1-A and IG1-B compressor designs, respectively.

Inter-cooling to lower temperatures has the advantage of lowering the power requirements of the compressor, but it also requires larger heat exchangers and gives off less heat that can be integrated elsewhere in the plant. These two different designs are analyzed separately and then compared, along with the other compressors, in Section 4.4.

The manufacturer's specifications for the IG compressor can be found in Appendix B.1.

4.3.1 Manufacturer's Specifications for Cooling (IG1-A)

The manufacturer only inter-cools to 149°F in the intercoolers, and does not inter-cool at all between the last two stages. The manufacturer does not specify a post-cooler, so after the final stage, the CO_2 was cooled to 120°F as it was for the previous compressor designs. All the inter-coolers and the post-cooler were designed to use cooling water that was at 100°F and 300 psia.

The CO_2 stream exits the first compressor at 217.6°F and is cooled to 149°F. The cooling water in the first inter-cooler is designed to exit at 212.6°F, and in order to achieve this, 175,417 lb/hr of cooling water is required. This results in a heat duty of 19.75 MBtu/hr.

The CO_2 stream exits the second compressor at 236.3°F and is cooled to 149°F. The cooling water in the second inter-cooler is designed to exit at 231.6°F, and in order to achieve this, 222,056 lb/hr of cooling water is required. This results in a heat duty of 29.19 MBtu/hr.

The manufacturer does not use an inter-cooler between the third and fourth compressor. The CO_2 stream exits the third compressor at 230.9°F and then enters the fourth compressor at this temperature. The CO_2 stream then exits the fourth compressor at 301.0°F and is cooled to 120°F. The cooling water in this post-cooler is designed to exit at 296.1°F, and in order to achieve this, 572,086 lb/hr of cooling water is required. This results in a heat duty of 112.85 MBtu/hr.

With the manufacturer's specifications for cooling, the compressor power requirements for each stage are 6,737 kW, 5,089 kW, 4,258 kW, and 4,085 kW, respectively which makes the total power 20,168 kW. The total heat duty is 161.79 MBtu/hr and the total cooling water needed is 969,560 lb/hr.

Table 4.3 summarizes the operating parameters and results of the IG 1-A compressor.

Table 4.3 - Operating Parameters and Results Summary for IG 1-A Compressor.

Compressor

Stage 1	**CO_2**
T_{in} (F)	110
T_{out} (F)	217.6
P_{in} (psia)	300
P_{out} (psia)	572.4
Pressure Ratio	1.908
Polytropic Efficiency	0.8343
Isentropic Efficiency	0.8316
Mechanical Efficiency	0.9700
Power (kW)	6,737.1
Stage 2	**CO_2**
T_{in} (F)	149.0
T_{out} (F)	236.3
P_{in} (psia)	565.1
P_{out} (psia)	950
Pressure Ratio	1.681
Polytropic Efficiency	0.8846
Isentropic Efficiency	0.8898
Mechanical Efficiency	0.9700
Power (kW)	5,088.5
Stage 3	**CO_2**
T_{in} (F)	149
T_{out} (F)	230.9
P_{in} (psia)	928.2
P_{out} (psia)	1,505.3
Pressure Ratio	1.622
Polytropic Efficiency	0.8781
Isentropic Efficiency	0.9071
Mechanical Efficiency	0.9700
Power (kW)	4,258.2
Stage 4	**CO_2**
T_{in} (F)	230.9
T_{out} (F)	301.0
P_{in} (psia)	1505.3
P_{out} (psia)	2,220
Pressure Ratio	1.475
Polytropic Efficiency	0.8585
Isentropic Efficiency	0.9175
Mechanical Efficiency	0.9700
Power (kW)	4,084.6
Total Power (kW)	20,168.3

Inter-Coolers and Post-Cooler

IC1	**CO_2**	**Cooling Water**
T_{in} (F)	217.6	100.0
T_{out} (F)	149.0	212.6
P_{in} (psia)	572.4	300
P_{out} (psia)	565.1	285
Flowrate (lb_m/hr)	1,096,118	175,417
Heat Duty (MBtu/hr)	19.8	

IC2	**CO_2**	**Cooling Water**
T_{in} (F)	236.3	100.0
T_{out} (F)	149.0	231.3
P_{in} (psia)	950.0	300
P_{out} (psia)	928.2	285
Flowrate (lb_m/hr)	1,096,118	222,056
Heat Duty (MBtu/hr)	29.2	

PC1	**CO_2**	**Cooling Water**
T_{in} (F)	301.0	100.0
T_{out} (F)	120.0	296.0
P_{in} (psia)	2,220	300
P_{out} (psia)	2,215	285
Flowrate (lb_m/hr)	1,096,118	572,086
Heat Duty (MBtu/hr)	112.9	

Total Heat Duty (MBtu/hr)	161.8

A comparison of the results from ASPEN Plus and the results from first principle calculations using REFPROP data can be found in Appendix B.2.

Figure 4.5 below shows the compression and cooling specific for the IG 1-A compressor design.

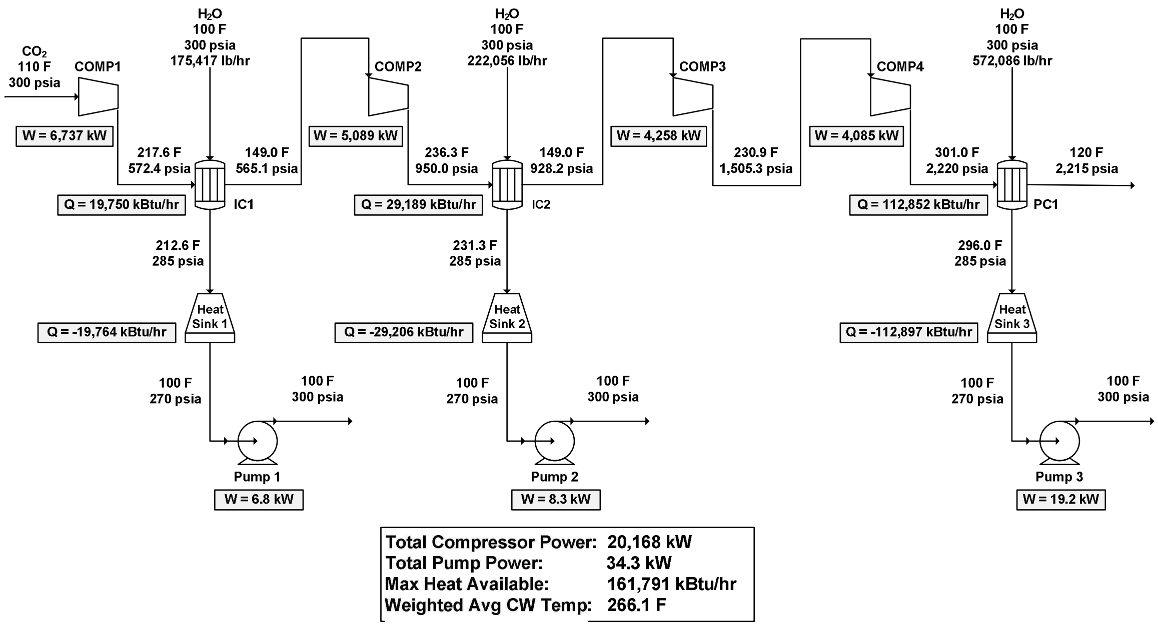

Figure 4.5 - Process Diagram for IG 1-A Compressor with 4 Stages.

4.3.2 Inter-Cooling to 110°F (IG1-B)

Analyses were also carried out for the integrally-geared compressor with the CO_2 inter-cooled to 110°F between compressor stages. After the final s tage, the CO_2 was cooled to 120°F as it was for the previous compressor designs. All the inter-coolers and the post-cooler were designed to use cooling water that was at 100°F and 300 psia.

Inter-cooling to lower temperatures reduces the power requirements of the compressor. On the other hand, these lower inter-cooling temperatures lead to less available heat that can be used elsewhere in the plant, which is discussed further in Section 5. The lower inter-cooling temperatures also require larger heat exchangers which increase the footprint of this compressor design.

The CO_2 stream exits the first compressor at 217.6°F and i s cooled to 110°F. The cooling water in the first inter-cooler is designed to exit at 212.6°F, and in order to achieve this, 280,662 lb/hr of cooling water is required. This results in a heat duty of 31.60 MBtu/hr.

The CO_2 stream exits the second compressor at 194.6°F and is cooled to 110°F. The cooling water in the second inter-cooler is designed to exit at 189.6°F, and in order to achieve this, 357,502 lb/hr of cooling water is required. This results in a heat duty of 31.99 MBtu/hr.

The CO_2 stream exits the third compressor at 187.0°F and is cooled to 110°F. The cooling water in the third inter-cooler is designed to exit at 182.0°F, and in order to achieve this, 873,125 lb/hr of cooling water is required. This results in a heat duty of 71.47 MBtu/hr.

The CO_2 stream exits the final compressor at 136.5°F and is cooled to 120°F. The cooling water in this post-cooler is designed to exit at 131.5°F, and in order to achieve this, 449,063 lb/hr of cooling water is required. This results in a heat duty of 14.08 MBtu/hr.

With inter-cooling to 110°F, the compressor power requirements for each stage are 6,737 kW, 4,535 kW, 3,522 kW, and 1,552 kW, respectively, which makes the total power 16,347 kW. The total heat duty is 149.14 MBtu/hr and the total cooling water needed is 1,960,352 lb/hr. Table 4.4 summarizes the operating parameters and results of the IG 1-B compressor.

Table 4.4 - Operating Parameters and Results Summary for IG 1-B Compressor.

Compressor

Stage 1	**CO_2**
T_{in} (F)	110
T_{out} (F)	217.6
P_{in} (psia)	300
P_{out} (psia)	572.4
Pressure Ratio	1.908
Polytropic Efficiency	0.8343
Isentropic Efficiency	0.8316
Mechanical Efficiency	0.9700
Power (kW)	6,737.1

Stage 2	**CO_2**
T_{in} (F)	110.0
T_{out} (F)	194.6
P_{in} (psia)	567.4
P_{out} (psia)	950
Pressure Ratio	1.674
Polytropic Efficiency	0.8846
Isentropic Efficiency	0.8898
Mechanical Efficiency	0.9700
Power (kW)	4,535.4

Inter-Coolers and Post-Cooler

IC1	**CO_2**	**Cooling Water**
T_{in} (F)	217.6	100.0
T_{out} (F)	110.0	212.6
P_{in} (psia)	572.4	300
P_{out} (psia)	567.4	285
Flowrate (lb_m/hr)	1,096,118	280,662
Heat Duty (MBtu/hr)	31.9	

IC2	**CO_2**	**Cooling Water**
T_{in} (F)	194.6	100.0
T_{out} (F)	110.0	189.6
P_{in} (psia)	950.0	300
P_{out} (psia)	945.0	285
Flowrate (lb_m/hr)	1,096,118	357,502
Heat Duty (MBtu/hr)	32.0	

Continued on next page

Stage 3	CO_2
T_{in} (F)	110
T_{out} (F)	187.0
P_{in} (psia)	845
P_{out} (psia)	1,505.3
Pressure Ratio	1.593
Polytropic Efficiency	0.8781
Isentropic Efficiency	0.9071
Mechanical Efficiency	0.9700
Power (kW)	3,522.0
Stage 4	**CO_2**
T_{in} (F)	110
T_{out} (F)	136.5
P_{in} (psia)	1500.3
P_{out} (psia)	2,220
Pressure Ratio	1.480
Polytropic Efficiency	0.8585
Isentropic Efficiency	0.9175
Mechanical Efficiency	0.9700
Power (kW)	1,522.3
Total Power (kW)	16,346.8

IC3	CO_2	Cooling Water
T_{in} (F)	187.0	100.0
T_{out} (F)	110.0	182.0
P_{in} (psia)	1,505.3	300
P_{out} (psia)	1,500.3	285
Flowrate (lb_m/hr)	1,096,118	873,125
Heat Duty (MBtu/hr)	71.5	

PC1	CO_2	Cooling Water
T_{in} (F)	136.5	100.0
T_{out} (F)	120.0	131.5
P_{in} (psia)	2,220	300
P_{out} (psia)	2,215	285
Flowrate (lb_m/hr)	1,096,118	449,063
Heat Duty (MBtu/hr)	14.1	

Total Heat Duty (MBtu/hr)	149.1

Figure 4.6 shows the compression and cooling for the IG 1-B compressor design.

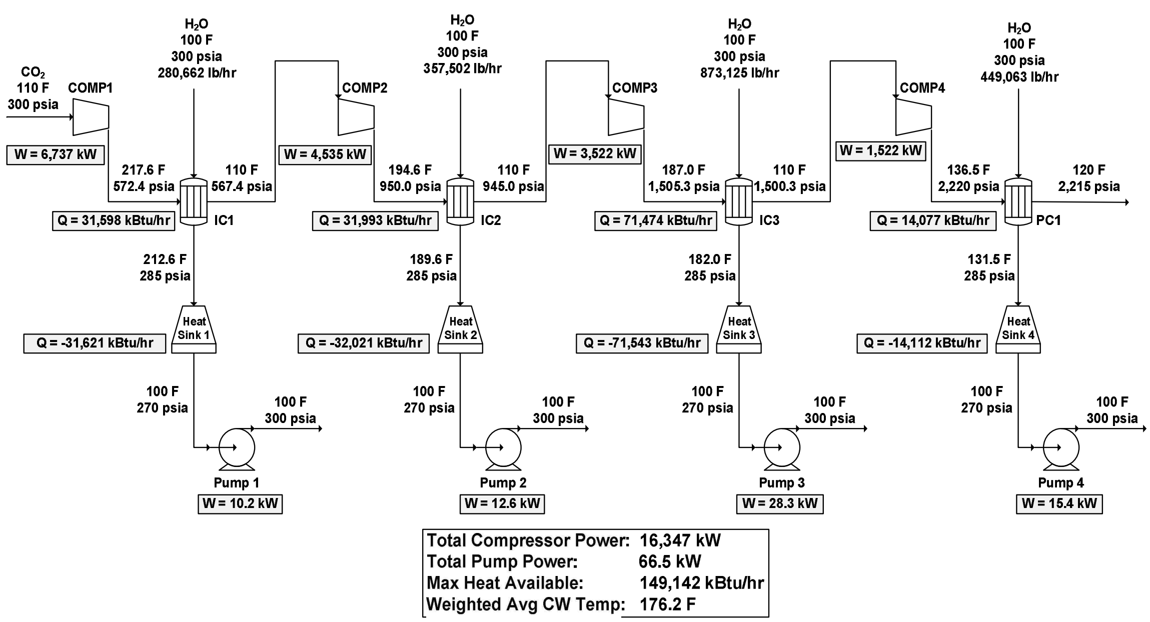

Figure 4.6 - Process Diagram for IG 1-B Compressor with 4 Stages.

A comparison of the results from ASPEN Plus and the results from first principle calculations using REFPROP data can be found in Appendix B.2.

4.4 Compressor Comparison

The Ramgen compressor requires the most power since it only uses one stage for compression. Alternatively, the Ramgen creates the most heat that can be used elsewhere in the plant to offset heat requirements. The Ramgen also has the smallest footprint in a plant since there is only one stage and one cooler, and it also requires the least amount of cooling water. The Inline 4 compressor requires less power than the Ramgen, but it also has less available heat for use elsewhere in the plant and requires more space within the plant. The integrally geared compressor requires the least amount of power, but it also has the least amount of available heat for use elsewhere in the plant and requires very large heat exchangers for the coolers.

Following the manufacturer's specifications for the integrally geared compressor, more power is required, but more heat is available for use from the coolers and less cooling water is used. When inter-cooling is done to 110°F, less power is required due to the lower temperatures in the compressors, but more cooling water is used and less heat is available from the coolers.

When no heat integration is being considered, the integrally geared compressor with inter-cooling to 110°F is the best option due to its low power requirements. Integrating the heat created from compression and its effects on the plant's performance will be examined more in Section 5.

Table 4.5 below contains the results for the different compressors.

Table 4.5 - Comparison of Compressor Designs.

	Ramgen	Inline	IG1-A	IG1-B
Total Compressor Power (kW)	23,894.4	22,011.2	20,168.3	16,346.8
Total Pump Power (kW)	15.7	16.8	34.3	66.5
Max Heat Available (MBtu/hr)	174.1	168.4	161.8	149.1
Cooling Water Used (lb/hr)	459,225	492,072	969,560	1,960,352
Mass Weighted Average Cooling Water Temperature (°F)	466.4	433.2	266.1	176.2

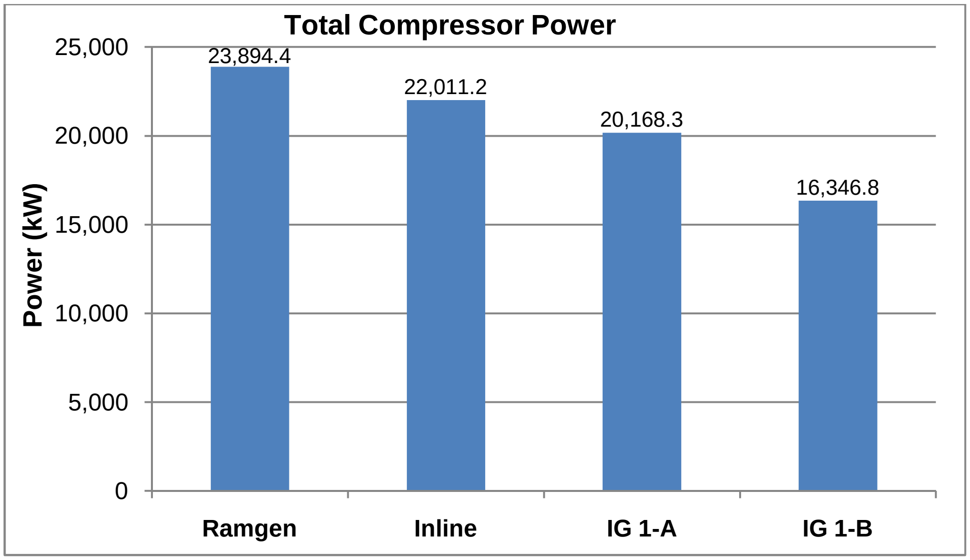

Figure 4.7 - Comparison of Compressor Power for Different Compressor Designs.

Figure 4.7 shows the trend for the total compressor power for each different compressor design. The effects of these compressors and the chilled ammonia capture system on the power plant are discussed below in Section 5.

5 Integrating NH_3 - CO_2 Capture System with Power Plant

Integrating the chilled-ammonia system with the power plant provides information on how the addition of carbon capture affects the performance of the power plant. The chilled ammonia carbon capture system was analyzed in Section 3 and the compressor cycle was analyzed in Section 4 for their individual power and heat loads. Installing this capture system and compressor cycle to the back-end of a power plant causes a decrease in the net power output of a plant. These analyses do not include the refrigeration loads that would be required to chill the lean ammonia and flue gas streams entering the absorber in the capture system.

5.1 Effects of Capture System on Plant Performance

In the capture system, the compressors require power input, along with the rich ammonia stream pump which requires power to operate. For the 587.8 MW plant analyzed, the compressors require between 16.3 and 23.9 MW of power, depending on the compressor design, and this is approximately 2.8 to 4.1% of the plant's power. The rich ammonia pump requires 1.2 MW, which has a small impact compared to the other equipment.

There are also the two reboilers in the capture system (CO_2 separation and NH_3 abatement) that require a heat input which would normally come from steam extraction from the steam turbine cycle, which reduces the net power output. The total heat duty of the two reboilers is 1,401.41 MBtu/hr which must be met with a steam extraction of 1,372,207 lb/hr from the steam turbine cycle between LP1 and LP2. The steam extraction required for the reboilers causes a loss of 116.5 MW of power from a 587.8 MW power plant, which is 19.8% of the plant's power output.

Depending on compressor design, the chilled-ammonia carbon capture system results in a loss of overall power output of between 132.3 and 139.8 MW from the 587.8 MW plant, which equates to between 22.5 and 23.8% of the plant's total power output. This is a significant amount of power lost for the capture of CO_2.

Without a post-combustion carbon capture system, the plant analyzed in this thesis had a unit heat rate of 9,362 Btu/kWh and a unit efficiency of 36.45%. The effect of the capture system on the plant performance is tabulated below in Table 5.1 for each compressor design analyzed when no heat integration is considered. The loss in net power includes the compressor power, the pump power required for the cooling water in the compressor cooler(s), the rich amine liquid pump in the capture system, and the loss of power due to the steam extraction for the stripper reboilers. The power required to chill the flue gas and lean ammonia streams to 32°F would also decrease the plant's net power, but that is not included in the analysis in this thesis. More detail of the effects on plant performance can be found in Appendix C.1.

Table 5.1 - Effect on Plant Performance for Each Compressor Design. (Note: Does not include the power required to chill the flue gas and lean ammonia streams to 32°F.)

	No Capture	Ramgen	Inline 4	IG 1-A	IG 1-B
Net Power (kW)	587,804	447,980	449,862	451,687	455,477
Δ Net Power (kW)	--	-138,436	-137,942	-134,116	-132,295
Unit HR (Btu/kWh)	9,291	12,191	12,140	12,091	11,991
Δ HR (Btu/kWh)	--	2,900	2,849	2,800	2,700
% HR Lost	--	31.21%	30.66%	30.14%	29.05%
Unit Efficiency (%)	36.72%	27.99%	28.10%	28.22%	28.46%
Δ Unit Efficiency (%)	--	-8.73%	-8.62%	-8.50%	-8.26%

In parts of the capture system where streams are cooled, there is heat transferred to cooling water, and this heat is available for use within the plant to offset heat requirements and improve plant performance. Using this heat elsewhere in the plant helps offset some of the impact that the capture system has on the power plant. The main contributor of available heat analyzed in this thesis is the compressor for the CO_2 stream, and the heat available from the cooler(s) in each compressor design was analyzed for thermal integration. The amount of heat available from the compressor cooler(s) varies for each compressor design and these values can be found in Sections 4.1 through 4.4. The stripper condenser for the CO_2 separation also has heat available that was analyzed in Section 5.2.4. The condenser provides 78.47 MBtu/hr of heat to 594,566 lb/hr of cooling water.

The heat requirements of the power plant and the capture system are shown in Table 5.2. The heat available for use from the carbon capture system and the various compressors is shown in Table 5.3. The heat available from the flue gas cooler, the lean amine cooler, and the stripper condenser are independent of the compressor design and remain constant when the different compressors are analyzed. The heat available can be integrated into the power plant where heat is required, and several methods of integrating the heat are analyzed in the following sections.

Table 5.2 - Heat Available vs. Heat Required Without Compression Cycle.

Heat Required		
Heat Sink	**Required Outlet Temperature (F)**	**Heat Required (Btu/hr)**
FWH1	151.9	173,265,266
FWH2	193.8	130,649,607
FWH3	231.4	120,226,508
FWH5	363.6	215,945,146
FWH6	414.2	228,582,814
Stripper Reboiler (CO_2 Separation)	264.6	1,051,404,890
Stripper Reboiler (NH_3 Abatement)	--	350,000,000
Total		**2,270,074,231**

Table 5.3 - Heat Available from the Capture System and Various Compressor Designs.

Heat Available			
	Hot CW Temperature (°F)	CW Flowrate (lb/hr)	Heat Available (Btu/hr)
Flue Gas Cooler	130.0	8,665,618	345,839,936
Lean Amine Cooler	177.1	3,148,771	242,163,669
Stripper Condenser (CO_2 Separation)	231.8	594,566	78,471,416
Total			**666,475,021**

Compression Heat Available			
Heat Source	**Hot CW Temperature (°F)**	**CW Flowrate (lb/hr)**	**Heat Available (Btu/hr)**
Ramgen Post-Cooler	466.4	459,225	**174,143,548**
Inline Inter-Cooler	433.2	492,072	**168,445,882**
IG 1-A			
IC1	212.6	175,417	19,749,668
IC2	231.3	222,056	29,188,834
PC	296.0	572,086	112,852,009
Total			**161,790,511**
IG 1-B			
IC1	212.6	280,662	31,592,543
IC2	189.6	357,502	31,993,147
IC3	182.0	873,125	71,473,676
PC	131.5	449,063	14,076,629
Total			**149,141,994**

5.2 Thermal Integration Options

There are many options for thermal integration within the plant with the various heat sources and heat sinks. Three methods of thermal integration were analyzed for the power plant in this thesis. The first option analyzed was to offset a portion of the heat duty of the feedwater heaters (FWHs). With the cooling water temperatures available, thermal integration could be analyzed for FWH-1 through FWH-3. The second option analyzed was to offset a portion of the heat duty of the stripper reboiler for CO_2 separation in the chilled ammonia system. The ammonia abatement stripper would also utilize a reboiler, but since this process was not modeled, the heat available from the compressors was only integrated to the CO_2 separation reboiler in this thesis. The third option was to use the available heat to partially dry the coal before it is sent to the boiler, which would improve boiler efficiency. Each of these options was analyzed individually for each compressor design in order to determine the improvement to the plant's performance when thermal integration is considered.

The heat available from the CO_2 stripper condenser was also analyzed for thermal integration, but the condenser only had enough heat available to integrate to the FWHs. Some combinations of thermal integration with the compressor and condenser heat were also considered in order to use a larger portion of the available heat to improve the plant's performance.

In the compression cycle and stripper condenser, the water used to cool the CO_2 streams was termed cooling water. After the cooling water cools the CO_2, it becomes a hot fluid and it is referred to as "hot cooling water from the capture system" in the following discussion. In the thermal integration diagrams in the following sections, the cooling water is often designated as CW.

As with previous heat exchanger analyses, a 15 psia cooling water pressure drop is assumed for each heat exchanger the water passes through. Since the hot cooling water is coming from heat exchangers within the capture system, it has already dropped in pressure and after thermal integration, the pressure is even lower due to this pressure drop. The more heat exchangers the hot cooling water passes through, the more the power requirement of the cooling water pump(s) increases.

5.2.1 Feedwater Heaters

As discussed in Section 2, the boiler feedwater is pre-heated in a series of feedwater heaters. These feedwater heaters are heat exchangers which heat the feedwater using steam extractions from the different turbines in the steam turbine cycle. Integrating the available heat from the carbon capture system into the lower temperature feedwater heaters allows some of the steam extractions to either be reduced or eliminated. This allows more steam to flow through the low pressure turbines, thereby generating more power. The first three feedwater heaters (FWH-1 through FWH-3) are at low enough temperatures that available heat from the carbon capture process can be utilized to offset the steam extraction requirements.

The first thermal integration analyzed was to the first feedwater heater (FWH-1), which has the lowest feedwater exit temperature, but has the highest heat duty of the three feedwater heaters analyzed. For all of the compression cases, there was only enough heat available to reduce the steam extraction for FWH-1, but not enough heat to completely replace the extraction.

In order to use the available heat to offset some of the heat duty of FWH-1, the feedwater heater must be divided into two heat exchangers. The first heat exchanger (FWH-1A) heats the feedwater as much as possible with the hot cooling water stream from the compressor. The first heat exchanger can be designed to have the cooling water stream exit at 100℉ so that it is at the proper temperature to return to the compressor cooler(s) to be used again for cooling water. This also allows all of the heat available in the stream to be used in FWH-1A.

The second heat exchanger (FWH-1B) heats the feedwater the rest of the way to the designated outlet temperature, and less extraction steam is now needed for FWH-1B than was needed originally for FWH-1. A pump is then required for the cooling water after it exits FWH-1A in order to increase the pressure before the cooling water is sent back to the compressor. Figure 5.1 below illustrates the design of FWH-1A and FWH-1B.

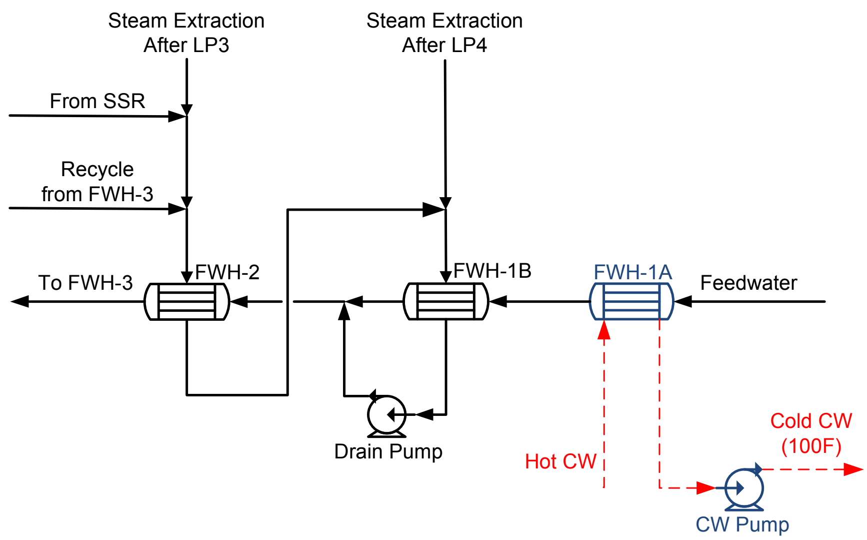

Figure 5.1 - Thermal Integration to FWH-1.

The effect on the plant's performance for this thermal integration to FWH-1 is discussed in the following sections for each compressor design. Figure 5.2 below shows this thermal integration to FWH-1 within the complete steam turbine cycle.

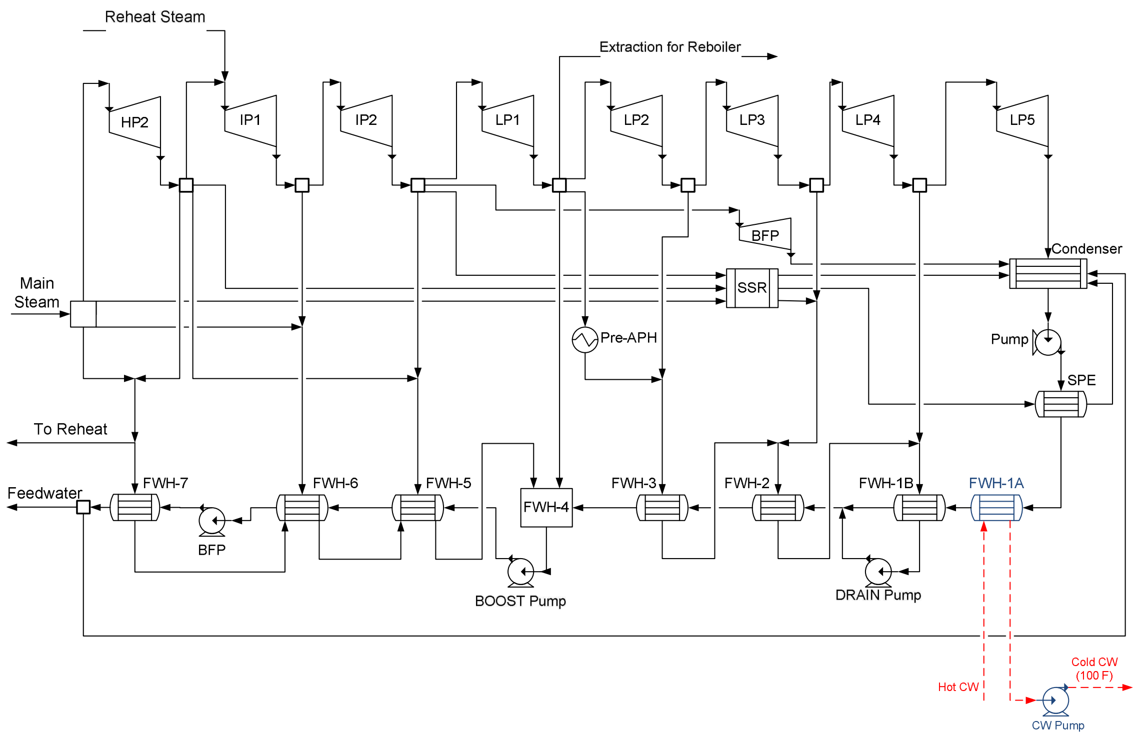

Figure 5.2 – Steam Turbine Cycle for Thermal Integration to FWH-1.

The second heat integration analyzed was to the second feedwater heater (FWH-2). For the Ramgen and the Inline 4 compressor designs, there was enough heat available in the hot cooling water to completely replace FWH-2 and eliminate the need for a steam extraction for this FWH. For the IG 1-A compressor design, there was only enough heat available to reduce the steam extraction to FWH-2, but not to eliminate it. The IG 1-B compressor did not have cooling water at a high enough temperature to integrate to FWH-2.

As discussed in Section 2, the steam that normally passes through FWH-2 is then sent through FWH-1 since there is still heat available in that stream. This cascading was also done with the hot cooling water stream so that it passed through FWH-2 and then was sent to FWH-1 in order to utilize the remaining heat available in the stream. Since the hot cooling water passes through two heat exchangers in this design, the pressure drop is greater which raises the power requirements of the cooling water pump. In order to cascade the hot cooling water back to FWH-

1, it is required to have two heat exchangers for FWH-1, one for the hot cooling water and one for the steam extraction. This two heat exchanger design is similar to what was done for the heat integration to FWH-1 above, and it was also designed to have the cooling water from the compressor exit FWH-1A at 100℉.

Since the Ramgen and the Inline 4 compressors provide enough heat to replace the steam extraction for FWH-2, only one heat exchanger is required for the hot cooling water to heat the feedwater in FWH-2. The hot cooling water exiting FWH-2 is then sent through FWH-1A in order to use the remaining heat available in the stream. FWH-2 usually has a stream from the SSR mix with the steam extraction between LP3 and LP4 before it passes through FWH-2. When FWH-2 is replaced for this thermal integration method, this stream from the SSR is mixed with the steam extraction for FWH-3 and sent through this FWH instead. Figure 5.3 below illustrates the design for the thermal integration to FWH-2 when the steam extraction to this FWH can be eliminated.

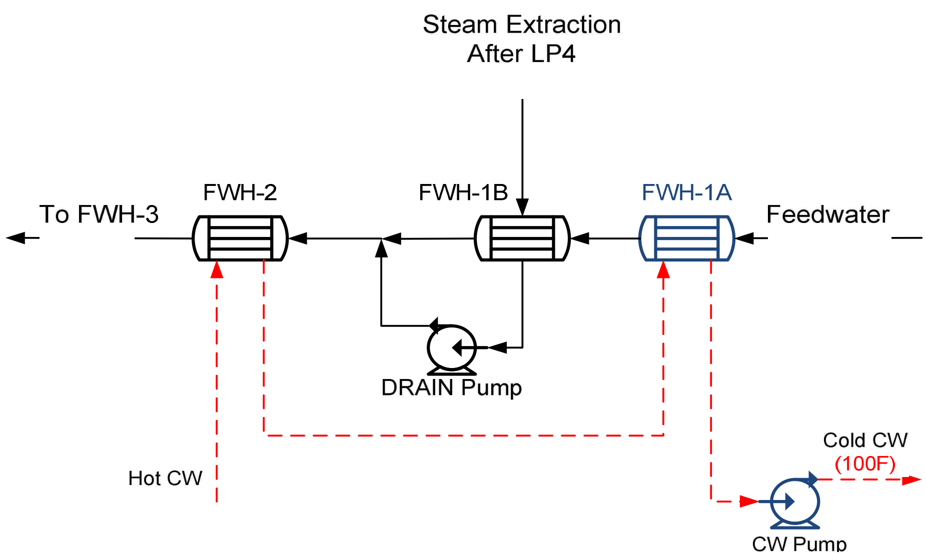

Figure 5.3 - Thermal Integration to FWH-2 for Ramgen and Inline 4.

Figure 5.4 shows the thermal integration to FWH-2 from the Ramgen and Inline 4 compressors within the complete steam turbine cycle.

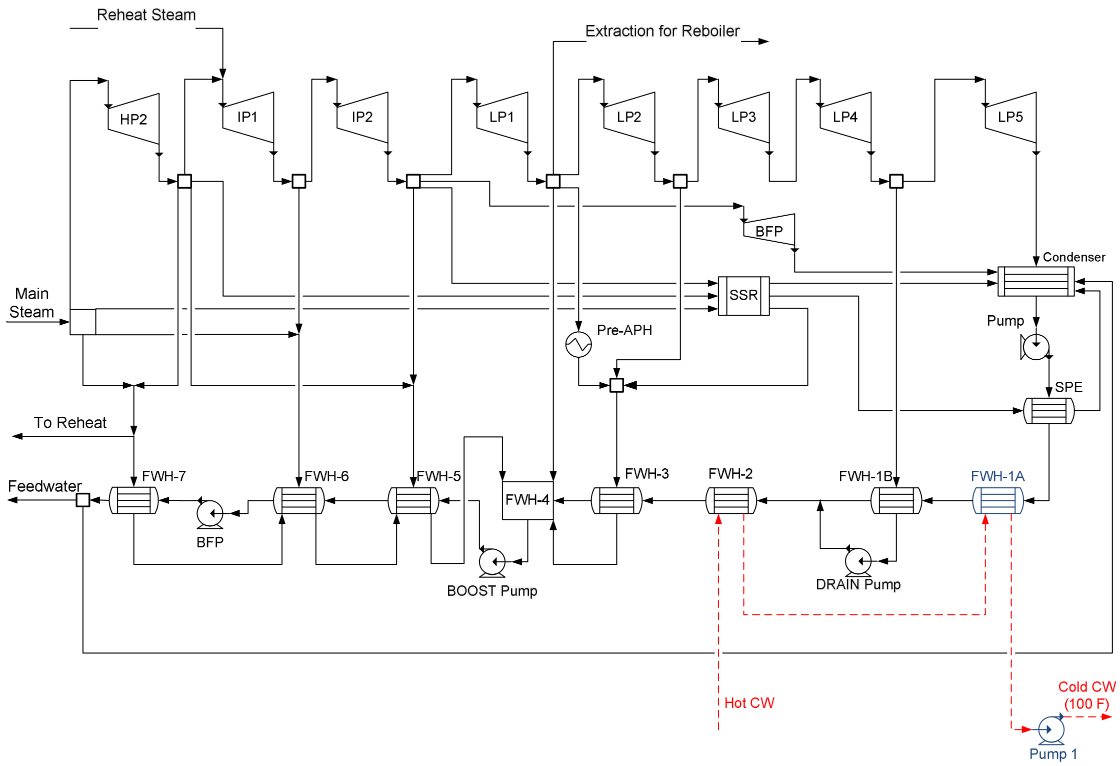

Figure 5.4 – Steam Turbine Cycle for Thermal Integration to FWH-2 for Ramgen and Inline 4.

Since the IG1-A compressor only provided enough heat to reduce the steam extraction to FWH-2, two heat exchangers are required for FWH-2, as it was for the heat integration to FWH-1. The first heat exchanger (FWH-2A) uses the hot cooling water from the compressor to heat the feedwater as much as possible. Once the hot cooling water passes through FWH-2A, it is then sent through FWH-1A in order to use the remaining heat available in the hot cooling water. The second heat exchanger (FWH-2B) heats the feedwater the remainder of the way to the designated outlet temperature with the steam extraction, but less steam is required since some of the heat duty of FWH-2 is offset by the hot cooling water. Figure 5.5 below illustrates the design for the thermal integration to FWH-2 from the IG1-A compressor. The affect of this thermal integration to FWH-2 on the plant's performance is discussed in the following sections for each compressor design.

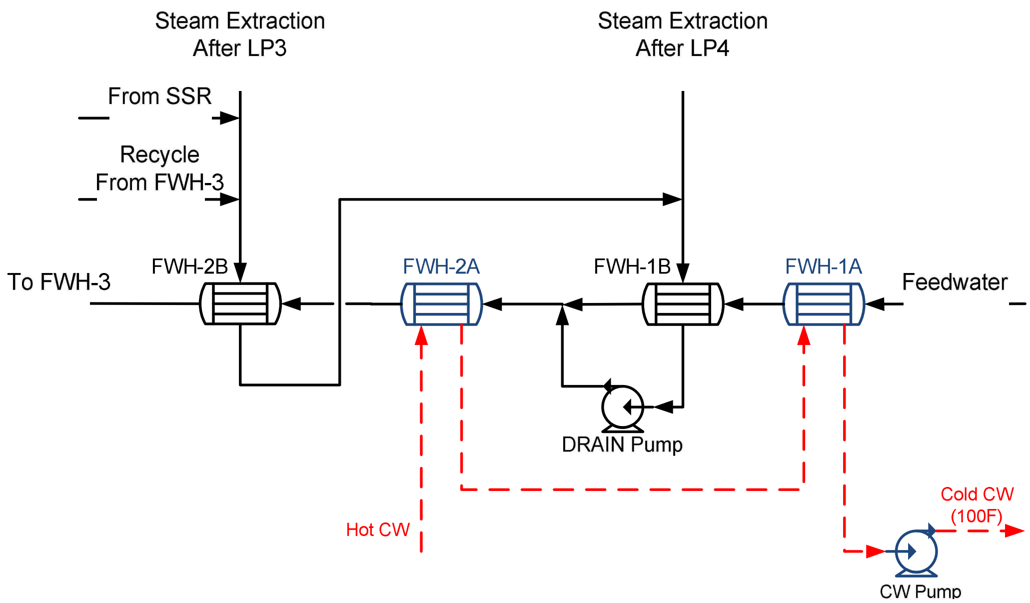

Figure 5.5 - Thermal Integration to FWH-2 for IG 1-A.

Figure 5.6 shows the heat integration to FWH-2 from the IG1-A compressor within the complete steam turbine cycle.

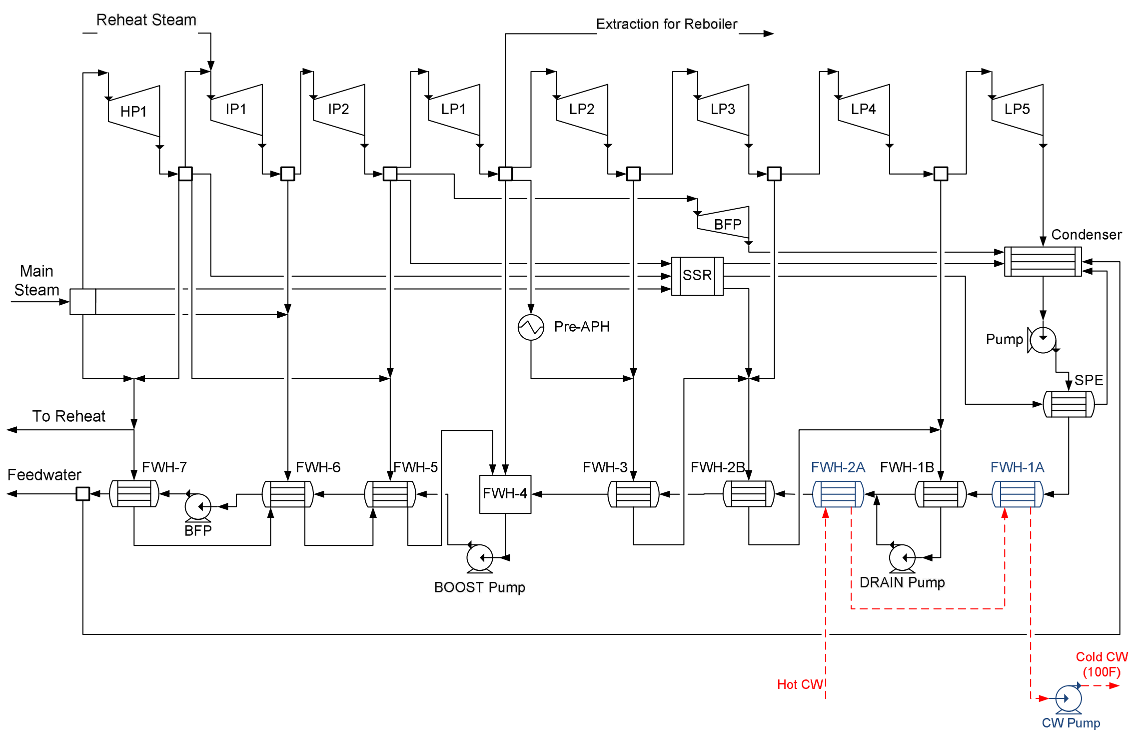

Figure 5.6 – Steam Turbine Cycle for Thermal Integration to FWH-2 for the IG 1-A.

The third thermal integration analyzed was to the third feedwater heater (FWH-3). The IG 1-B compressor did not have enough heat available to integrate to FWH-3. As with the thermal integration to FWH-2, the heat available from the Ramgen and Inline 4 compressor designs could replace FWH-3 and eliminate the need for a steam extraction for this FWH. This required only one heat exchanger for FWH-3 for the hot cooling water to pass through. Normally a stream from the pre-APH heater is mixed with the steam extraction for FWH-3, but since this FWH was replaced, this stream was sent to FWH-2 instead to help reduce the steam extraction required for that FWH.

The IG 1-A compressor only had enough heat available to provide some of the heat duty of FWH-3. The steam extraction could still be eliminated, but a second FWH (FWH-3B) was required for the pre-APH heat to provide the remaining heat duty of FWH-3. This requires the use of two heat exchangers for FWH-3, but still allows the steam extraction to be eliminated. The first FWH (FWH-3A) uses the hot cooling water to heat the feedwater, and the second FWH (FWH-3B) uses the heat available from the pre-APH to heat the feedwater up to the designated outlet temperature.

Also similar to the thermal integration to FWH-2, the hot cooling water exiting FWH-3, for both designs, is cascaded back through FWH-2A and FWH-1A, where any remaining heat is utilized. Since the hot cooling water travels through 3 heat exchangers in this design, the pressure drop is greater and requires greater cooling water pump power. The stream from the pre-APH that runs through FWH-3B is also cascaded down through FWH-2B and FWH-1B.

Figure 5.7 below illustrates the design for the thermal integration to FWH-3 from the Ramgen and Inline 4 compressor designs where FWH-3 can be replaced by the available heat from compression.

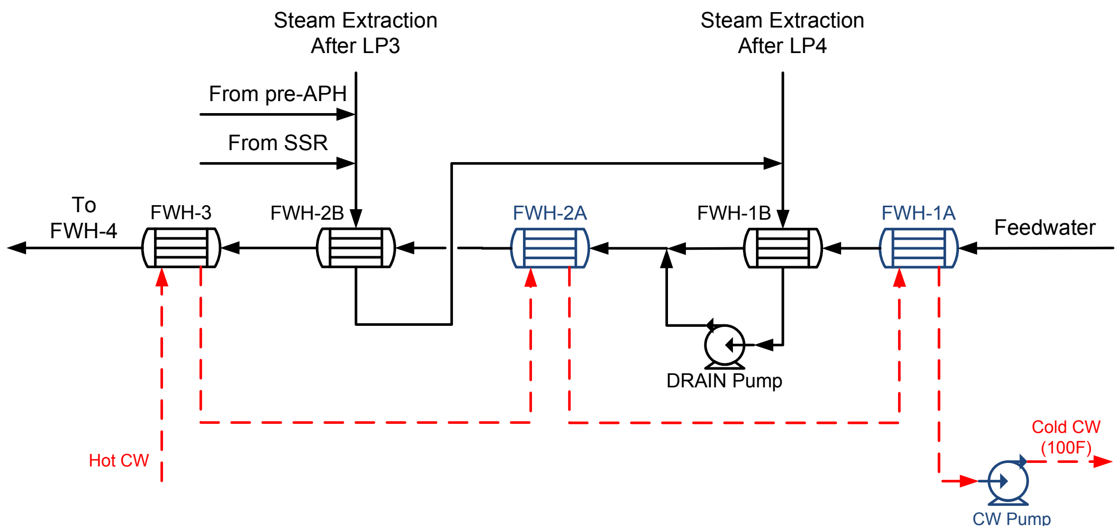

Figure 5.7 - Thermal Integration to FWH-3 for Ramgen and Inline 4.

Figure 5.8 illustrates the thermal integration to FWH-3 within the complete steam turbine cycle for the Ramgen and Inline 4 compressors.

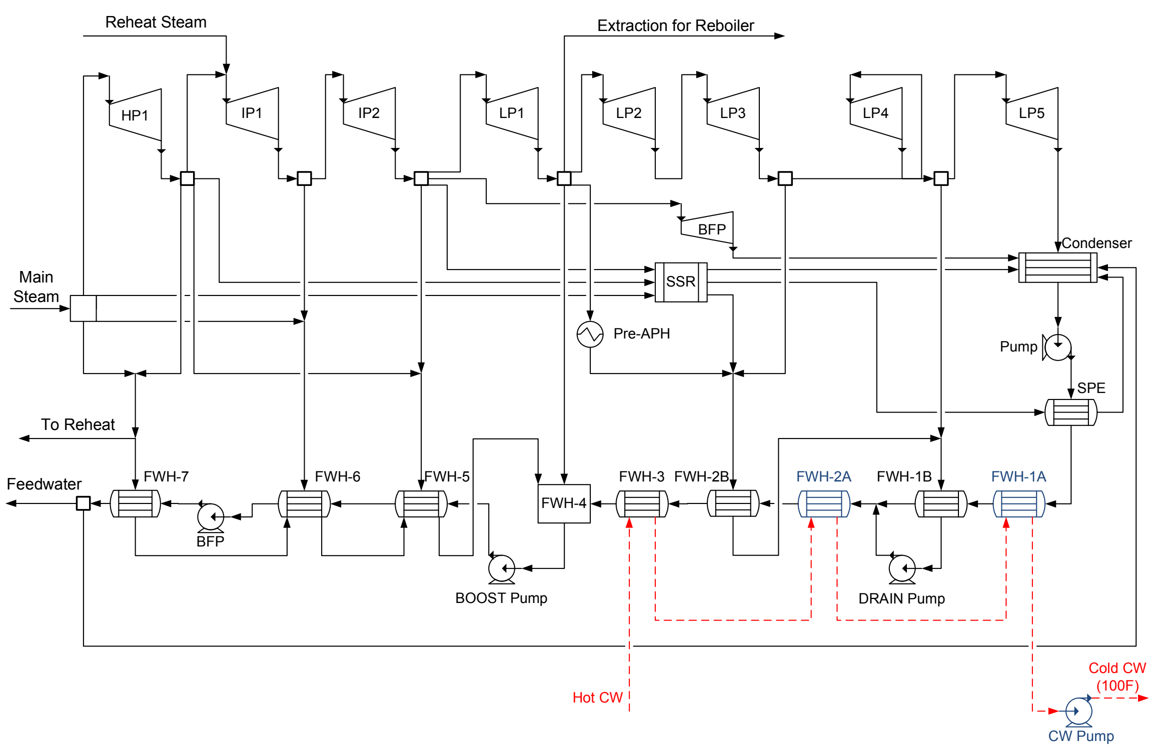

Figure 5.8 – Steam Turbine Cycle for Thermal Integration to FWH-3 for the Ramgen and Inline 4.

Figure 5.9 below illustrates the design for the thermal integration to FWH-3 from the IG 1-A compressor design where the pre-APH heat is still required for FWH-3.

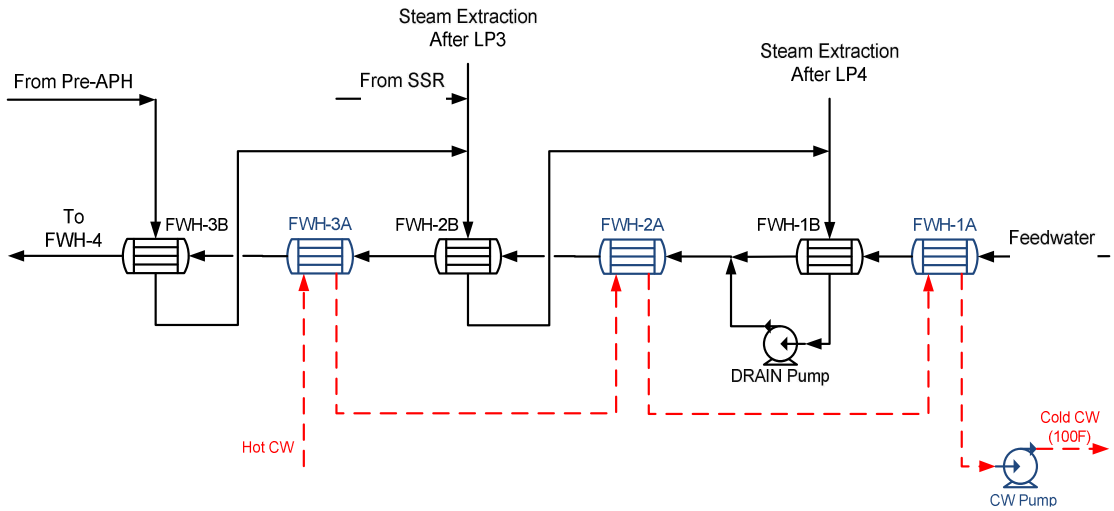

Figure 5.9 - Thermal Integration to FWH-3 for IG 1-A.

Figure 5.10 illustrates the thermal integration to FWH-3 within the complete turbine cycle for the IG 1-A compressor.

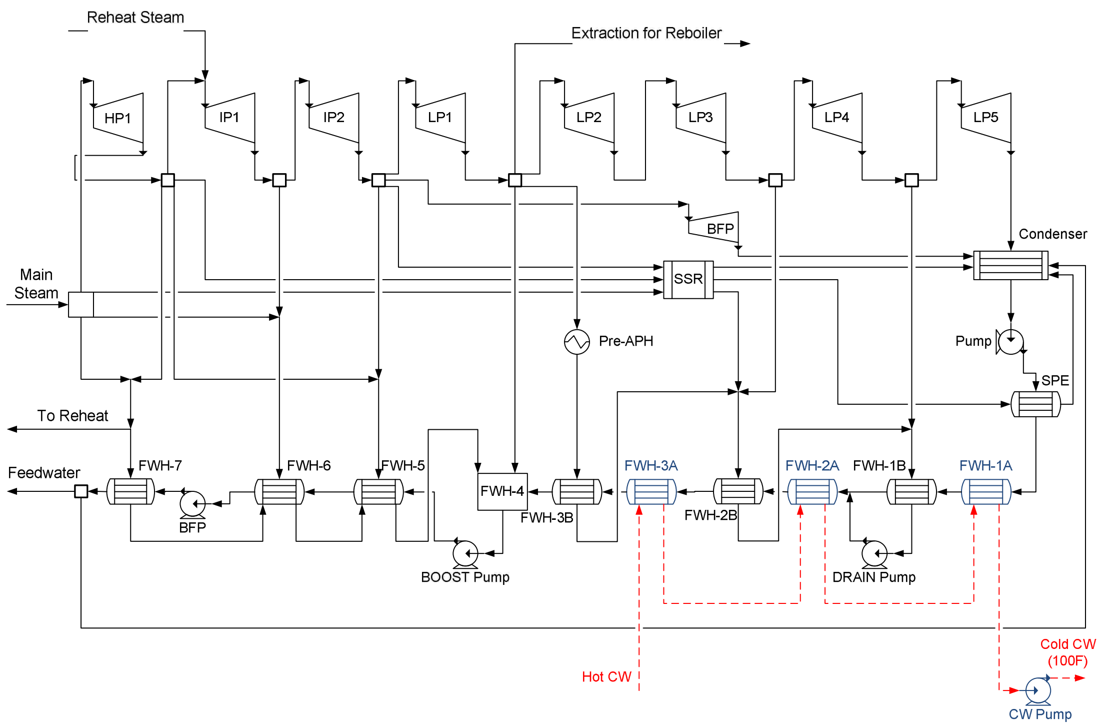

Figure 5.10 – Steam Turbine Cycle for Thermal Integration to FWH-3 for IG 1-A.

The effect of this thermal integration to FWH-3 on the plant's performance is discussed in the following sections for each compressor design.

5.2.1.1 Ramgen

The heat available from the post-cooler of the Ramgen compressor is 174.1 MBtu/hr, which is available in 459,225 lb/hr of 466.4℉ water. When this heat is integrated to FWH-1, the steam extraction to this FWH can be reduced from 178,947 to 9,325 lb/hr. When this heat is integrated to FWH-2, the steam extraction to FWH-2 can be eliminated, and the steam extraction required for FWH-1 is 140,575 lb/hr. When this heat is integrated to FWH-3, the steam extraction to FWH-3 can be eliminated, and the steam extractions required for FWH-1 and FWH-2 are 159,350 lb/hr and 86,305 lb/hr, respectively.

The improvement of the plant performance as a result of these FWH thermal integration designs is shown in Table 5.4 along with the plant performance when no thermal integration is used from the Ramgen compressor. A more detailed table of these results can be found in Appendix C.2.1.

Table 5.4 – Results for Thermal Integration to FWHs for Ramgen Compressor.

	No Thermal Integration	FWH-1	FWH-2	FWH-3
Net Power (kW)	447,980	451,870	454,202	455,377
∆ Net Power (kW)	--	3,890	6,222	7,397
Unit HR (Btu/kWh)	12,191	12,087	12,024	11,993
∆ HR (Btu/kWh)	--	-104	-167	-198
% HR Improvement	--	0.86%	1.37%	1.62%
Unit Efficiency (%)	27.99%	28.23%	28.38%	28.45%
∆ Unit Efficiency (%)	--	0.24%	0.39%	0.46%

It can be seen that integrating to FWH-3 is more beneficial than integrating to FWH-2, which is more beneficial than integrating to FWH-1. This is because reducing the steam extraction for FWH-3 allows more steam to travel through three turbines (LP3 through LP5) while reducing the extraction for FWH-1 only allows more steam to travel through one turbine (LP5). Integrating to FWH-3 allows the plant to recover 7.40 MW of the 138.4 MW of power lost due to the capture system with this compressor design. This equates to a recovery of 5.3% of the power that is lost. This thermal integration to FWH-3 allows a 1.62% heat rate improvement for the plant and an increase of 0.46% in the unit efficiency.

5.2.1.2 Inline 4

The heat available from the inter-cooler of the Inline compressor is 168.4 MBtu/hr, which is available in 492,072 lb/hr of 433.2°F water. When this heat is integrated to FWH-1, the steam extraction to this FWH can be reduced from 178,947 to 14,875 lb/hr. When this heat is integrated to FWH-2, the steam extraction to FWH-2 can be eliminated, and the steam extraction required for FWH-1 is 144,525 lb/hr. When this heat is integrated to FWH-3, the steam extraction to FWH-3 can be eliminated, and the steam extractions required for FWH-1 and FWH-2 is 161,800 lb/hr and 89,300 lb/hr, respectively.

The improvement of the plant performance as a result of these FWH thermal integration designs is shown in Table 5.5 along with the plant performance when no thermal integration is used from the Inline compressor. A more detailed table of these results can be found in Appendix C.2.2.

Table 5.5 – Results for Thermal Integration to FWHs for Inline Compressor.

	No Thermal Integration	FWH-1	FWH-2	FWH-3
Net Power (kW)	449,862	453,625	455,168	457,086
Δ Net Power (kW)	--	3,763	5,306	7,224
Unit HR (Btu/kWh)	12,140	12,040	11,999	11,949
Δ HR (Btu/kWh)	--	-100	-141	-191
% HR Improvement	--	0.83%	1.17%	1.58%
Unit Efficiency (%)	28.10%	28.34%	28.44%	28.56%
Δ Unit Efficiency (%)	--	0.24%	0.33%	0.45%

As with the Ramgen compressor, integrating to FWH-3 is more beneficial than integrating to FWH-2, which is more beneficial than integrating to FWH-1 for the Inline compressor design. Integrating to FWH-3 allows the plant to recover 7.22 of the 137.9 MW of power lost due to the capture system with this compressor design. This equates to a recovery of 5.2% of the power that is lost. This thermal integration to FWH-3 allows a 1.58% heat rate improvement for the plant and an increase of 0.45% in the unit efficiency.

5.2.1.3 IG1-A

The stream with the most heat available from this IG compressor design is the post-cooler, which has 112.9 MBtu/hr available in 572,086 lb/hr of 296.0℉ water. When this heat is integrated to FWH-1, the steam extraction to this FWH can be reduced from 178,947 to 69,025 lb/hr. When this heat is integrated to FWH-2, the steam extraction to FWH-2 can be reduced to 39,025 lb/hr, and the steam extraction required for FWH-1 is 148,750 lb/hr. When this heat is integrated to FWH-3, the steam extraction to FWH-3 can be eliminated, but a second heat exchanger (FWH-3B) must still be used with the heat from the pre-APH that is usually fed to FWH-3 in order to reach the necessary outlet temperature from FWH-3. For this thermal integration to FWH-3, the steam extraction required for FWH-1 and FWH-2 is 171,275 lb/hr and 99,875 lb/hr, respectively.

The improvement of the plant performance as a result of these three different thermal integration designs is shown below in Table 5.6 along with the plant performance when no thermal integration is used from the IG 1-A compressor. A more detailed table of these results can be found in Appendix C.2.3.

Table 5.6 – Results for Thermal Integration to FWHs for IG1-A Compressor.

	No Thermal Integration	FWH-1	FWH-2	FWH-3
Net Power (kW)	451,687	454,209	455,342	456,781
∆ Net Power (kW)	--	2,522	3,655	5,094
Unit HR (Btu/kWh)	12,091	11,998	11,994	11,957
∆ HR (Btu/kWh)	--	-67	-97	-134
% HR Improvement	--	0.56%	0.80%	1.12%
Unit Efficiency (%)	28.22%	28.38%	28.45%	28.54%
∆ Unit Efficiency (%)	--	0.16%	0.23%	0.32%

For this IG compressor design, integrating to FWH-3 is more beneficial than integrating to the other FWHs. Integrating to FWH-3 allows the plant to recover 5.09 of the 134.1 MW of power lost due to the capture system with this compressor design. This equates to a recovery of 3.8% of the power that is lost. This thermal integration to FWH-3 allows a 1.12% heat rate improvement for the plant and an increase of 0.32% in the unit efficiency.

5.2.1.4 IG1-B

The stream with the most heat available from this IG compressor design is the third inter-cooler (IC-3), which has 71.5 MBtu/hr available in 873,125 lb/hr of 182.0℉ water. This heat can only be integrated to FWH-1 due to the lower temperature of this hot cooling water stream compared to the other compressor designs. When the heat is integrated to FWH-1, the steam extraction to this FWH can be reduced from 178,947 to 109,325 lb/hr.

The improvement of the plant performance as a result of this FWH thermal integration design is shown below in Table 5.7 along with the plant performance when no thermal integration is used from the IG 1-B compressor. A more detailed table of these results can be found in Appendix C.2.3.

Table 5.7 – Results for Thermal Integration to FWHs for IG1-B Compressor.

	No Thermal Integration	FWH-1
Net Power (kW)	455,477	457,073
∆ Net Power (kW)	--	1,597
Unit HR (Btu/kWh)	11,991	11,949
∆ HR (Btu/kWh)	--	-42
% HR Improvement	--	0.35%
Unit Efficiency (%)	28.46%	28.55%
∆ Unit Efficiency (%)	--	0.10%

For this IG compressor design, integrating to FWH-1 is the only option. Integrating to FWH-1 for allows the plant to recover 1.60 MW of the 132.3 MW of power lost due to the capture system with this compressor design. This equates to a recovery of 1.2% of the power that is lost by the plant for the operation of the carbon capture system with this IG compressor. This thermal integration to FWH-1 allows a 0.35% heat rate improvement for the plant and an increase of 0.10% in the unit efficiency.

5.2.1.5 Comparison

Integrating heat to FWH-3 is more beneficial to the plant's performance than integrating to FWH-2, which is more beneficial than integrating to FWH-1. This is due to the fact that reducing or eliminating the extraction for FWH-3 increases the steam flow through three turbines, LP3, LP4, and LP5. Integrating the heat to FWH-2 and reducing or eliminating the extraction for this FWH only increases the steam flow through turbines LP4 and LP5. Integrating the heat to FWH-1 and reducing the extraction for this FWH only increases the steam flow through turbine LP5. One disadvantage of integrating to FWH-3, over FWH-2 or FWH-1, is that this requires more heat exchangers since the excess heat from the hot cooling water is cascaded through the lower FWHs leading to the need for two heat exchangers for each of the previous FWHs. Another disadvantage of integrating the heat to FWH-3 is that this causes the largest pressure drop in the cooling water and therefore requires the largest power for the cooling water pump.

Figures 5.11 through 5.15 illustrate the plant improvement for each compressor design when thermal integration to the FWHs is considered. These figures show the results for integration to the feedwater heater providing the greatest thermal performance for each compressor. For the Ramgen, Inline, and IG 1-A compressors, these results are for integrating to FWH-3. For the IG 1-B compressor, the results are for integrating to FWH-1.

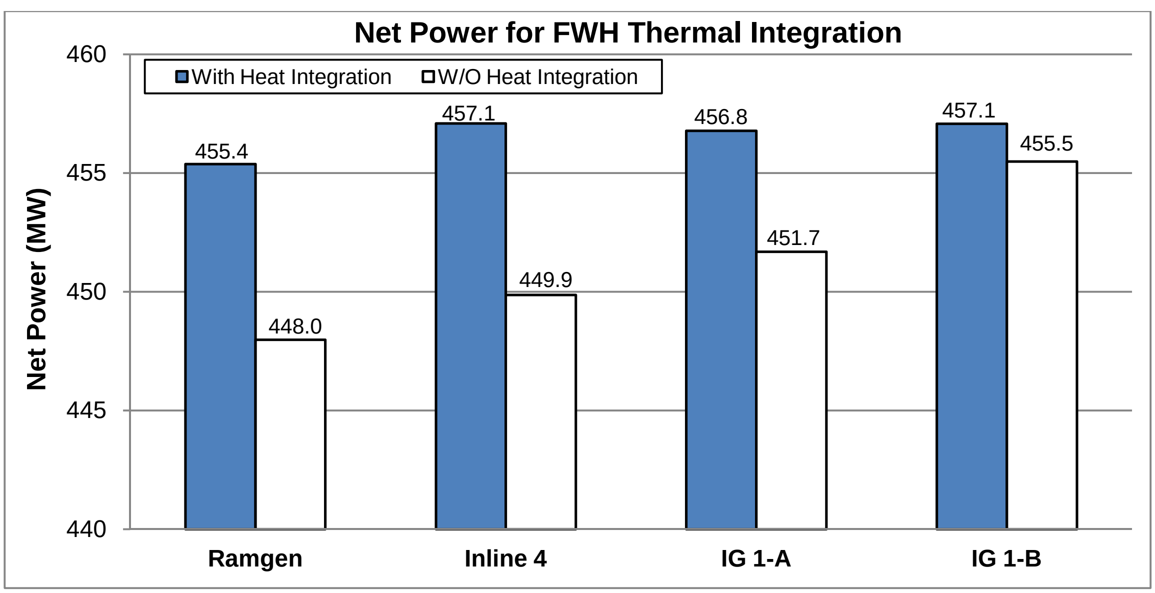

Figure 5.11 - Net Power Results for Thermal Integration to the Best FWHs.

Figure 5.12 illustrates the change in net power when thermal integration to the FWHs is considered, and this chart uses the Ramgen with no thermal integration as the base case.

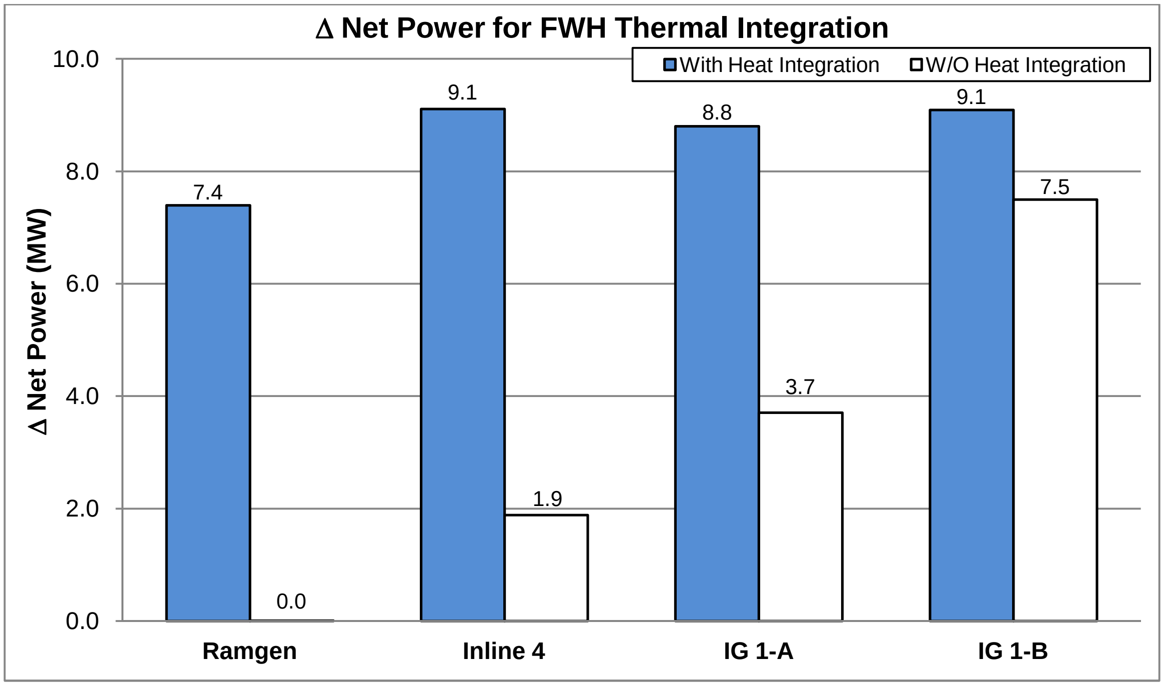

Figure 5.12 - Δ Net Power for Thermal Integration to the Best FWHs.

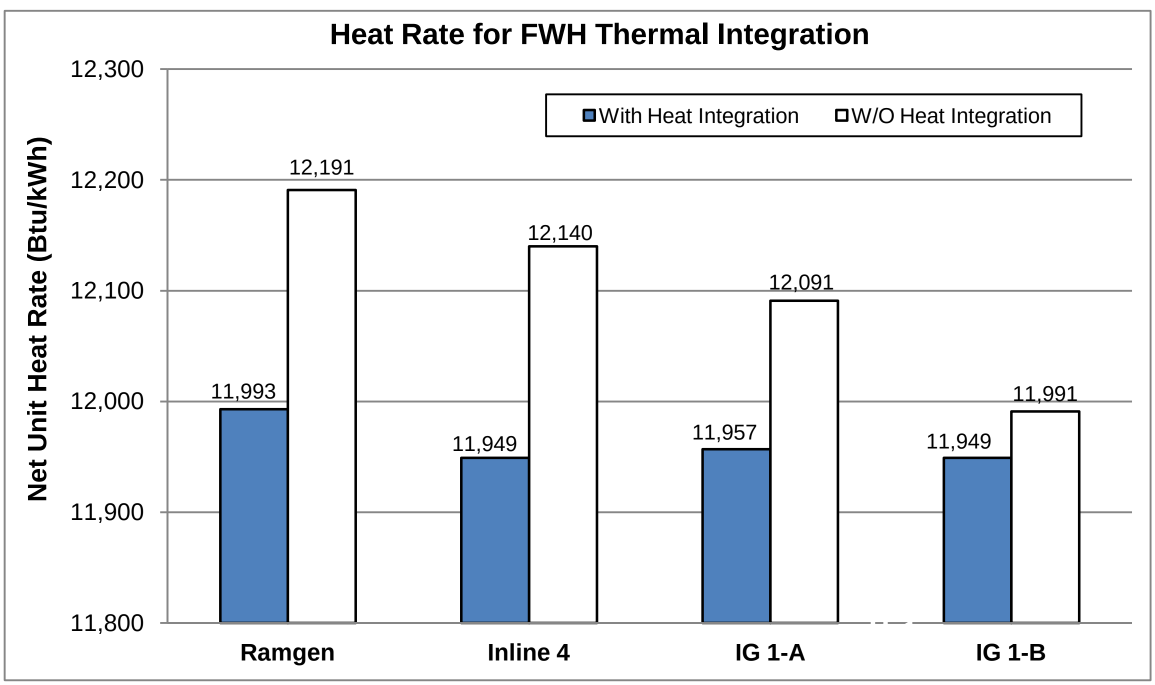

Figure 5.13 - Heat Rate Results for Thermal Integration to Best FWHs.

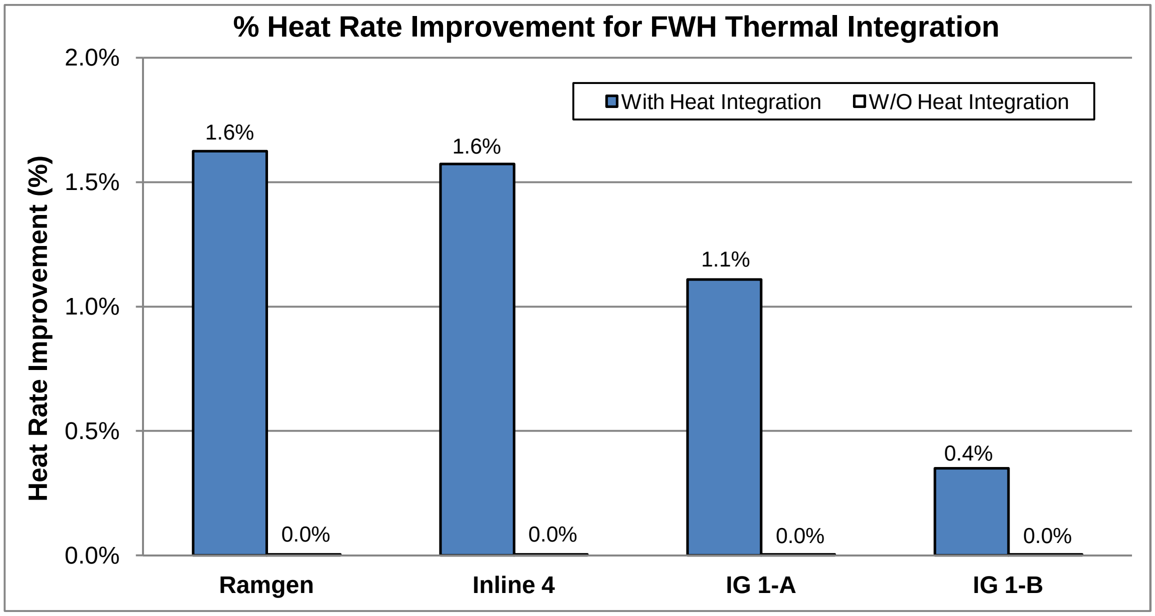

Figure 5.14 - Heat Rate Improvement for Thermal Integration to Best FWHs.

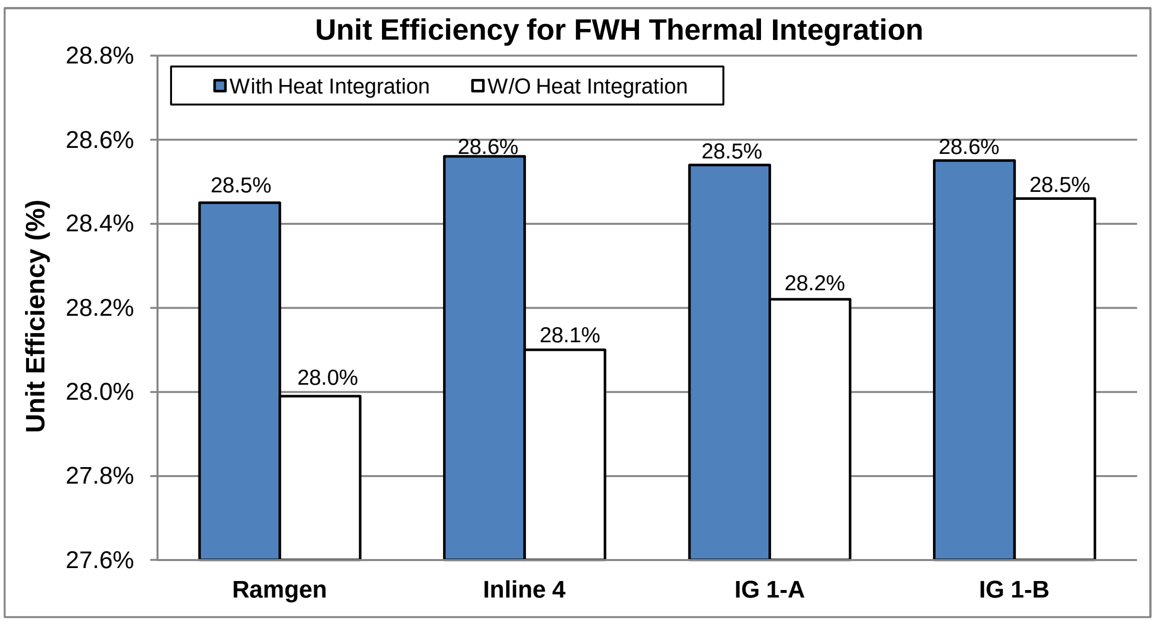

Figure 5.15 - Unit Efficiency for Thermal Integration to FWHs.

These figures show that the Inline 4 and IG 1-B compressor designs have similar plant performance when the heat is integrated to the FWHs, and they have the highest net power and unit efficiency and the lowest net unit heat rate of the compressor designs. It can also be seen that when the best option for FWH thermal integration is considered for each compressor, the difference in the plant performance between the four compressor designs is fairly small.

5.2.2 Stripper Reboiler

The second option for thermal integration that was analyzed was to use the available heat for the reboiler in the CO_2 separation stripper. This reboiler requires a large amount of heat, and this would normally be provided from steam extraction between turbines LP1 and LP2. Extracting steam at this point reduces the steam flow through turbines LP2, LP3, LP4, and LP5. As it was stated before, this steam extraction for the reboiler leads to a loss of about 20% of the plant's total power output. The hot cooling water streams from the IG 1-B compressor coolers are not at a high enough temperature to use the heat for the stripper reboiler. Therefore, the Ramgen, Inline 4 and IG 1-A compressor designs were analyzed for thermal integration to the stripper reboiler.

None of these three compressor designs has enough heat available to provide all of the heat necessary for the reboiler and eliminate the need for the steam extraction. In fact, these streams are only able to provide between 1.9 and 7.5% of the total heat duty of the reboiler. Despite this, integrating the compressor heat to the reboiler provides a significant improvement to the plant's performance since reducing the steam extraction for the reboiler increases the steam flow through turbines LP2, LP3, LP4, and LP5. This increases the power output of four turbine stages as opposed to the FWH thermal integration which only increases the power output of one to three turbine stages.

Since a steam extraction is still required, the design of the reboiler must contain two heat exchangers. The first reboiler heat exchanger (REB HX 1) heats the stripper bottoms as much as possible with the heat available from the compressor cycle. A pressure drop of 15 psia for the hot cooling water was assumed across this heat exchanger. The second reboiler heat exchanger (REB HX 2) provides the remaining heat duty with steam extracted from the steam turbine cycle between LP1 and LP2. Without thermal integration, the heat duty of this CO_2 separation reboiler is 1,051.4 MBtu/hr, and the total reboiler duty for the CO_2 separation and the NH_3 abatement is 1,401.4 M Btu/hr. This total reboiler duty requires a steam extraction of 1,372,207 lb/hr

Figure 5.16 illustrates the design for integrating the compressor heat to the stripper reboiler. The thermal integration to the stripper reboiler is analyzed in the following sections.

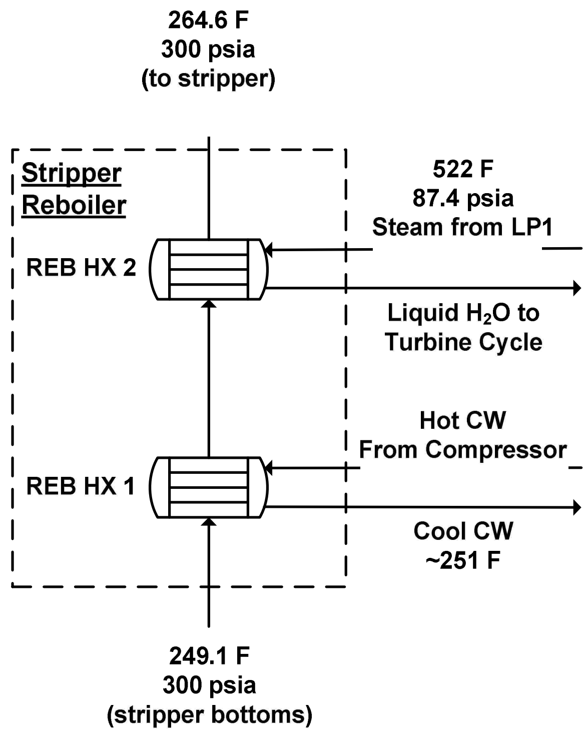

Figure 5.16 - Thermal Integration To Stripper Reboiler from Compressor.

Since the stripper bottoms are entering the reboiler at 249°F, the hot cooling water can only be cooled to about 251°F, which means that only a port ion of the available heat can be used in the stripper reboiler. For this method of thermal integration, this 251°F cooling water would need to then pass through a cooling tower or other heat sink to bring the temperature down to 100°F so that it can be used again as cooling water. Other uses for this remaining heat are discussed in Section 5.2.4

5.2.2.1 Ramgen

The heat available from the post-cooler of the Ramgen compressor is 174.1 MBtu/hr, which is available in 459,225 lb/hr of 466.4°F water. When this heat is integrated to the stripper reboiler, 104.9 MBtu/hr is transferred to the lean ammonia stream, which is 60.2% of the available heat that is used. The remaining heat duty of this reboiler is reduced from 1,051.4 to 946.5 MBtu/hr, and with the abatement reboiler heat duty of 350 MBtu/hr, the total reboiler heat duty is 1,296.5

MBtu/hr. This reduces the required steam extraction from between LP1 and LP2 for the two reboilers by 1,269,536 lb/hr, which is a 7.5% reduction in the steam extraction required for the case without thermal integration.

The effect that thermal integration to the stripper reboiler has on the plant's performance is shown in Table 5.8, along with the plant's performance with no thermal integration from the Ramgen compressor. Also included in this table are the results of the thermal integration to FWH-3 from section 5.2.1.1 since this was the best case for the FWH integration with this compressor design. A more detailed table of these results can be found in Appendix C.2.1.

Table 5.8 - Results for Thermal Integration to the Stripper Reboiler for Ramgen Compressor.

	No Thermal Integration	**Stripper Reboiler**	**FWH-3**
Net Power (kW)	447,980	456,558	455,377
Δ Net Power (kW)	--	8,578	7,397
Unit HR (Btu/kWh)	12,191	11,962	11,993
Δ HR (Btu/kWh)	--	-229	-198
% HR Improvement	--	1.88%	1.62%
Unit Efficiency (%)	27.99%	28.52%	28.45%
Δ Unit Efficiency (%)	--	0.54%	0.46%

This table shows that for the Ramgen compressor design, integrating the compressor heat to the stripper reboiler improves the plant performance more than integrating the heat to the FWHs. Integrating to the stripper reboiler results in a recovery of 8.6 of the 138.4 MW of power lost due to the operation of the capture system with this compressor design. This equates to a recovery of 6.2% of the power that is lost. When integrating to FWH-3, only 5.3% of the power lost is recovered. This thermal integration to the stripper reboiler results in a 1.88% heat rate improvement for the plant and an increase of 0.54% in the unit efficiency.

5.2.2.2 Inline 4

The heat available from the Inline 4 compressor inter-cooler is 168.4 MBtu/hr, which is available in 492,072 lb/hr of 433.2℉ water. When this heat is integrated to the stripper reboiler, 94.2 MBtu/hr is transferred to the lean ammonia stream, which is 55.9% of the available heat that is used. The remaining heat duty of this reboiler is reduced from 1,051.4 to 957.2 MBtu/hr, and with the abatement reboiler heat duty of 350 MBtu/hr, the total reboiler heat duty is 1,307.2 MBtu/hr. This reduces the required steam extraction from between LP1 and LP2 for the two reboilers by 92,199 lb/hr, which is a 6.7% reduction in the steam extraction required for the case without thermal integration.

The effect that thermal integration to the stripper reboiler has on the plant's performance is shown in Table 5.9, along with the plant's performance with no thermal integration from the Inline 4 compressor. Also included in this table are the results of the heat integration to FWH-3 from section 5.2.1.2. A more detailed table of these results can be found in Appendix C.2.2.

Table 5.9 - Results for Heat Integration to the Stripper Reboiler for Inline 4 Compressor.

	No Thermal Integration	Stripper Reboiler	FWH-3
Net Power (kW)	449,862	457,564	457,086
Δ Net Power (kW)	--	7,702	7,224
Unit HR (Btu/kWh)	12,140	11,936	11,949
Δ HR (Btu/kWh)	--	-204	-191
% HR Improvement	--	1.68%	1.58%
Unit Efficiency (%)	28.10%	28.59%	28.56%
Δ Unit Efficiency (%)	--	0.49%	0.45%

Like the Ramgen compressor, integrating the compressor heat to the stripper reboiler improves the plant performance more than integrating the heat to the FWHs for the Inline 4 compressor design, but only slightly. Integrating to the stripper reboiler results in a recovery of 7.7 of the 137.9 MW of power lost due to the operation of the capture system with this compressor design. This equates to a recovery of 5.6% of the power that is lost. When integrating the heat to FWH-3 with this compressor design, 5.2% of the power lost is recovered. This thermal integration to the stripper reboiler results in a 1.68% heat rate improvement for the plant and an increase of 0.49% in the unit efficiency.

5.2.2.3 IG1-A

The heat available from the post-cooler of this IG compressor is 112.9 MBtu/hr, which is available in 572,086 lb/hr of 296.0℉ water. When this heat is integrated to the stripper reboiler, 26.5 MBtu/hr is transferred to the lean ammonia stream, which is 16.4% of the available heat that is used. The remaining heat duty of this reboiler is reduced from 1,051.4 to 1,024.9 MBtu/hr, and with the abatement reboiler heat duty of 350 MBtu/hr, the total reboiler heat duty is 1,374.9 MBtu/hr. This reduces the required steam extraction from between LP1 and LP2 for the two reboilers by 25,905 lb/hr, which is a 1.9% reduction in the steam extraction required for the case without thermal integration.

The effect that thermal integration to the stripper reboiler has on the plant's performance is shown in Table 5.10, along with the plant's performance with no thermal integration from this IG compressor. Also included in this table are the results of the thermal integration to FWH-3 from section 5.2.1.3. A more detailed table of these results can be found in Appendix C.2.3.

Table 5.10 - Results for Thermal Integration to the Stripper Reboiler for IG1-A Compressor.

	No Thermal Integration	Stripper Reboiler	FWH-3
Net Power (kW)	451,687	453,844	456,781
Δ Net Power (kW)	--	2,157	5,094
Unit HR (Btu/kWh)	12,091	12,034	11,957
Δ HR (Btu/kWh)	--	-57	-134
% HR Improvement	--	0.48%	1.12%
Unit Efficiency (%)	28.22%	28.35%	28.54%
Δ Unit Efficiency (%)	--	0.13%	0.32%

Unlike the Ramgen and Inline compressors, integrating the compressor heat to the stripper reboiler does not improve the plant performance as much as integrating the heat to FWH-3 for this IG compressor design. Integrating to the stripper reboiler results in a recovery of 2.16 of the 134.1 MW of power lost due to the operation of the capture system with this compressor design. This equates to a recovery of 1.6% of the power that is lost. When integrating to FWH-3, 3.8% of the power lost is recovered. This thermal integration to the stripper reboiler results in a 0.48% heat rate improvement for the plant and an increase of 0.13% in the unit efficiency.

5.2.2.4 Comparison

For the Ramgen, Inline 4, and IG 1-A compressors, the improvement of the plant performance is better when the heat is integrated to the stripper reboiler as opposed to the FWHs. This is despite the fact that not all of the heat available is being utilized. Thermal integration to the stripper reboiler provides a greater improvement to the plant performance because it increases the steam flow through four turbine stages (LP2 through LP5) where integration to FWH-3 only increases the steam flow through three turbine stages (LP3 through LP5).

An added advantage of integrating the heat to the stripper reboiler over the FWHs is that it only requires the addition of one additional heat exchanger, as opposed to integrating to FWH-3 where two or three additional heat exchangers are required. The disadvantage of integrating to the stripper reboiler over the FWHs is that all of the available heat is not used, and as a result there is a need for a heat sink for the remaining heat in the hot cooling water stream exiting the reboiler.

Figures 5.17 through 5.21 below illustrate the plant performance for each compressor design when thermal integration to the stripper reboiler is analyzed. Figure 5.18 illustrates the change in net power when thermal integration to the stripper reboiler is considered, and this chart uses the Ramgen with no thermal integration as the base case.

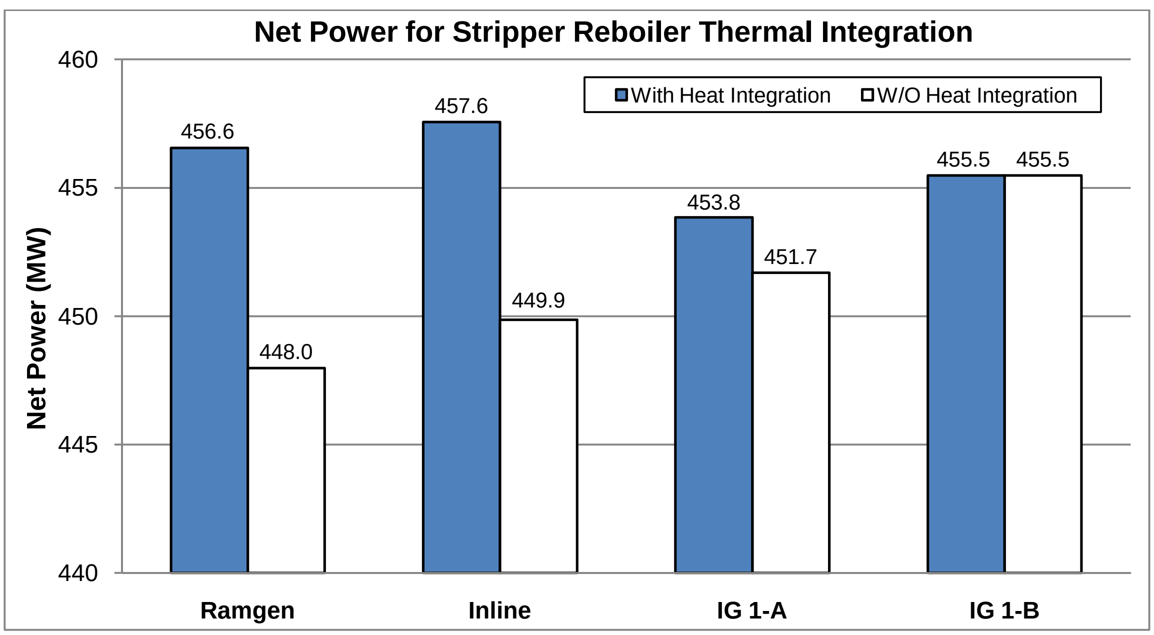

Figure 5.17 - Net Power Results for Thermal Integration to the Stripper Reboiler.

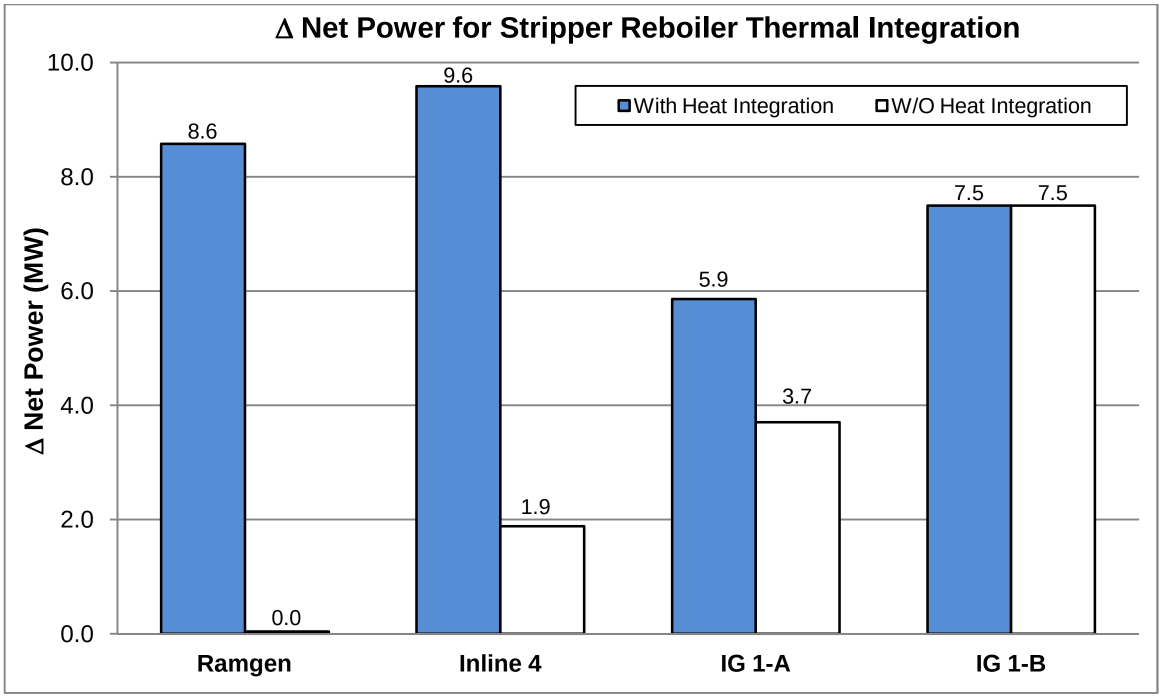

Figure 5.18 - Δ Net Power Results for Thermal Integration to the Stripper Reboiler.

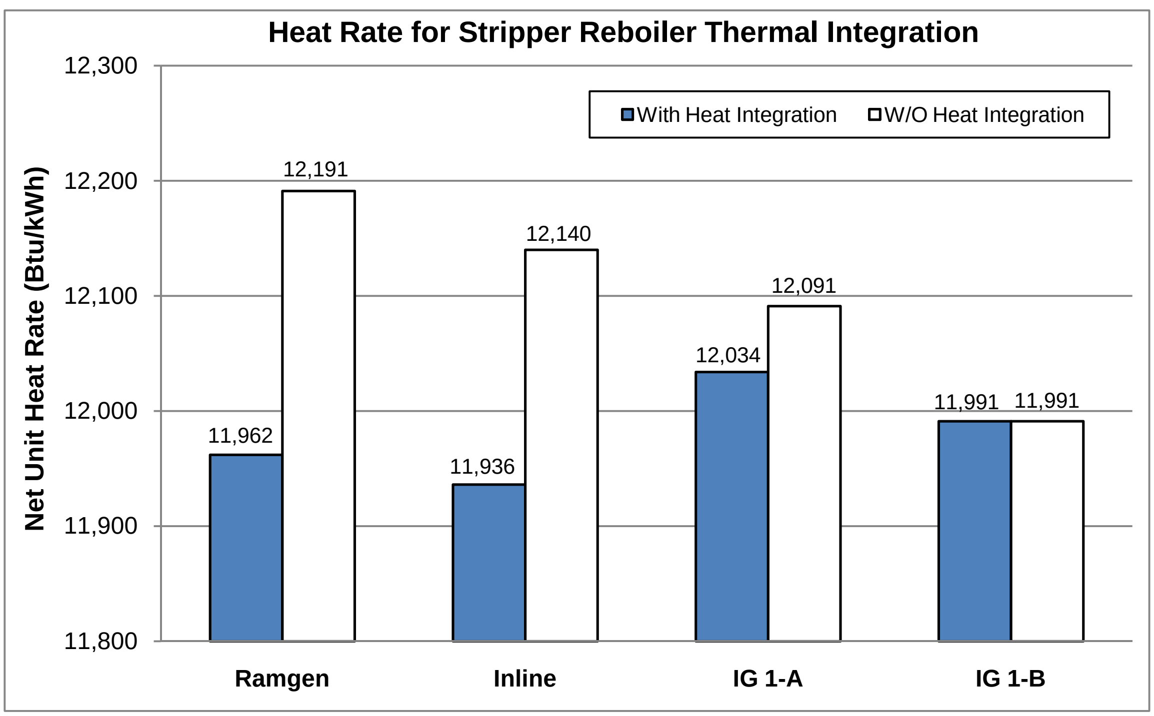

Figure 5.19 - Heat Rate Results for Thermal Integration to the Stripper Reboiler.

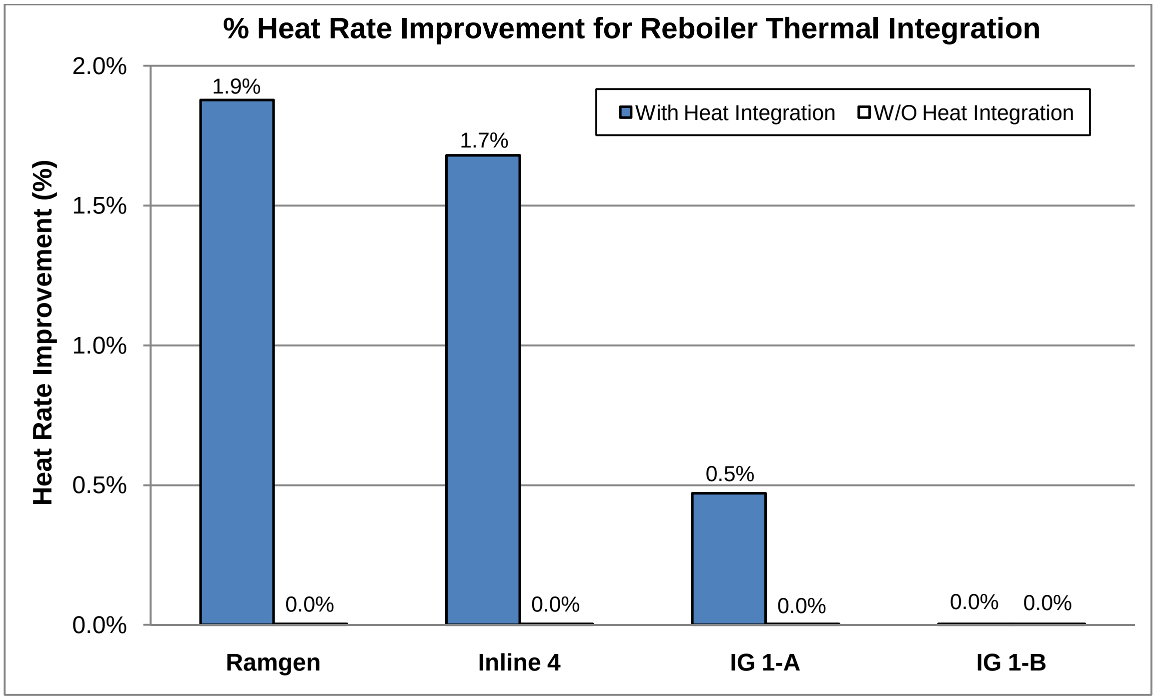

Figure 5.20 - Heat Rate Improvement for Thermal Integration to the Stripper Reboiler.

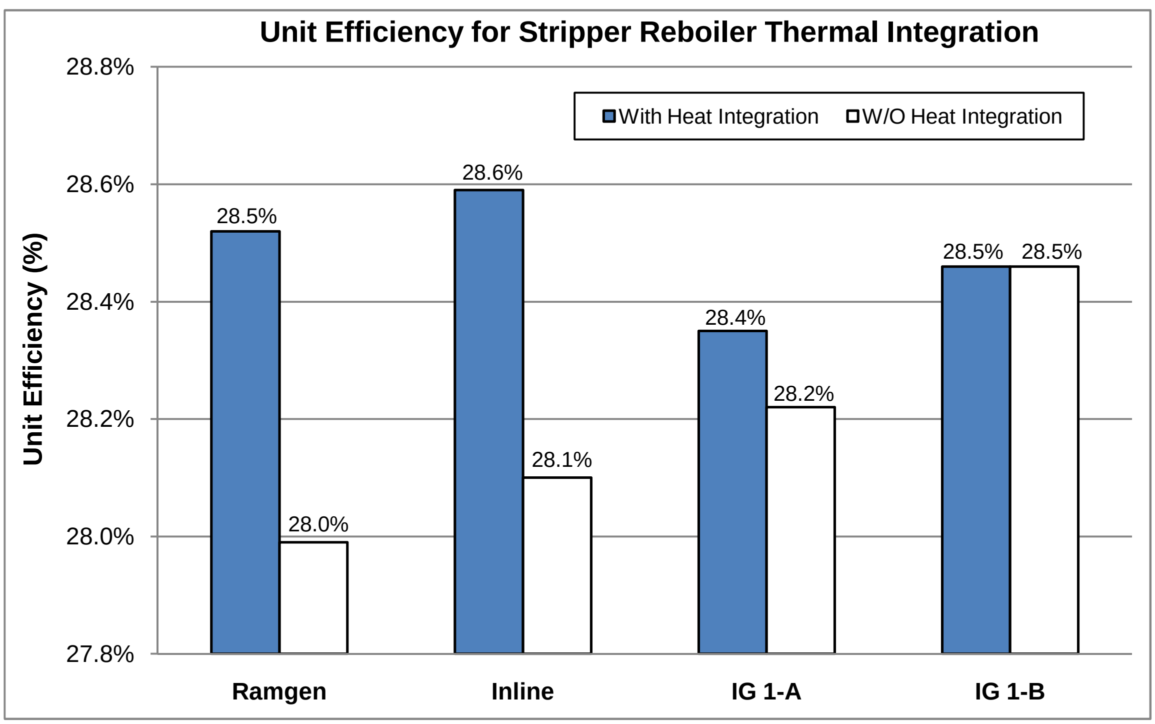

Figure 5.21 - Unit Efficiency for Thermal Integration to the Stripper Reboiler.

5.2.3 Coal Drying

The third method of thermal integration was to use the available heat to dry the coal before it enters the boiler in the power plant. The PRB coal analyzed in this thesis has a moisture content of 28.09%. When the moisture content of the coal is reduced, the boiler becomes more efficient and less coal is needed to generate the same amount of power. This is due to less energy being expended to evaporate the moisture in the coal, and therefore more energy being transferred to the boiler feedwater for steam generation.

Coal is made up of carbon, hydrogen, oxygen, nitrogren, and sulfur. When coal is burned in a boiler, the carbon reacts with the oxygen in air in order to generate heat to convert the boiler feedwater to steam. When the coal is dried, the levels of hydrogen and oxygen are reduced, which leads to a coal with a higher concentration of carbon.

For the analysis in this thesis, the boiler and steam turbine cycle throttle conditions were held constant. This means that the energy transferred from the boiler to the boiler feedwater remained constant at 4,776 MBtu/hr ($\dot{Q}_{steam}$). In order to maintain a balance of energy entering the boiler and energy exiting the boiler, the Btu's of energy entering the boiler in the coal ($\dot{Q}_{coal}$) also had to remain constant. The equation for $\dot{Q}_{coal}$ is below in Equation 5.1.

$$\dot{Q}_{coal} = \dot{m}_{coal} * HHV_{coal} \tag{5.1}$$

When the moisture content of coal is reduced, the higher heating value (HHV) of the coal increases as a result of the higher concentration of carbon in the fuel. As mentioned in Section 2, the HHV is a measurement of the heat released from combustion and is reported in Btu per pound of coal. For the original 28.09% moisture in the PRB coal, the HHV is 8,426 Btu/lb, and when the coal is dried to 15% moisture, the HHV is 9,957 Btu/lb. With an increase in HHV, the heat released per pound of coal increases, and therefore the flowrate of the coal can decrease while maintaining the same energy input into the boiler. Since less coal carbon has to be burned in the boiler to evaporate the moisture in the coal, this also reduces the flowrate of the coal. The reduction in federate of carbon to the boiler results in a reduction of CO_2 formation.

Another difference between burning wet coal and burning dried coal is the amount of flue gas that is produced. When the coal is dried and less coal is burned, less flue gas is generated. Since less flue gas is generated and less CO_2 is formed, the power and heat requirements of the ammonia capture system and the compression cycle are reduced. When maintaining a 90% CO_2 capture rate with the chilled ammonia system, the amount of CO_2 that is not captured and sent to the stack with the flue gas is reduced when the moisture content of the coal is reduced.

For this thermal integration analysis, the heat available from the compressors was used to reduce the moisture content of the coal from 28.09% to 15%, if possible. If there was not enough heat available to dry the coal to 15% moisture, then it was dried as much as possible with the available heat. The coal drying analyses were carried out using a coal dryer model developed by Lehigh graduate student Joshua Charles (2011). This program simulates a fluidized bed coal dryer that uses hot air and hot water to dry the coal. For more detail on this coal dryer and how the model was created, please refer to Charles' thesis (2011).

This model was used to analyze the coal drying process with the heat available from the different compressors, and to determine how this coal drying would affect the performance of the plant. The operation of the plant with the dried coal was also analyzed in ASPEN Plus, and the results of the coal dryer program and the results from ASPEN Plus were compared to verify accuracy. Figure 5.22 below illustrates how the coal dryer works, and this process is described in more detail below.

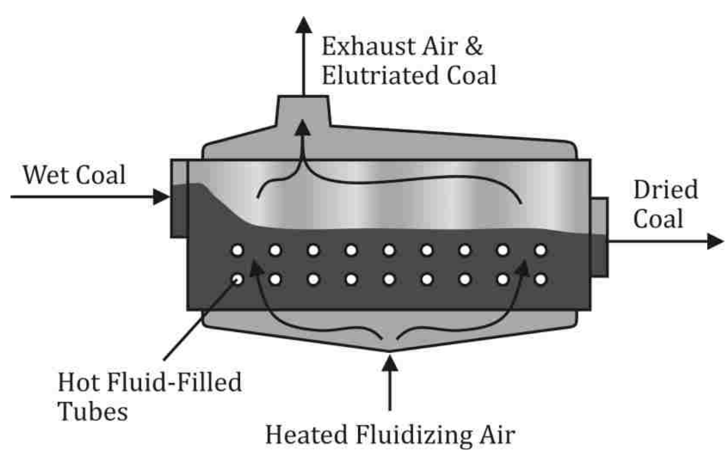

Figure 5.22 - Coal Dryer Bed Diagram (Charles, 2011).

In the fluidized bed coal dryer, hot water travels the length of the bed horizontally to transfer heat to the coal. The hot fluidizing air travels vertically through the bed to transfer heat to the coal. The dryer bed was designed for low operating temperatures that would avoid any possibility of combustion within the bed.

The hot cooling water streams available from the Ramgen, Inline 4, and IG 1-A compressors are at temperatures too high for the dryer bed. In order to reduce the temperature of the hot cooling water from these compressor coolers, the flowrate of the cooling water through the coolers was increased. Increasing these flowrates of cooling water through these heat exchangers led to lower temperatures for the hot cooling water streams than were used for the FWH and stripper reboiler thermal integrations in the previous sections. These hot cooling water streams from the compressors are also used to heat the fluidizing air before it is sent through the dryer bed.

In these analyses, the dryer bed was designed in order to have the cooling water exit the bed close to 100°F so that it can be sent back to be us ed as cooling water. In order to achieve 100°F exiting the bed, the temperature of the cooling water entering the bed was adjusted within the operating parameters of the bed. Depending on the entrance temperature that is determined, the temperature of the cooling water may need to be reduced after it heats the fluidizing air and before it enters the dryer bed, and this can be done with a cooling tower or other type of heat sink. These temperature drops were minimized in order to waste the least amount of heat as possible.

The original flowrate of the PRB coal was 648,177 lb/hr which led to a production of 587.8 MW without the capture system. The amount of CO_2 that was captured and compressed from this amount of burned coal was 1,096,118 lb/hr. In the following sections, the heat available from each compressor design and the stripper condenser were used for coal drying in order to determine the effect that this heat integration had on the plant's performance.

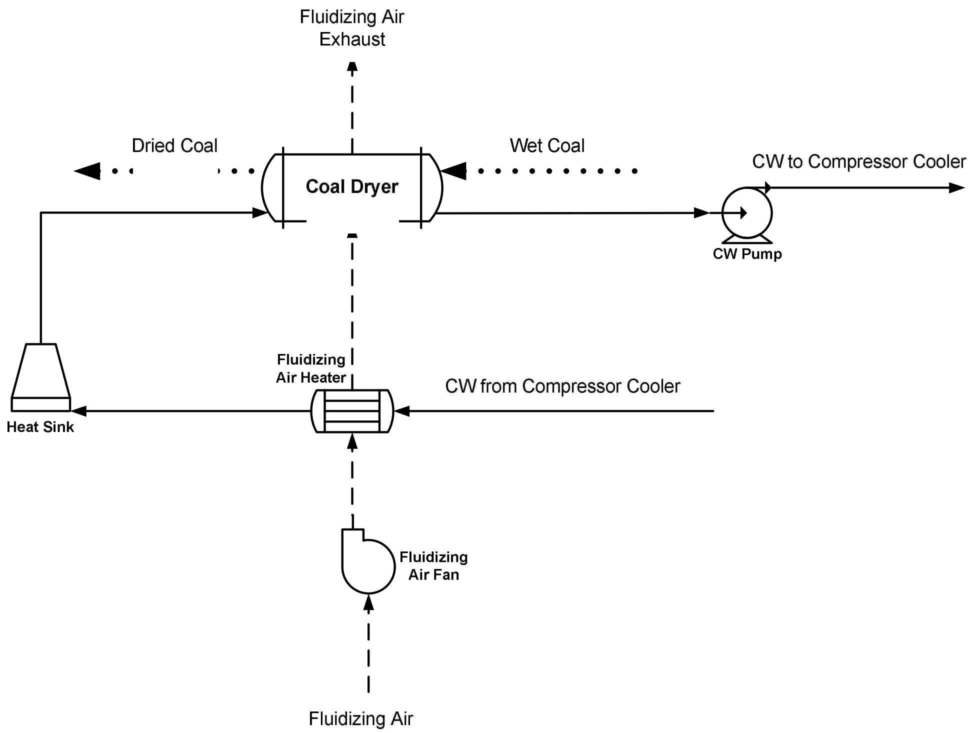

Figure 5.23 - Coal Drying Process.

Figure 5.23 illustrates the coal drying process including the fluidizing air heater and the dryer bed.

5.2.3.1 Ramgen

The Ramgen compressor's post-cooler has enough heat available to dry the coal to 15% moisture. When the coal is dried to 15%, the required flowrate of wet coal is reduced to 630,216 lb/hr which is a 2.77% reduction in coal burned. This also decreases the flue gas produced that the capture system has to process. The power of the liquid pump is reduced to 1,125.6 kW, and the heat duties of the CO_2 separation and NH_3 abatement reboilers are reduced to 1,021.3 and 340.0 MBtu/hr, respectively. The CO_2 stream that is sent to the compression cycle is now 1,064,769 lb/hr, which is a 2.86% reduction in CO_2 that has to be compressed. This reduces the compressor power to 22,519 kW.

The heat available from the compressor post-cooler is reduced to 169.2 MBtu/hr when the amount of CO_2 compressed is reduced. In order to use the cooling water from the post-cooler for coal drying, the flowrate of cooling water is increased which lowers the temperature of the hot cooling water stream exiting the post-cooler.

This hot cooling water first passes through the fluidizing air-heater to heat the fluidizing air, and this decreases the temperature of the cooling water, but it is still too high for the dryer bed. Therefore, the hot cooling water temperature was reduced after it heated the fluidizing air and before it entered the dryer bed. This resulted in 36.9 MBtu/hr in the hot cooling water not being used in this thermal integration, therefore only 75.2% of the heat available from the Ramgen compressor was used for coal drying.

Table 5.11 below shows the results of the thermal integration to coal drying for the Ramgen compressor. A more detailed table of these results can be found in Appendix C.2.1.

Table 5.11 – Results for Thermal Integration to Coal Drying for Ramgen Compressor.

	Coal Drying to 15%	No Thermal Integration
Net Power (kW)	452,481	448,181
Δ Net Power (kW)	4,300	--
Unit HR (Btu/kWh)	11,736	12,276
Δ HR (Btu/kWh)	-540	--
% HR Improvement	4.40%	--
Unit Efficiency (%)	29.07%	27.79%
Δ Unit Efficiency (%)	1.28%	--
Wet Coal Flow (lb/hr)	630,216	648,177
% Less Coal Burned	-2.77	--
CO_2 to Stack (lb/hr)	118,308	121,791
Δ CO_2 to Stack (lb/hr)	3,483	--

It can be seen that using the heat available to dry the coal to 15% moisture has a much greater plant performance improvement than integrating the heat to the FWHs or the stripper reboiler. Drying the coal to 15% moisture resulted in a net power increase of 4.3 MW, a 4.4% heat rate improvement and an increase of 1.28% for the unit efficiency. In addition, the amount of coal being burned was reduced by 2.77% which decreased the flow of CO_2 to the stack by 3,483 lb/hr.

The amount of steam produced by the boiler remains constant, so the increase in net power from coal drying is a result of the reduction of flue gas formed with the dried coal. Since less flue gas is being produced, the capture system requires less energy to process the flue gas. As a result, the power requirements of the rich amine pump and the compressors decrease. The heat duties of the CO_2 separation reboiler and the NH_3 abatement reboiler also decrease, and this reduces the required steam extractions from the steam turbine cycle resulting in more net power.

The decrease in coal burned is due to an increased HHV of the dried coal while maintaining the Btu's per hour of energy required from the coal entering the boiler (see Equation 5.1). The decrease in coal burned is also due to less carbon being required to evaporate the moisture in the coal due to the decreased moisture content.

5.2.3.2 Inline 4

The Inline compressor's inter-cooler has enough heat available to dry the coal to 15% moisture as with the Ramgen compressor's post-cooler. When the coal is dried to 15%, the plant operates the same as explained previously for the Ramgen compressor in Section 5.2.3.1. The only difference for the Inline compressor is the compressor power and the heat available from the compressor inter-cooler. When the CO_2 flowrate is reduced as a result of drying the coal, the compressor power is reduced to 21,382 kW.

The heat available from the compressor inter-cooler is reduced to 163.6 MBtu/hr when the amount of CO_2 compressed is reduced. The dryer bed was designed the same as explained for the Ramgen in Section 5.2.3.1. In order to use the cooling water from the inter-cooler for coal drying, the flowrate of cooling water is increased which lowers the temperature of the hot cooling water stream exiting the inter-cooler.

This hot cooling water first passes through the fluidizing air-heater to heat the fluidizing air, and this decreases the temperature of the cooling water, but it is still too high for the dryer bed. Therefore, the hot cooling water temperature was reduced after it heated the fluidizing air and before it entered the dryer bed. This meant that 31.3 MBtu/hr in the hot cooling water was not used in this thermal integration, therefore only 77.7% of the heat available from the Inline 4 compressor was used for coal drying.

Table 5.12 below shows the results of the thermal integration to coal drying for the Inline compressor. A more detailed table of these results can be found in Appendix C.2.2.

Table 5.12 – Results for Thermal Integration to Coal Drying for Inline 4 Compressor.

	Coal Drying to 15%	No Thermal Integration
Net Power (kW)	453,618	449,862
Δ Net Power (kW)	3,756	--
Unit HR (Btu/kWh)	11,706	12,140
Δ HR (Btu/kWh)	-434	--
% HR Improvement	3.58%	--
Unit Efficiency (%)	29.15%	28.10%
Δ Unit Efficiency (%)	1.04%	--
Wet Coal Flow (lb/hr)	630,216	648,177
% Less Coal Burned	-2.77	--
CO_2 to Stack (lb/hr)	118,308	121,791
Δ CO_2 to Stack (lb/hr)	3,483	--

It can be seen that using the heat available to dry the coal has a much greater plant performance improvement than integrating the heat to the FWHs or the stripper reboiler. Drying the coal to 15% moisture resulted in a 3.58% heat rate improvement and an increase of 1.04% for the unit efficiency. Like the Ramgen compressor, reducing the coal moisture to 15% reduces the amount of coal being burned by 2.77% and the amount of CO_2 going to the stack by 3,483 lb/hr.

5.2.3.3 Integrally-Geared 1-A

Two coal drying heat integrations were analyzed for the IG 1-A compressor. The first option uses just the cooling water from the post-cooler (PC) which has the most heat available, and with this design, there is enough heat available to dry the coal to 17.38% moisture. This design uses the cooling water from the post-cooler to heat the air in the fluidizing air-heater and then to dry the coal in the dryer bed.

When the coal is dried to 17.38%, the required flowrate of coal is reduced to 632,268 lb/hr which is a 2.45% reduction in coal burned. This also decreases the flue gas produced that the capture system has to process. The power of the liquid pump is reduced to 1,130.1 kW, and the heat duties of the CO_2 separation and NH_3 abatement reboilers are reduced to 1,025.4 and 341.4 MBtu/hr, respectively. The CO_2 stream that is sent to the compression cycle is now 1,069,004 lb/hr, which is a 2.47% reduction in CO_2 that has to be compressed. This reduces the compressor power to 19,671 kW.

The total heat available from the compressor coolers is reduced to 157.8 MBtu/hr when the amount of CO_2 compressed is reduced. In order to use the cooling water from the post-cooler for coal drying, the flowrate of cooling water is increased which lowers the temperature of the hot cooling water stream exiting the post-cooler.

This hot cooling water first passes through the fluidizing air-heater to heat the fluidizing air, and this decreases the temperature of the cooling water, but it is still too high for the dryer bed. Therefore, the hot cooling water temperature was reduced after it heated the fluidizing air and before it entered the dryer bed. This meant that 2.3 MBtu/hr in the post-cooler hot cooling water was not used in this thermal integration, and the cooling water from the two inter-coolers was not used either. This resulted in only 69.3% of the total heat available from the IG 1-A compressor being used for coal drying.

The second option uses the cooling water from the post-cooler (PC) in the dryer bed, and the cooling water from the second inter-cooler (IC-2) in the fluidizing air heater. With this design, there is enough heat available to dry the coal to 16.06% moisture.

When the coal is dried to 16.06%, the required flowrate of coal is reduced to 631,099 lb/hr which is a 2.63% reduction in coal burned. This also decreases the flue gas produced that the capture system has to process. The power of the liquid pump is reduced to 1,129.2 kW, and the heat duties of the CO_2 separation and NH_3 abatement reboilers are reduced to 1,023.1 and 340.6 MBtu/hr, respectively. The CO_2 stream that is sent to the compression cycle is now 1,066,632 lb/hr, which is a 2.69% reduction in CO_2 that has to be compressed. This reduces the compressor power to 19,626 kW.

The total heat available from the compressor coolers is reduced to 157.4 MBtu/hr when the amount of CO_2 compressed is reduced. In order to use the cooling water from the post-cooler for coal drying, the flowrate of cooling water is increased which lowers the temperature of the hot cooling water stream exiting the post-cooler, but it is still too high for the dryer bed. Therefore, the temperature of the hot cooling water from the post-cooler was reduced before it entered the dryer bed. This meant that 3.8 MBtu/hr in the post-cooler hot cooling water was not used in this heat integration.

After it heats the fluidizing air, the temperature of the cooling water from inter-cooler 2 must be reduced to 100°F before it can be used again as cooling water. This means that 13.9 MBtu/hr of available heat in this IC-2 stream is not used. The cooling water from the first inter-cooler was not used for this heat integration, and this resulted in 76.0% of the total heat available from the IG 1-A compressor being used for coal drying.

Table 5.13 below shows the results of the coal drying thermal integrations for the IG 1-A compressor. A more detailed table of these results can be found in Appendix C.2.3.

Table 5.13 – Results for Thermal Integration to Coal Drying for IG 1-A Compressor.

	PC → Coal Drying to 17.38%	IC-2 → Air Heater PC → Coal Drying to 16.06%	No Thermal Integration
Net Power (kW)	454,626	454,564	451,687
Δ Net Power (kW)	2,939	2,877	--
Unit HR (Btu/kWh)	11,718	11,698	12,091
Δ HR (Btu/kwh)	-373	-393	--
% HR Improvement	3.09%	3.25%	--
Unit Efficiency (%)	29.12%	29.17%	28.01%
Δ Unit Efficiency (%)	0.90%	0.95%	--
Wet Coal Flow (lb/hr)	632,268	631,099	648,177
% Less Coal Burned	-2.45%	-2.63%	--
CO_2 to Stack (lb/hr)	118,783	118,515	121,791
Δ CO_2 to Stack (lb/hr)	3,008	3,276	--

It can be seen that using the heat available to dry the coal has a much greater plant performance improvement than integrating the heat to the FWHs or the stripper reboiler. Drying the coal to 17.38% moisture resulted in a 3.1% heat rate improvement and an increase of 0.90% for the unit efficiency. In addition, the amount of coal being burned was reduced by 2.45%. Drying the coal to 16.06% moisture resulted in a 3.3% heat rate improvement and an increase of 0.95% for the unit efficiency. In addition, the amount of coal being burned was reduced by 2.63%.

5.2.3.4 Integrally-Geared 1-B

The IG 1-B compressor coolers have enough heat available to dry the coal to 18.24% moisture. For this compressor, the cooling water from the first inter-cooler (IC-1) was used to heat the air in the fluidizing air heater. The cooling water from the second and third inter-coolers (IC-2 and IC-3) were mixed together and used for the coal dryer bed. The cooling water from the post-cooler has minimal heat available, so this stream was not used for coal drying.

When the coal is dried to 18.24%, the required flowrate of coal is reduced to 633,053 lb/hr which is a 2.33% reduction in coal burned. This also decreases the flue gas produced that the capture system has to process. The power of the liquid pump is reduced to 1,132.5 kW, and the heat duties of the CO_2 separation and NH_3 abatement reboilers are reduced to 1,027.5 and 342.1 MBtu/hr, respectively. The CO_2 stream that is sent to the compression cycle is now 1,071,236 lb/hr, which is a 2.27% reduction in CO_2 that has to be compressed. This reduces the compressor power to 15,769 kW.

The temperature of the hot cooling water from IC-2 and IC-3 was too high for the dryer bed, so the temperature was reduced before it entered the dryer bed. This meant that 1.78 MBtu/hr in the hot cooling water from IC-2 and IC-3 was not used in this heat integration. From the IC-1 cooling water, 10.18 of the 30.97 MBtu/hr of available heat was used to heat the fluidizing air. The heat available in the post-cooler was not used for coal drying, and this resulted in 67.5% of the total heat available from the IG 1-B compressor being used for coal drying.

Table 5.14 below shows the results of the thermal integration to coal drying for the IG 1-B compressor. A more detailed table of these results can be found in Appendix C.2.4.

Table 5.14 – Results for Thermal Integration to Coal Drying for IG 1-B Compressor.

	Coal Drying to 18.24%	No Heat Integration
Net Power (kW)	458,146	455,477
Δ Net Power (kW)	2,668	--
Unit HR (Btu/kWh)	11,643	11,991
Δ HR (Btu/kwh)	-348	--
% HR Improvement	2.90%	--
Unit Efficiency (%)	29.31%	28.46%
Δ Unit Efficiency (%)	0.85%	--
Wet Coal Flow (lb/hr)	633,053	648,177
% Less Coal Burned	-2.33%	--
CO_2 to Stack (lb/hr)	119,026	121,791
Δ CO_2 to Stack (lb/hr)	2,765	--

It can be seen that using the heat available to dry the coal has a much greater plant performance improvement than integrating the heat to the FWH-1. Drying the coal to 18.24% moisture resulted in a 2.9% heat rate improvement and an increase of 0.85% for the unit efficiency. In addition, the amount of coal being burned was reduced by 2.33%.

5.2.3.5 Comparison

Using the heat from compression for coal drying is more advantageous than using the heat for the stripper reboiler or the FWHs. This is because drying the coal reduces the reboiler duties and the power requirements of the capture system and the compressor cycle. This provides a greater plant improvement than just reducing the reboiler duty or reducing steam extractions for FWHs. Drying the coal also increases the boiler efficiency and therefore reduces the amount of coal that must be burned in the boiler. In addition, less CO_2 is formed and therefore less CO_2 is sent to the stack and released into the atmosphere when the coal is dried.

The IG 1-B compressor has the best plant performance of any of the compressors when heat integration to coal drying is considered. This is due to the initial lower heat rate and compressor power compared to the other compressors. Even though the heat available from this compressor cannot dry the coal to 15% moisture, it can dry the coal to 18.24% which provides a significant improvement to the plant performance.

Figures 5.24 through 5.30 illustrate the plant performance for each compressor design when thermal integration to coal drying is analyzed. Figure 5.25 illustrates the change in net power when thermal integration to coal drying is considered, and this chart uses the Ramgen with no thermal integration as the base case.

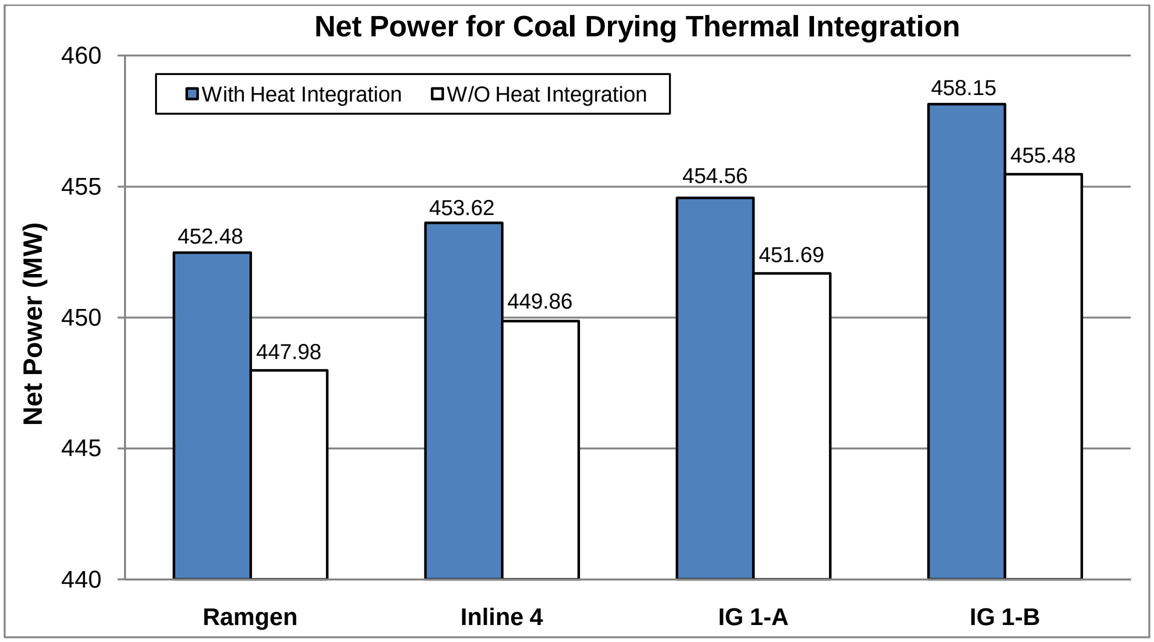

Figure 5.24 - Net Power Results for Thermal Integration to Coal Drying.

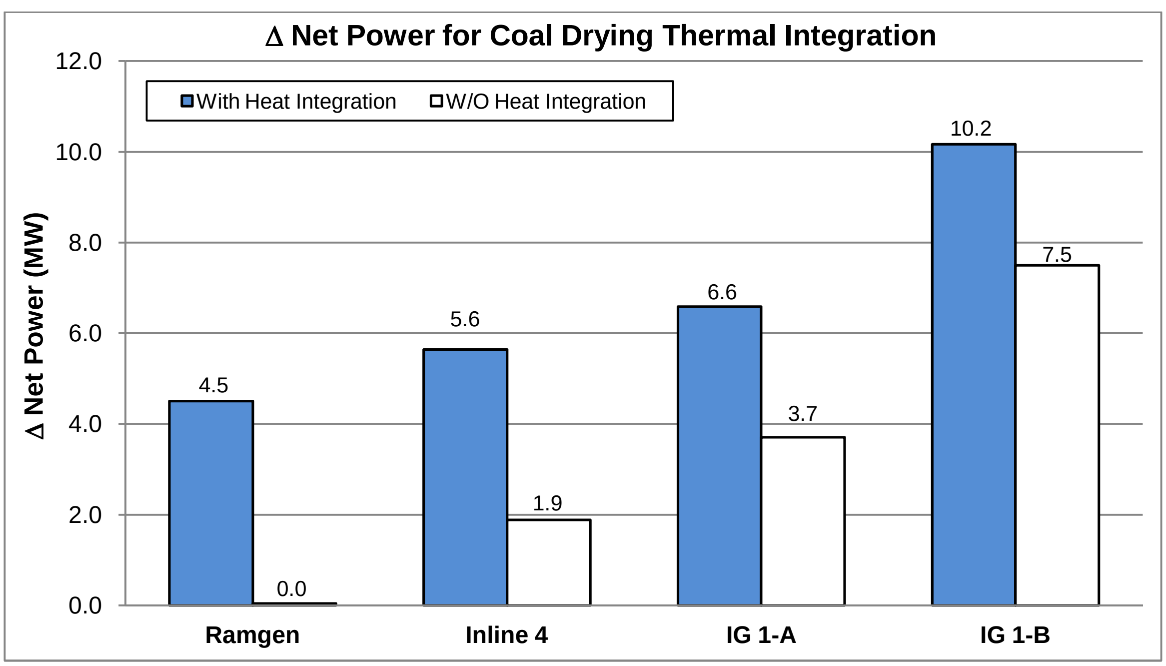

Figure 5.25 - Δ Net Power Results for Thermal Integration to Coal Drying.

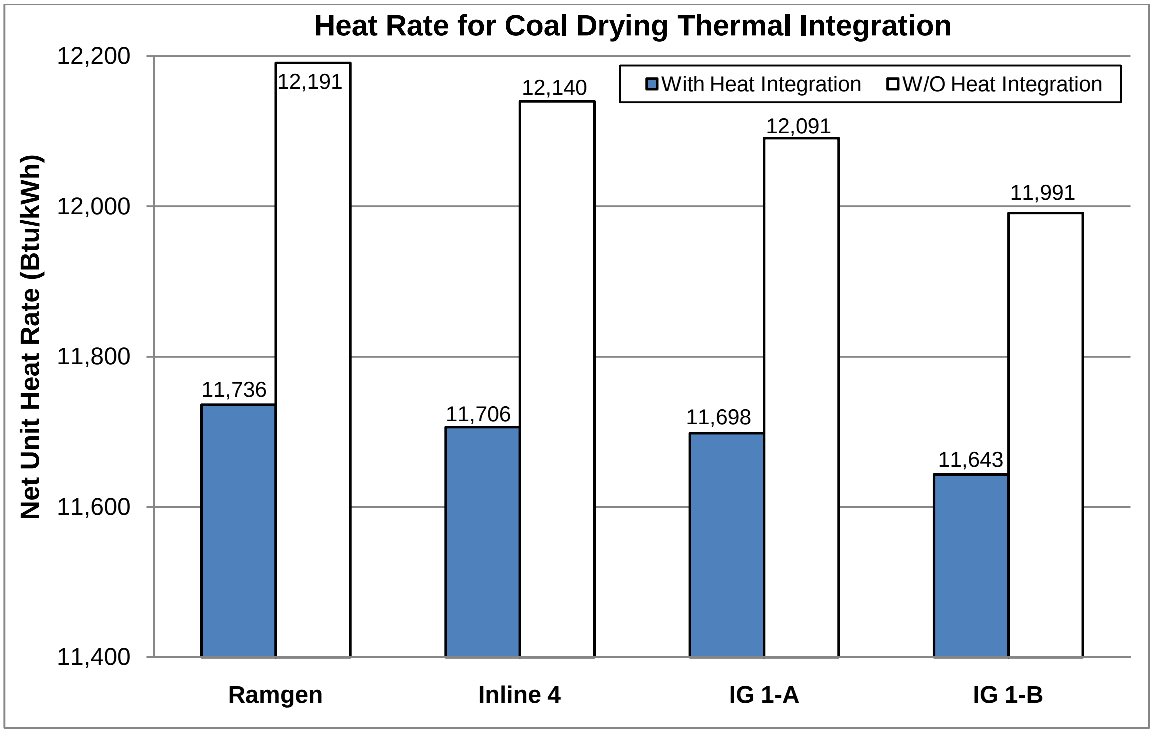

Figure 5.26 - Heat Rate Results for Thermal Integration to Coal Drying.

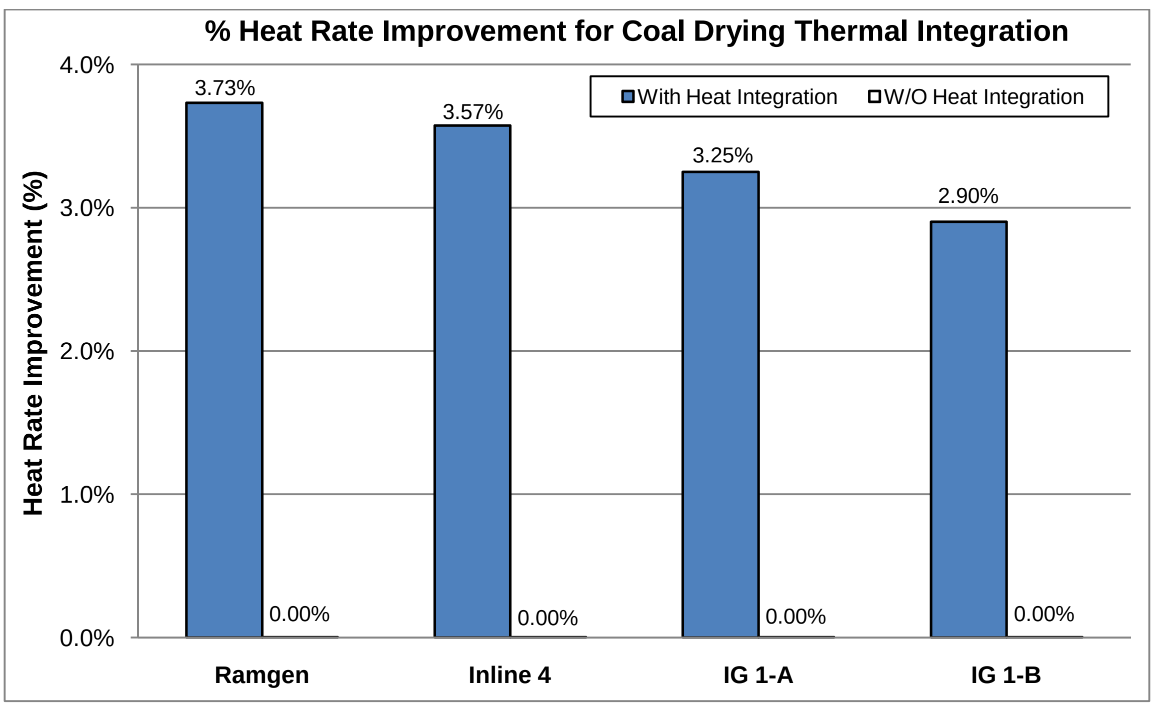

Figure 5.27 - Heat Rate Improvement for Thermal Integration to Coal Drying.

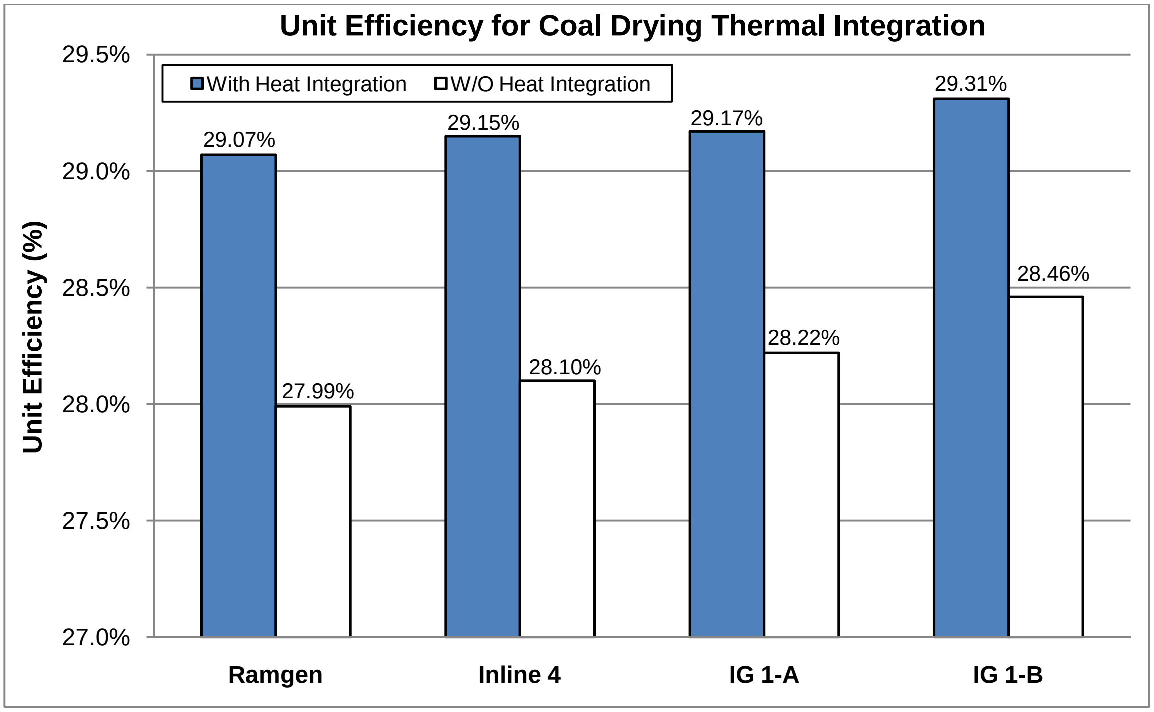

Figure 5.28 - Unit Efficiency for Thermal Integration to Coal Drying.

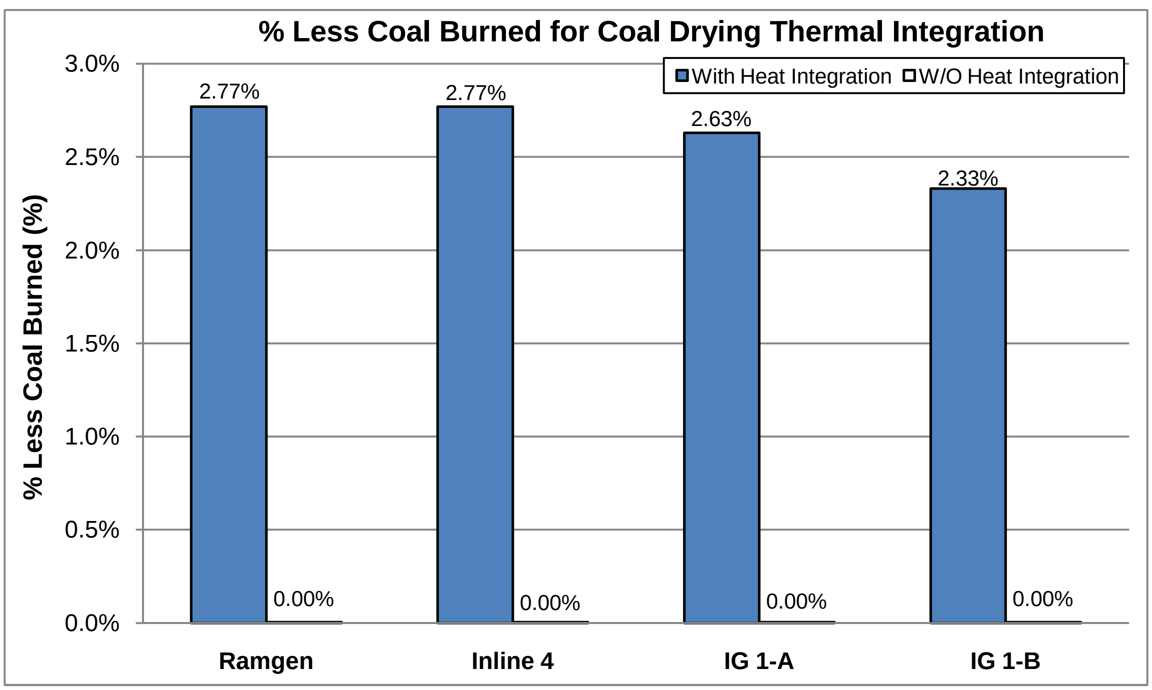

Figure 5.29 - % Less Coal Burned for Thermal Integration to Coal Drying.

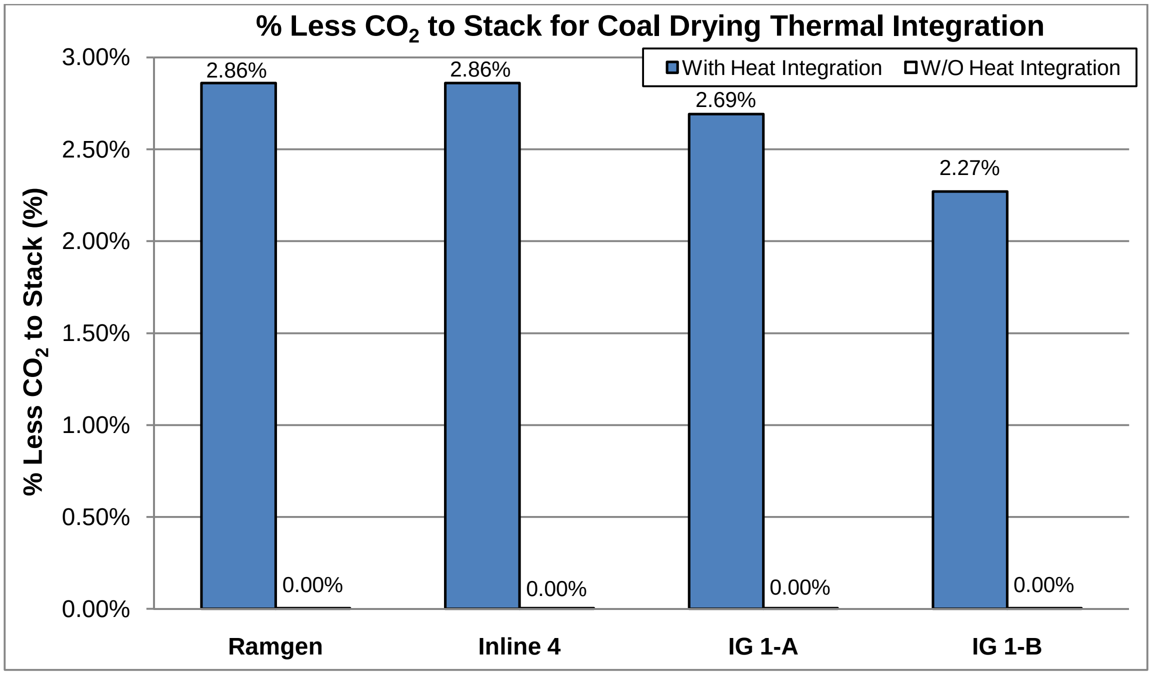

Figure 5.30 - % Less CO_2 to Stack for Thermal Integration to Coal Drying.

Even though the IG 1-B compressor has the best plant performance when thermal integration to coal drying is done, the Ramgen and Inline 4 reduce the amount of coal burned and the amount of CO_2 sent to the stack more than the two IG compressor designs. All of the compressors have significant plant performance improvement when the available heat is used for coal drying. A comparison of the FWH, stripper reboiler, and coal drying thermal integration results is done in Section 6.

5.2.4 Stripper Condenser Thermal Integrations

The condenser in the CO_2 separation stripper also has heat available that can be integrated in the plant in order to improve the plant performance. The condenser does not have enough heat available to integrate to the stripper reboiler or for coal drying, so thermal integration to the FWHs was analyzed for this heat source. Since the condenser heat can be used in the FWHs, some combinations of using the condenser heat in the FWHs while using the compressor heat in the stripper reboiler were also analyzed. Combinations of integrating the condenser heat to FWH-2 while integrating the compressor heat to coal drying could also be considered, but these were not analyzed in this thesis.

5.2.4.1 Feedwater Heaters

The heat available from the stripper condenser is 78.47 MBtu/hr, which is available in 594,566 lb/hr of 231.8°F water. The stripper condenser hea t cannot be integrated to FWH-3 due to the temperature of the hot cooling water being too low. Since it was shown with the different compressors that integrating heat to FWH-2 was better than integrating to FWH-1, the heat available from the stripper condenser was integrated to FWH-2. When this heat is integrated to FWH-2, the steam extraction to FWH-2 can be reduced to 71,250 lb/hr, and the steam extraction required for FWH-1 is 148,200 lb/hr.

The amount of heat available from the stripper condenser is independent of which compressor design is used, but the plant performance is dependent on the power requirements of the compressor. The results of integrating the stripper condenser heat to the FWHs are calculated with each type of compressor design since the compressor power must be included in the plant performance calculations. For these calculations, no heat is integrated from the compressors and only the integration of the stripper condenser heat is considered.

The improvement of the plant performance as a result of integrating the stripper condenser heat to FWH-2 is tabulated below in Table 5.15 along with the plant performance when no thermal integration is done for each compressor design. A more detailed table for this thermal integration can be found in Appendix C.3.

Table 5.15 - Thermal Integration to FWH-2 for Stripper Condenser.

	Ramgen		**Inline**	
	Cond. to FWH-2	No Thermal Integration	Cond. to FWH-2	No Thermal Integration
Net Power (kW)	450,358	447,980	452,240	449,862
∆ Net Power (kW)	2,378	--	2,378	--
Heat Rate (Btu/kWh)	12,127	12,191	12,077	12,140
∆ Heat Rate (Btu/kWh)	-64	--	-63	--
% HR Improvement	0.53%	--	0.53%	--
Unit Efficiency (%)	28.14%	27.99%	28.25%	28.10%
∆ Unit Efficiency (%)	0.15%	--	0.15%	--

continued on next page

	IG 1-A		IG 1-B	
	Cond. to FWH-2	No Thermal Integration	Cond. to FWH-2	No Thermal Integration
Net Power (kW)	454,066	451,687	457,855	455,477
Δ Net Power (kW)	2,379	--	2,379	--
Heat Rate (Btu/kWh)	12,028	12,091	11,929	11,991
Δ Heat Rate (Btu/kWh)	-63	--	-62	--
% HR Improvement	0.52%	--	0.52%	--
Unit Efficiency (%)	28.37%	28.22%	28.61%	28.46%
Δ Unit Efficiency (%)	0.15%	--	0.15%	--

The change in net power, heat rate, and unit efficiency are the same for each compressor design since integrating the condenser heat to FWH-2 adds the same amount of power for each compressor design. It is more advantageous to integrate the heat from the compressors to the FWHs than to integrate the heat from the condenser, except for the IG 1-B compressor. This is because the heat from the IG 1-B compressor can only be integrated to FWH-1, where the heat from the other compressor designs can be integrated to FWH-3.

5.2.4.2 Thermal Integration Combination 1

The first combination of thermal integration uses the heat available from the compressor for the stripper reboiler while the heat available from the stripper condenser is integrated to the FWHs. The best case for integrating the stripper condenser heat to the FWHs is to use it in FWH-2 since the temperature is too low to integrate to FWH-3. The IG 1-B compressor design does not have enough heat available to integrate to the stripper reboiler, so this compressor design was not analyzed in these thermal integration combination cases. When the condenser heat is integrated to FWH-2, all of the heat available is used, but when the compressor heat is integrated to the stripper reboiler, not all of the heat available is utilized.

Table 5.16 below illustrates the plant performance for this combination of thermal integration methods for the three compressor designs where heat integration to the stripper reboiler was possible. Also included in this table are the individual results of integrating the compressor heat to the reboiler and integrating the stripper condenser heat to FWH-2. A more detailed table for this thermal integration combination can be found in Appendix C.3.

Table 5.16 - Heat Integration Combination 1 Results.

Ramgen	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**Compressor to Reboiler**	**Condenser To FWH-2**	**No Thermal Integration**
Net Power (kW)	458,936	456,558	450,358	447,980
Δ Net Power (kW)	10,957	8,578	2,378	--
Heat Rate (Btu/kWh)	11,900	11,962	12,127	12,191
Δ Heat Rate (Btu/kWh)	-291	-229	-64	--
% HR Improvement	2.39%	1.88%	0.53%	--
Unit Efficiency (%)	28.67%	28.52%	28.14%	27.99%
Δ Unit Efficiency (%)	0.68%	0.54%	0.15%	--

Inline	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**Compressor to Reboiler**	**Condenser To FWH-2**	**No Thermal Integration**
Net Power (kW)	459,939	457,564	452,240	449,862
Δ Net Power (kW)	10,077	7,702	2,378	--
Heat Rate (Btu/kWh)	11,874	11,936	12,077	12,140
Δ Heat Rate (Btu/kWh)	-266	-204	-63	--
% HR Improvement	2.19%	1.68%	0.53%	--
Unit Efficiency (%)	28.73%	28.59%	28.25%	28.10%
Δ Unit Efficiency (%)	0.63%	0.49%	0.15%	--

IG 1-A	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**Compressor to Reboiler**	**Condenser To FWH-2**	**No Thermal Integration**
Net Power (kW)	456,208	453,844	454,066	451,687
Δ Net Power (kW)	4,521	2,157	2,379	--
Heat Rate (Btu/kWh)	11,972	12,034	12,028	12,091
Δ Heat Rate (Btu/kWh)	-119	-57	-63	--
% HR Improvement	0.99%	0.48%	0.52%	--
Unit Efficiency (%)	28.50%	28.35%	28.37%	28.22%
Δ Unit Efficiency (%)	0.28%	0.13%	0.15%	--

(Comp = Compressor, Cond = Stripper Condenser, Reb = Stripper Reboiler)

It can be seen in Table 5.16 that integrating the heat available from the compressor to the stripper reboiler while integrating the heat available from the stripper condenser to the FWHs provides a greater increase in plant performance than integrating either heat source alone.

The disadvantage of this heat integration combination is that it requires the addition of three heat exchangers to the original plant. One of these required heat exchangers is for the integration of the heat available from compression to the stripper reboiler. The other two heat exchangers required are for the two FWHs in order to allow all of the heat available from the stripper condenser to be used in FWH-2 and then FWH-1.

5.2.4.3 Thermal Integration Combination 2

Similar to the previous section, for this combination the heat available from the compressor is integrated to the stripper reboiler, and the heat available from the stripper condenser is integrated to FWH-2. Additionally, the remaining heat available in the compressor cooling water after it exits the reboiler is then sent to FWH-3 in order to reduce the steam extraction needed for this FWH. The heat integration to the reboiler is the same as was illustrated previously in Figure 5.16. The heat integration of these streams to the FWHs is illustrated below in Figure 5.31.

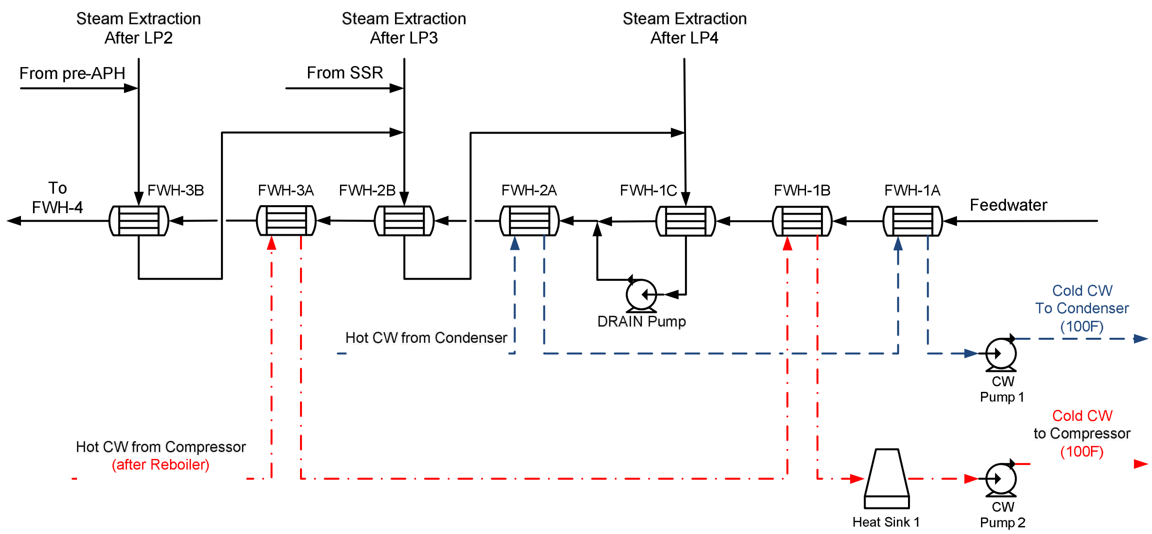

Figure 5.31 - Thermal Integration Combination 2.

The cooling water from the compressor is used in FWH-3 and is then cascaded back to the earlier FWHs. The results of this heat integration combination are shown in Table 5.17 for the three compressor designs where heat integration to the stripper reboiler was possible. More detailed results can be found in Appendix C.3.

Table 5.17 - Heat Integration Combination 2 Results.

Ramgen	**Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2**	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**No Thermal Integration**
Net Power (kW)	460,918	458,936	447,980
∆ Net Power (kW)	12,938	10,957	--
Heat Rate (Btu/kWh)	11,849	11,900	12,191
∆ Heat Rate (Btu/kWh)	-342	-291	--
% HR Improvement	2.81%	2.39%	--
Unit Efficiency (%)	28.79%	28.67%	27.99%
∆ Unit Efficiency (%)	0.80%	0.68%	--

Inline	**Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2**	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**No Thermal Integration**
Net Power (kW)	462,022	459,939	449,862
∆ Net Power (kW)	12,106	10,077	--
Heat Rate (Btu/kWh)	11,821	11,874	12,140
∆ Heat Rate (Btu/kWh)	-320	-266	--
% HR Improvement	2.63%	2.19%	--
Unit Efficiency (%)	28.86%	28.73%	28.10%
∆ Unit Efficiency (%)	0.76%	0.63%	--

IG 1-A	**Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2**	**Combination 1: Comp. to Reboiler + Cond. to FWH-2**	**No Thermal Integration**
Net Power (kW)	458,539	456,208	451,687
∆ Net Power (kW)	6,852	4,521	--
Heat Rate (Btu/kWh)	11,911	11,972	12,091
∆ Heat Rate (Btu/kWh)	-180	-119	--
% HR Improvement	1.49%	0.99%	--
Unit Efficiency (%)	28.65%	28.50%	28.22%
∆ Unit Efficiency (%)	0.43%	0.28%	--

It can be seen in Table 5.17 that integrating the heat available from compression to the stripper reboiler and then to FWH-3 is more advantageous than just integrating the heat to the stripper reboiler. This is because this thermal integration method allows a greater portion of the heat available to be utilized within the plant.

The disadvantage of this thermal integration method is that it requires the addition of four heat exchangers to the FWH cycle. These new heat exchangers include an additional heat exchanger for FWH-3, an additional heat exchanger for FWH-2, and two additional heat exchangers for FWH-1. There is also the additional heat exchanger required for the stripper reboiler for the integration of the available heat from compression.

5.2.3.3 Comparison

Since it allows more of the available heat to be utilized, integrating the compressor heat to the stripper reboiler and then to FWH-3 along with integrating the stripper condenser heat to FWH-2 provides the greatest improvement to the plant performance. The disadvantage of these heat integration combinations is the large amount of additional equipment that is required for relatively small plant performance improvements over integrating to the stripper reboiler alone. Figures 5.32 through 5.35 below illustrate the impact of these thermal integration combinations on the plant's performance when compared to no thermal integration. It can be seen in these figures that the Inline 4 has the best performance when these thermal heat integration combinations are considered.

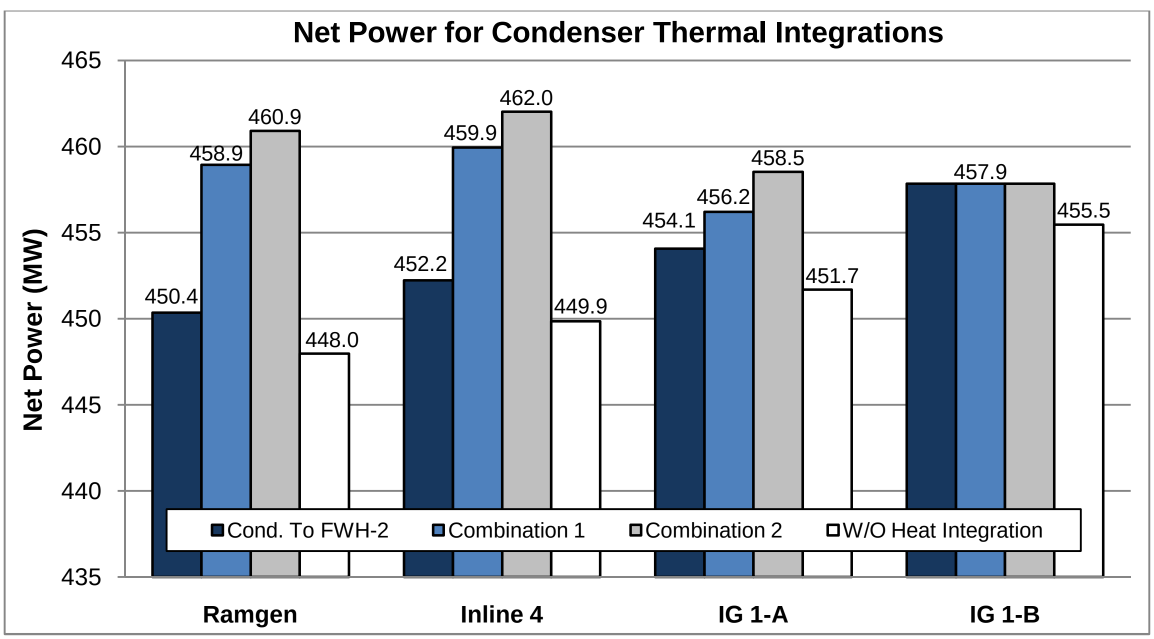

Figure 5.32 - Net Power Results for Condenser Thermal Integrations.

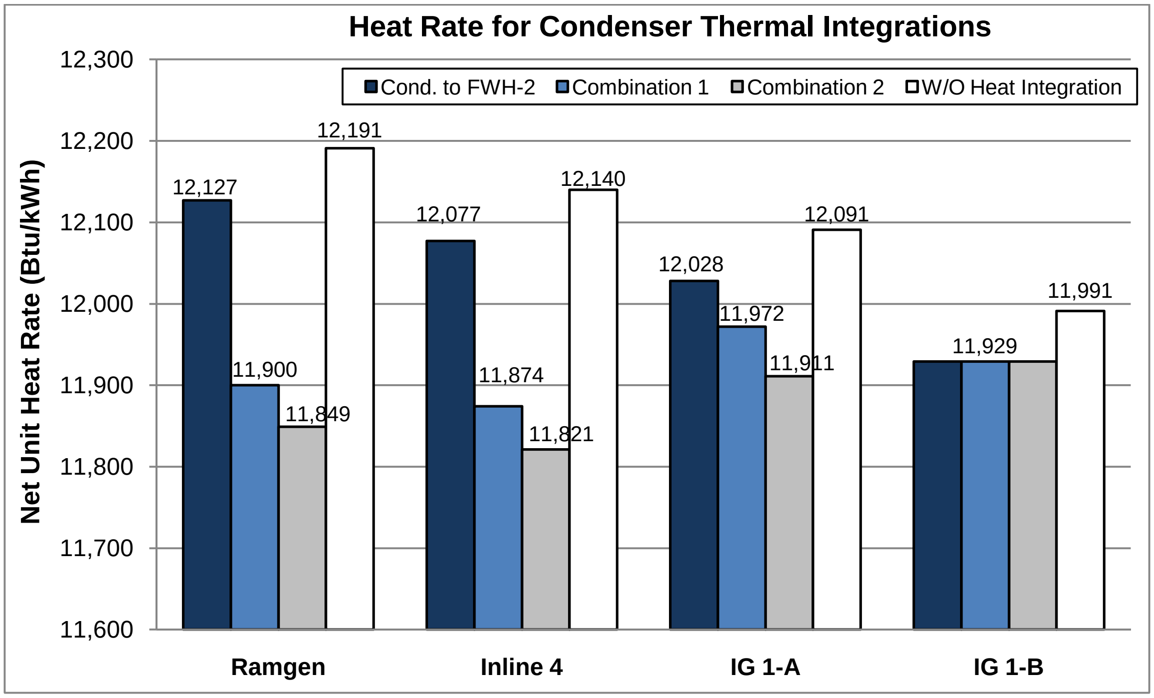

Figure 5.33 - Heat Rate Results for Condenser Thermal Integrations.

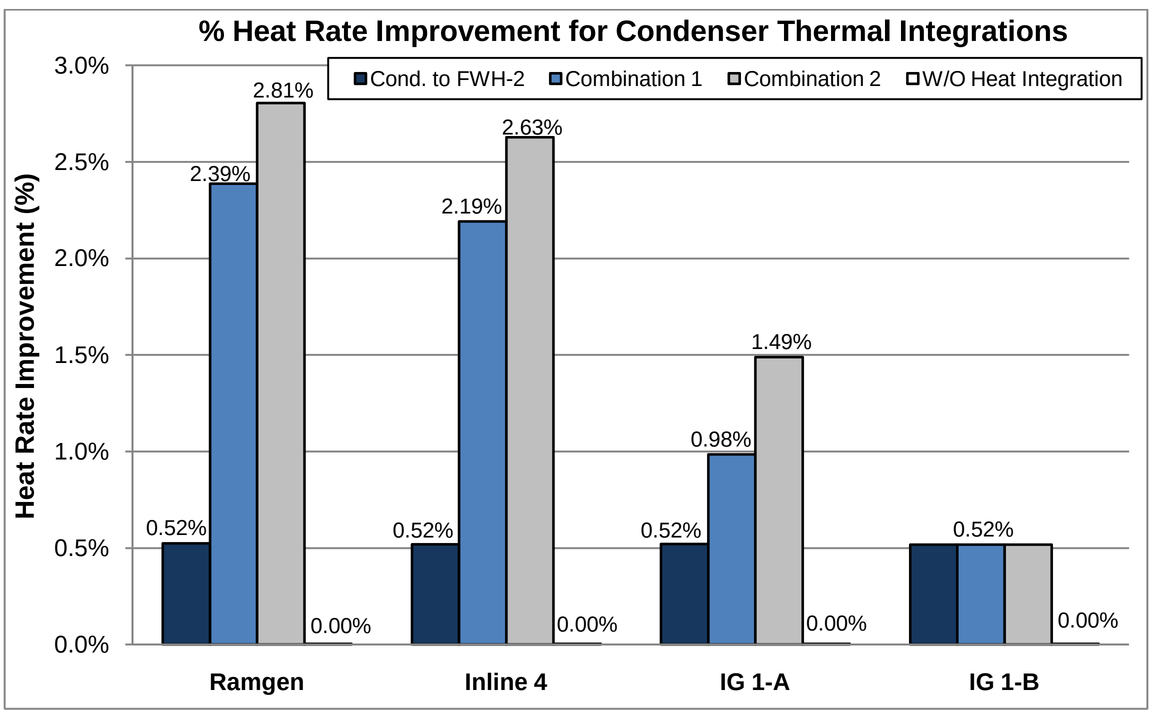

Figure 5.34 - Heat Rate Improvement for Condenser Thermal Integrations.

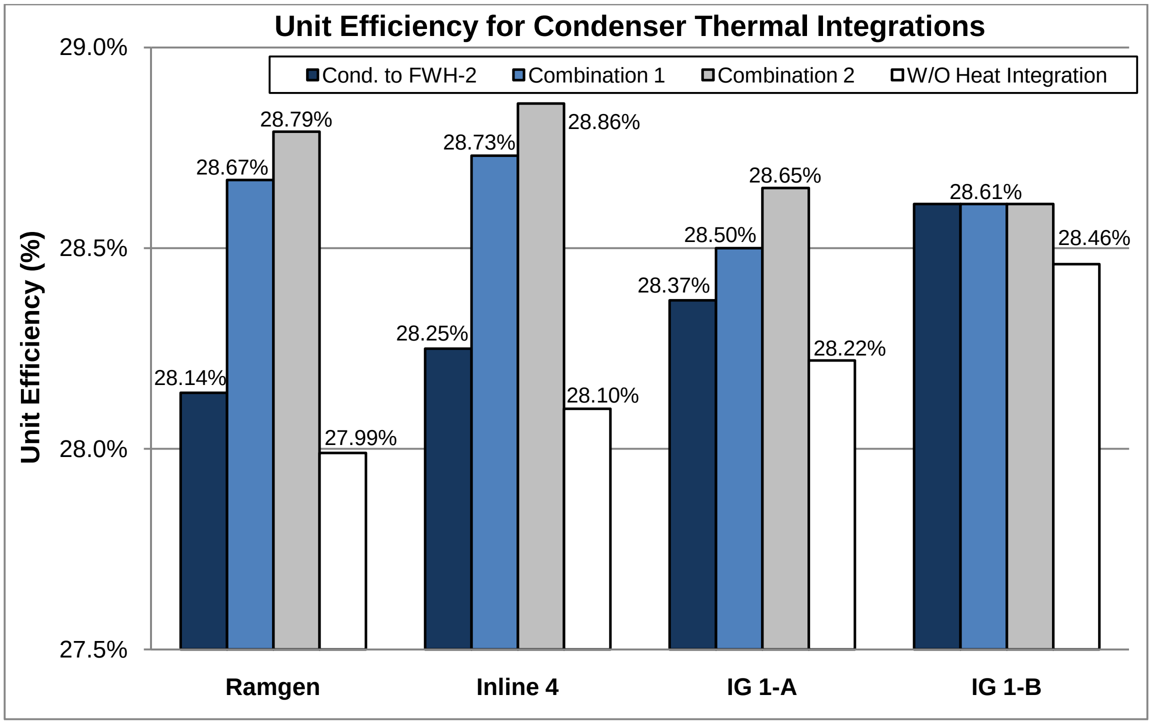

Figure 5.35 - Unit Efficiency for Condenser Thermal Integrations.

6 Conclusions

The addition of a chilled ammonia carbon capture system to the back end of a power plant causes a significant impact on the plant's performance. The additional equipment requires power and heat that must be met with steam extractions from the steam turbine cycle. Approximately 22.5 to 23.8% of the plant's power output is lost with the addition of this capture system depending on which compressor design is used. The heat rate of the plant is increased by 2,700 to 2,900 Btu/kWh, and the unit efficiency is decreased by 8.26 to 8.73%. The effect on plant performance is dependent on which compressor design is utilized and which thermal integration option, if any, is considered.

Excluded from the analyses in this thesis are the refrigeration loads that would be required to chill the flue gas and the lean ammonia streams before they enter the absorber. These refrigeration cycles would have a significant impact on the plant performance so it is important to note that this is not included when comparing this capture system to other amine-based capture systems.

Figures 6.1 through 6.7 illustrate the performance of the four compressors with the various thermal integration options studied in this thesis alongside the performance when no thermal integration is used.

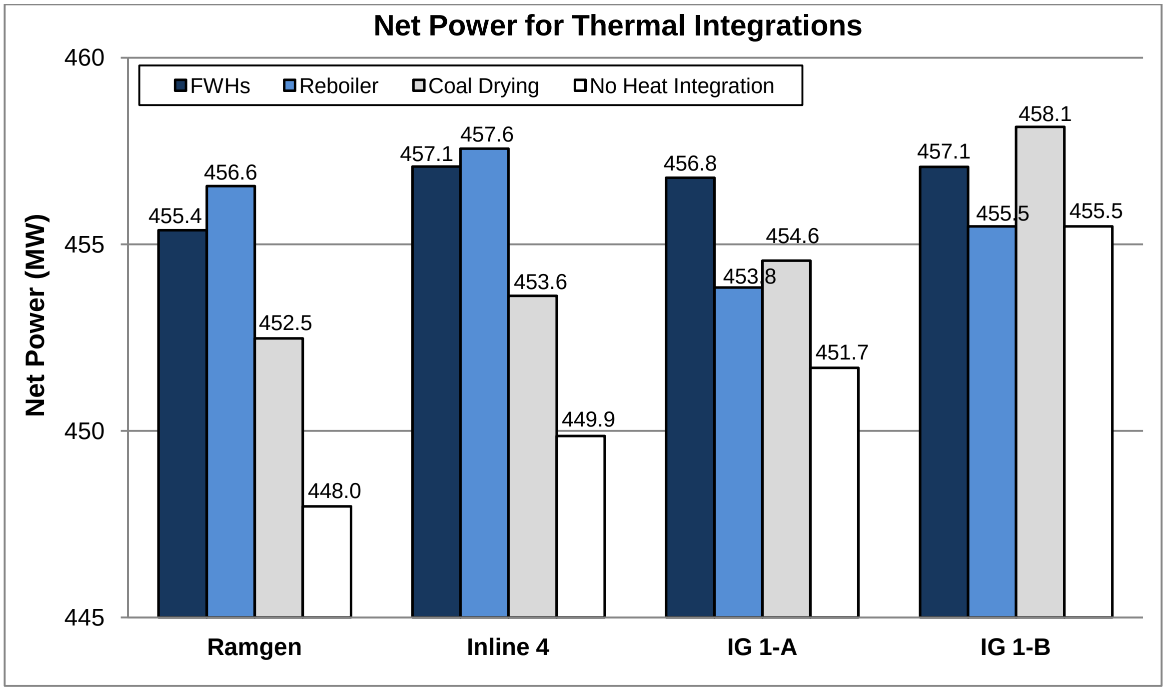

Figure 6.1 - Net Power for Thermal Integrations.

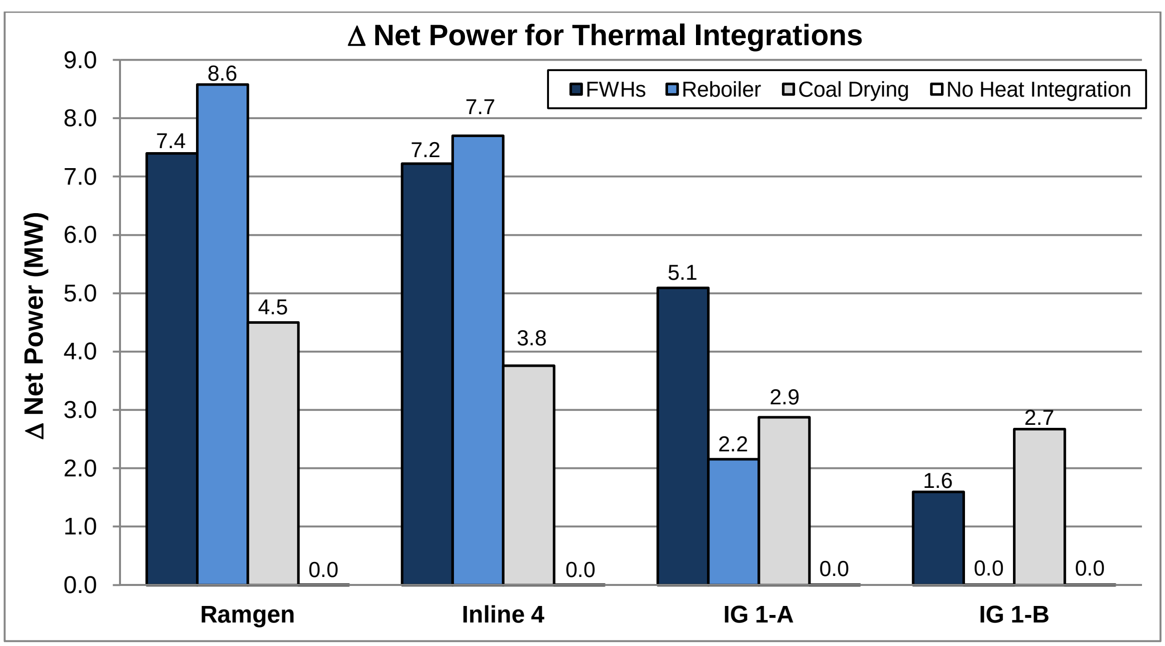

Figure 6.2 - Δ Net Power for Thermal Integrations.

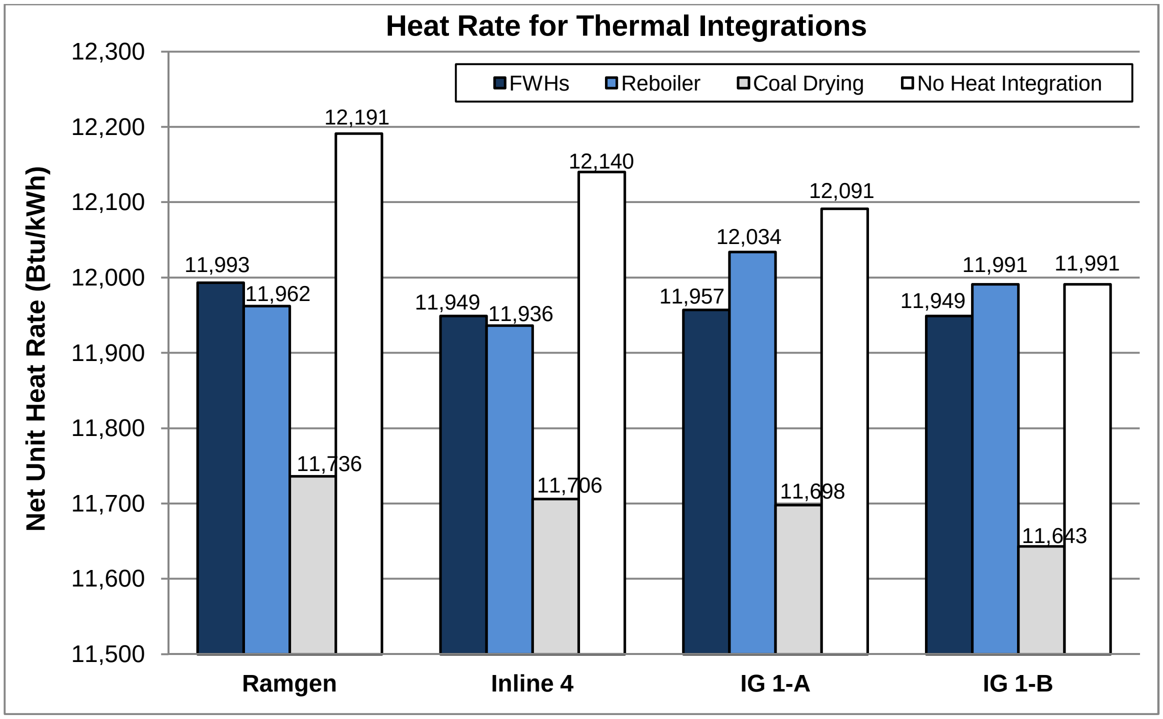

Figure 6.3 - Heat Rate for Thermal Integrations.

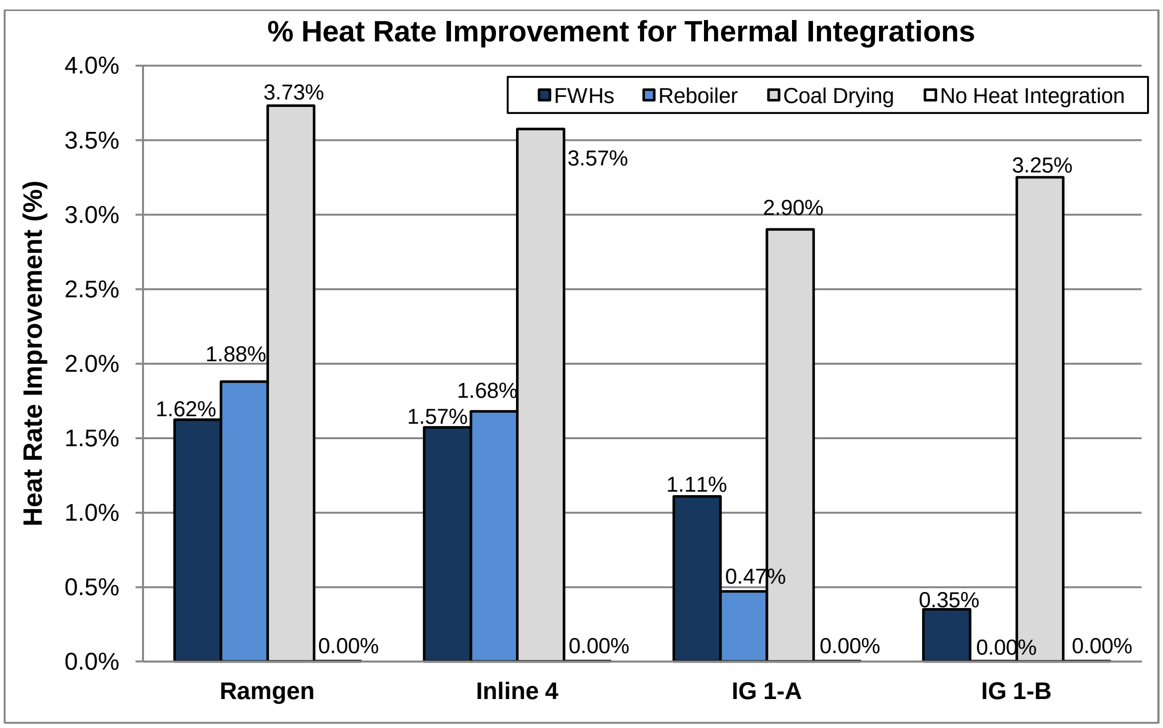

Figure 6.4 - Heat Rate Improvement for Thermal Integrations.

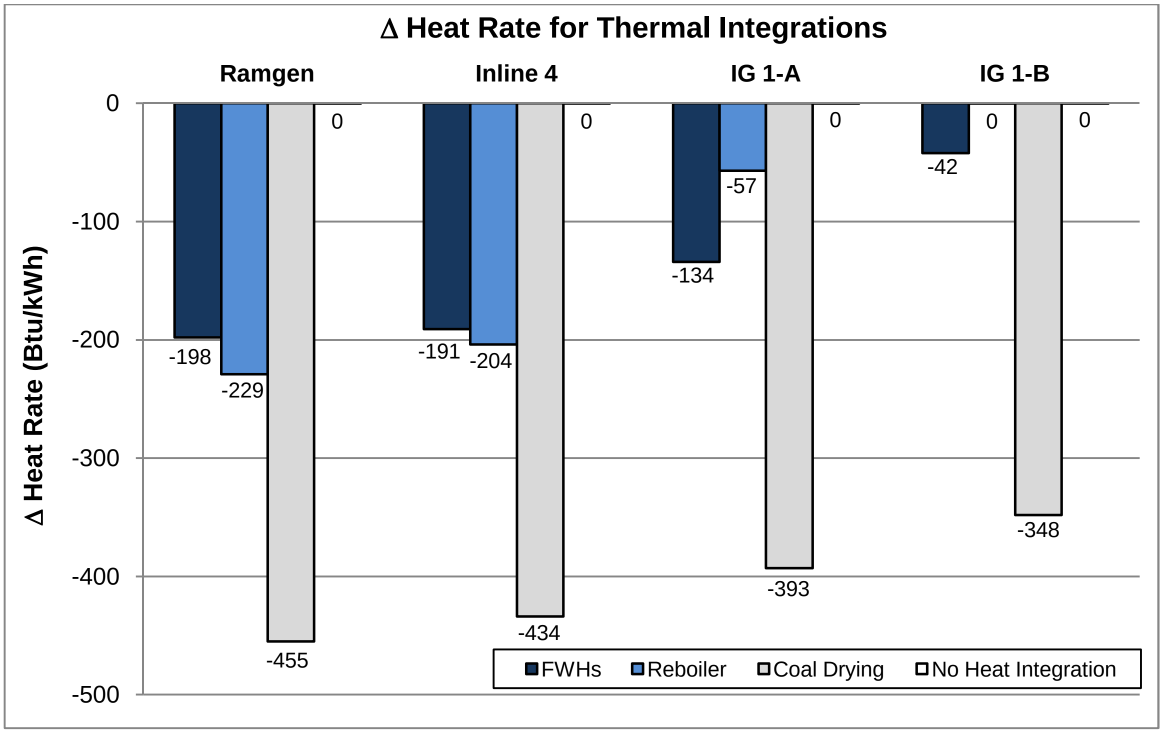

Figure 6.5 - Δ Heat Rate for Thermal Integrations.

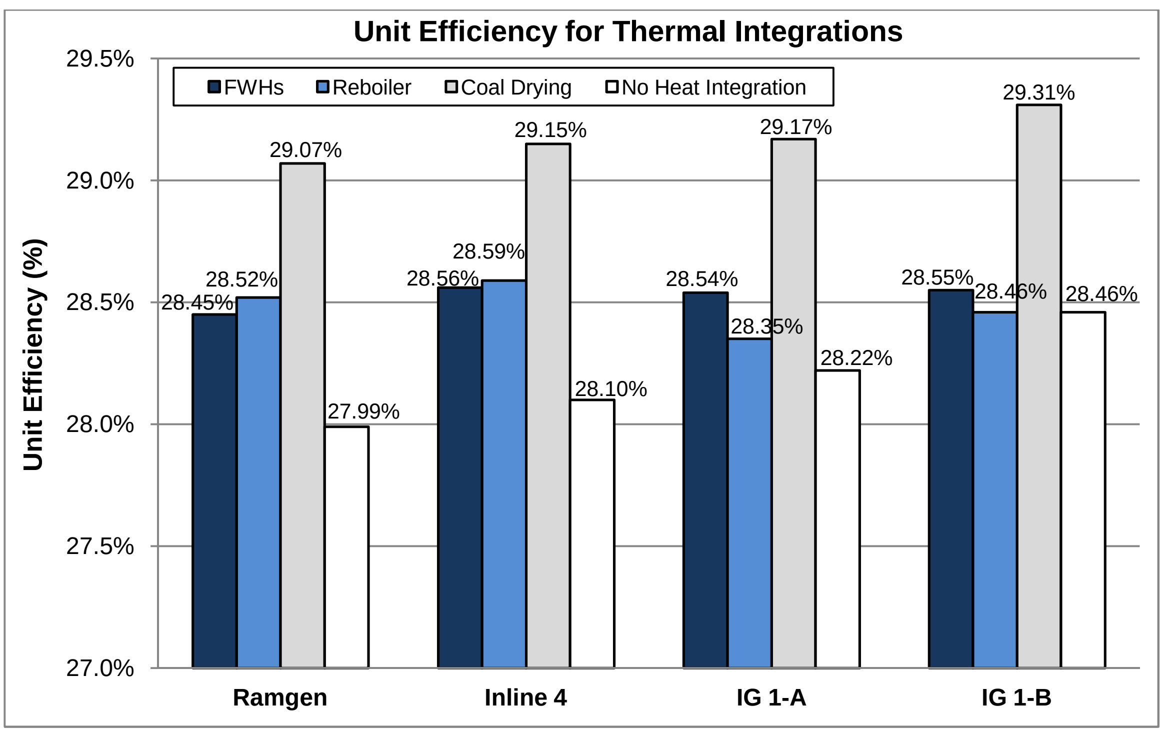

Figure 6.6 - Unit Efficiency for Thermal Integrations.

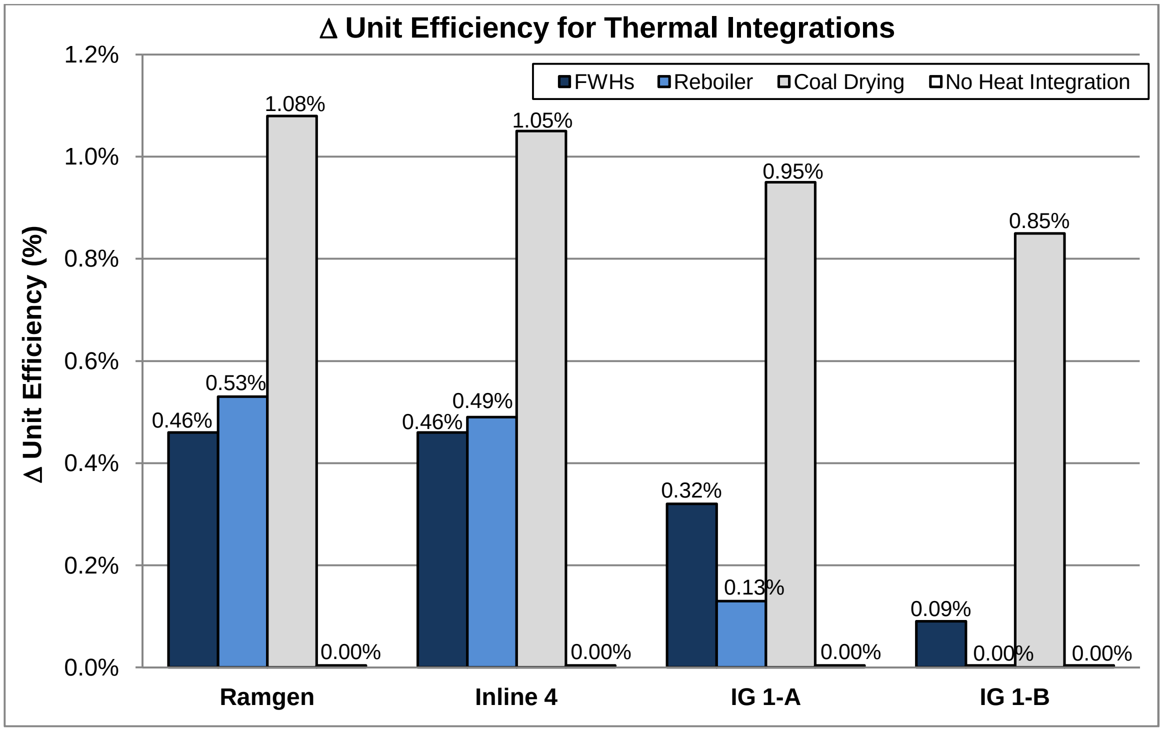

Figure 6.7 - ∆ Unit Efficiency for Thermal Integrations.

When no thermal integration is considered, the IG 1-B has the best plant performance of the compressors. For the IG 1-B compressor, the plant produces the most net power while having the lowest heat rate and highest efficiency. The plant performance for the IG 1-B compressor with no heat integration is almost as good as the plant performance of the other compressor designs when thermal integration to the FWHs or to the stripper reboiler is considered.

The Ramgen compressor has only one stage of compression which results in the largest power requirement and worst plant performance of the compressors, but it has the advantage of having the smallest footprint of the compressors. This is important if available space is limited in a plant considering adding a capture system. As more stages and more inter-cooling are added, the power requirements decrease, but the footprint of the compressor greatly increases. The IG 1-B compressor would require the largest amount of space within a plant, but it also requires the least amount of power to operate.

It can be seen in these figures that using the available heat to dry the coal results in significantly greater improvement to the plant's performance than the other two thermal integration options.

Using the heat for coal drying has almost twice the improvement in heat rate and efficiency as integrating the heat to the stripper reboiler or FWH for the Ramgen and Inline 4 compressors. For the IG 1-A and IG 1-B compressors, coal drying improves the heat rate and efficiency about three times as much as the other thermal integration options. These improvements are in addition to the fact that less coal is being burned when the coal is dried and less CO_2 is formed in the flue gas.

When considering thermal integration to the stripper reboiler and to the FWHs, it can be seen that there is not a significant difference in the plant performance for these two thermal integration options. For the compressors where more heat is available, like the Ramgen and Inline 4, integrating the available heat to the stripper reboiler provides a slightly greater improvement to plant performance. For the compressors where less heat is available, like the integrally-geared compressors, integrating the available heat to the FWHs provides a slightly greater improvement to plant performance. Choosing between these two thermal integration options would likely be plant specific. One factor in choosing which thermal integration is best would be how close the compressor is to either the stripper reboiler or the FWHs and how feasible it is to pipe the hot cooling water from compression to these locations for thermal integration. Another factor would be how much space is available within the steam turbine cycle to install the additional heat exchangers that would be required.

It is important to note again that these results do not include the refrigeration loads for chilling the flue gas and lean ammonia streams to 32°F prior to entering the absorber. Adding these refrigeration loads to this analysis would increase the station service power equally for each compressor design which would decrease the net power equally for each compressor design. The comparison between the different compressor designs would be the same in that the Ramgen would have the most impact on plant performance while the IG 1-B compressor would have the smallest impact. The comparison of the thermal integrations would also be the same when refrigeration loads are considered in that coal drying would still have a larger improvement to the plant performance than either FWH or stripper reboiler thermal integration. The absolute values for net power, heat rate, and efficiency would change with the addition of refrigeration loads, but the changes in net power, heat rate, and efficiency as a result of thermal integration would be similar to the values found in this thesis. Without the refrigeration loads included in the analyses, these results cannot be accurately compared to other amine-based capture systems.

The choice of compressor design and method of thermal integration would most likely be plant specific and depend on how much space is available and how feasible thermal integration is within the plant. The thermal integrations require piping for the cooling water, pumps, and additional heat exchangers, and if a plant is not capable or willing to add this additional equipment, then the IG 1-B compressor may be the best compressor option. If space is limited for the compressor, but it is feasible to add the additional equipment to integrate the compression heat, then the Ramgen compressor may be a better choice.

A cost analysis would also need to be completed for the addition of a capture system in order to determine which compressor and which thermal integration option would be the best value to the plant.

If a plant already owns a compressor and is deciding which thermal integration method to use from this compressor, then coal drying would be the best option. Otherwise, the choice between integrating to the FWHs and the stripper reboiler would depend on two main factors. The first factor is if the hot cooling water from the compressor is at a high enough temperature to integrate the heat to the stripper reboiler. The second factor would be the plant layout and how feasible it is to pipe the hot cooling water to either the stripper reboiler or the FWHs within the plant.

Overall, the plant performance is significantly affected by the addition of a capture system, no matter what type of compressor is utilized. Coal drying would result in the largest improvement to the plant performance of the thermal integration options analyzed. Which compressor and what type of thermal integration to utilize would need to be chosen based on what suits the plant's specific needs and provides the greatest improvement to that plant's performance.

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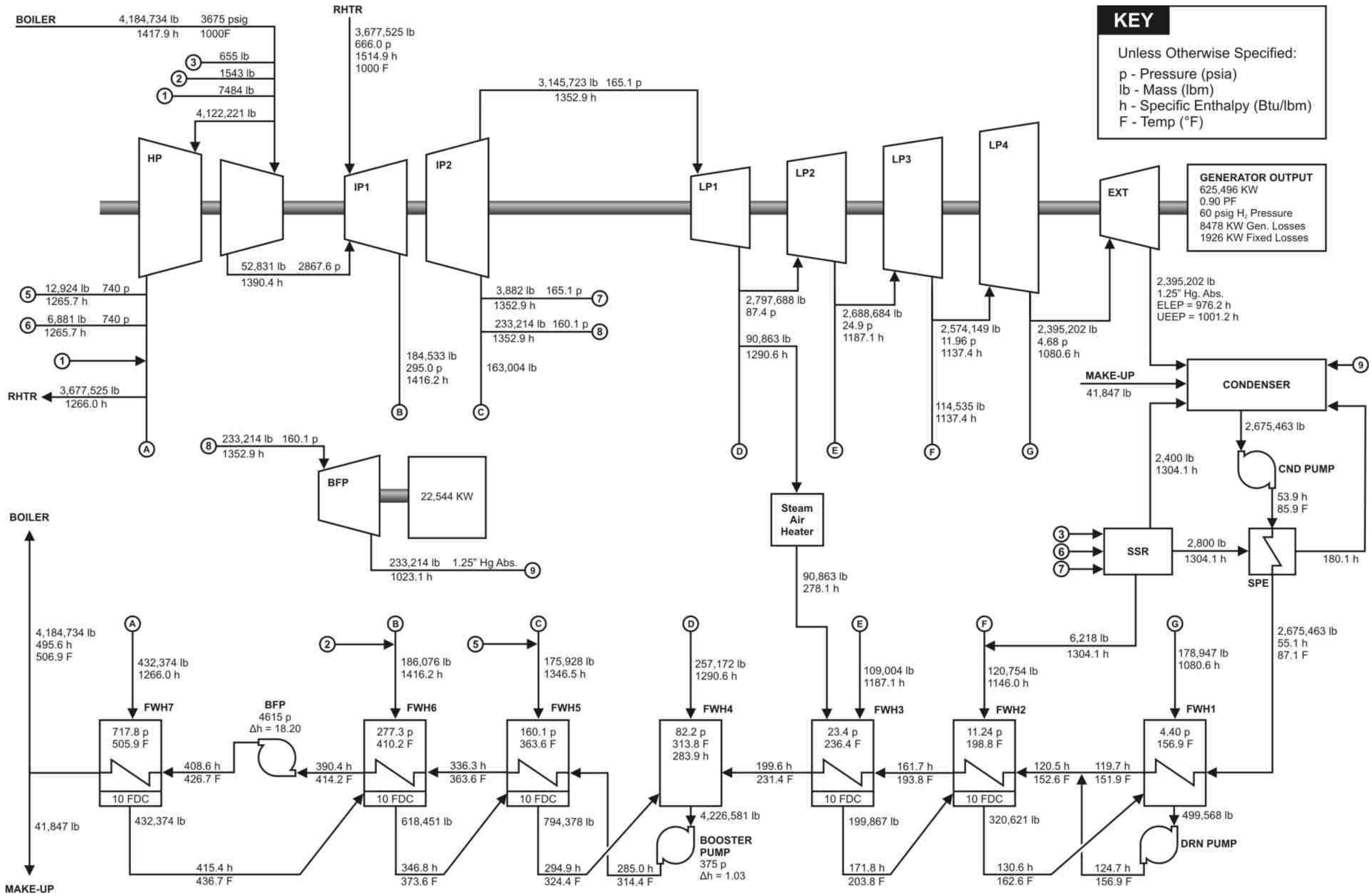
Appendix A – Steam Turbine Cycle



Appendix B – Compressor Data

B.1 Manufacturer's Specifications

Manufacturer's Specifications for Ramgen Compressor

Stage	1	2
Temperature$_{in}$ [°F]	69.0	69.0
Temperature$_{exit}$ [°F]	442.5	462.2
Pressurein [psia]	23.5	226.2
Pressure$_{exit}$ [psia]	230.7	2,220
Pressure Ratio	9.810	9.814
Mass Flow$_{in}$ [lbm/hr]	629,820	629,820
Volume Flow$_{in}$ [ft^3/hr]	-	-
Compressibility	-	-
Composition [mol-frac]		
CO_2	1.0000	1.0000
H_2O	0.0000	0.0000
Isentropic Eff.	0.850	0.850
Polytropic Eff.	-	-
Mechanical Eff.	0.9704	0.9701
Mechanical Power [kW]	15,664	14,300
Gas Power [kW]	15,200	13,872

Manufacturer's Specifications for Inline 4 Compressor

Stage	1	2	3
$Temperature_{in}$ [°F]	70.0	90.0	90.0
$Temperature_{exit}$ [°F]	381.6	397.4	321.7
Pressurein [psia]	16	100	600
$Pressure_{exit}$ [psia]	105	605	2,220
Pressure Ratio	6.563	6.05	3.7
Mass $Flow_{in}$ [lbm/hr]	1,300,020	1,300,020	1,300,020
Volume $Flow_{in}$ [ft^3/hr]	173,869	28,140	3,763
Compressibility	1.0000	0.9702	0.7957
Composition [mol-frac]			
CO_2	1.0000	1.0000	1.0000
H_2O	0.0000	0.0000	0.0000
Isentropic Eff.	0.8125	0.8188	0.8114
Polytropic Eff.	0.8445	0.8488	0.8364
Mechanical Eff.	0.9930	0.9920	0.9980
Mechanical Power [kW]	25,951	24,674	14,196
Gas Power [kW]	25,771	24,477	14,168

Manufacturer's Specifications for Integrally-Geared 1 Compressor

Stage	1	2	3	4	5	6	7
$Temperature_{in}$ [°F]	149.0	149.0	149.0	149.0	149.0	149.0	226.8
$Temperature_{exit}$ [°F]	294.1	278.8	278.2	275.4	233.1	226.9	301.3
Pressurein [psia]	27.56	59.03	124.30	271.01	565.15	928.24	1,504.81
$Pressure_{exit}$ [psia]	66.3	131.6	278.3	572.4	950.0	1,505.3	2,320.6
Pressure Ratio	2.405	2.228	2.239	2.112	1.681	1.622	1.542
Mass $Flow_{in}$ [lbm/hr]	670,018	647,026	637,815	633,523	631,675	630,037	630,037
Volume $Flow_{in}$ [ft^3/hr]	3,917,949	1,676,771	758,079	329,817	144,514	76,490	53,513
Compressibility	0.9920	0.9850	0.9700	0.9350	0.8600	0.7520	0.7560
Composition* [mol-frac]							
CO_2	0.8539	0.9246	0.9566	0.9719	0.9789	0.9842	0.9842
H_2O	0.1461	0.0754	0.0435	0.0281	0.0211	0.0158	0.0158
Isentropic Eff.	0.8542	0.8615	0.8757	0.8316	0.8898	0.9071	0.9175
Polytropic Eff.	0.8597	0.8649	0.8754	0.8343	0.8846	0.8781	0.8585
Mechanical Eff.†	0.9700	0.9700	0.9700	0.9700	0.9700	0.9700	0.9700
Mechanical Power [kW]	6,875	5,655	5,342	4,920	2,855	2,326	2,425
Gas Power [kW]	6,669	5,486	5,182	4,773	2,769	2,256	2,353

* Taken from values shown in "Thermal Integration of CO2 Compression Processes with Coal-Fired Power Plants..." - Edward Levy
† Losses including gear losses between LP & HP casings.

B.2 ASPEN Plus Results vs. REFPROP Data for Compressors, Inter-Coolers and Post-Coolers

Ramgen (1 stage)

Stage 1	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**471.4**	**464.6**	**1.45%**
CO_2 P_{in} (psia)	300.0		-
CO_2 P_{out} (psia)	2,200.0		-
P ratio	7.33		-
Isentropic Eff.	0.8500		-
Mechanical Eff.	0.9702		-
Power (kW)	**23,894**	**22,788.0**	**4.86%**

PC1	Aspen	REFPROP	% Diff
H_2O Flow (lb/hr)	459,225		-
H_2O P_{in} (psia)	550.0		-
H_2O P_{out} (psia)	535.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**466.4**	**462.3**	**0.89%**
CO_2 T_{in} (F)	471.4		-
CO_2 T_{out} (F)	120.0		-
Heat Duty (kBtu/hr)	**174,144**	**172,167**	**1.15%**

Inline 4 (2 stages)

Stage 1	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**438.2**	**431.6**	**1.54%**
CO_2 P_{in} (psia)	300.0		-
CO_2 P_{out} (psia)	1,815.0		-
P ratio	6.05		-
Isentropic Eff.	0.8188		-
Mechanical Eff.	0.9920		-
Power (kW)	**21,225.8**	**20,308.7**	**4.52%**

IC1	Aspen	REFPROP	% Diff
H_2O Flow (lb/hr)	492,072		-
H_2O P_{in} (psia)	400.0		-
H_2O P_{out} (psia)	385.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**433.2**	**426.3**	**1.61%**
CO_2 T_{in} (F)	438.2		-
CO_2 T_{out} (F)	110.0		-
Heat Duty (kBtu/hr)	**168,446**	**164,834**	**2.19%**

Stage 2	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**121.3**	**120.2**	**0.94%**
CO_2 P_{in} (psia)	1,810.0		-
CO_2 P_{out} (psia)	2,215.0		-
P ratio	1.224		-
Polytropic Eff.	0.8114		-
Mechanical Eff.	0.9980		-
Power (kW)	**785.4**	**674.4**	**16.45%**
Total Power (kW)	**22,011.2**	**20,983.1**	**4.90%**

IG 1-A (4 stages)

Stage 1	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**217.6**	**213.5**	**1.90%**
CO_2 P_{in} (psia)	300.0		-
CO_2 P_{out} (psia)	572.4		-
P ratio	1.908		-
Polytropic Eff.	0.8343		-
Mechanical Eff.	0.9700		-
Power (kW)	**6,737.1**	**6,438.5**	**4.64%**

IC1	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	175,417		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**212.6**	**212.3**	**0.15%**
CO_2 T_{in} (F)	217.6		-
CO_2 T_{out} (F)	149.0		-
Heat Duty (kBtu/hr)	**19,750**	**19,708**	**0.21%**

Stage 2	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	149.0		-
CO_2 T_{out} (F)	**236.3**	**233.3**	**1.29%**
CO_2 P_{in} (psia)	565.1		-
CO_2 P_{out} (psia)	950.0		-
P ratio	1.681		-
Polytropic Eff.	0.8846		-
Mechanical Eff.	0.9700		-
Power (kW)	**5,088.5**	**4,896.9**	**3.91%**

IC2	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	222,056		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**231.3**	**229.4**	**0.82%**
CO_2 T_{in} (F)	236.3		-
CO_2 T_{out} (F)	149.0		-
Heat Duty (kBtu/hr)	**29,189**	**28,784**	**1.41%**

Continued on next page

Stage 3	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	149.0		-
CO_2 T_{out} (F)	**230.9**	**228.1**	**1.23%**
CO_2 P_{in} (psia)	928.2		-
CO_2 P_{out} (psia)	1,505.3		-
P ratio	1.622		-
Polytropic Eff.	0.8781		-
Mechanical Eff.	0.9700		-
Power (kW)	**4,258.2**	**4,069.5**	**4.64%**

Stage 4	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,096,118		-
CO_2 T_{in} (F)	**230.9**	**228.1**	**1.23%**
CO_2 T_{out} (F)	**301.0**	**297.1**	**1.32%**
CO_2 P_{in} (psia)	1,505.3		-
CO_2 P_{out} (psia)	2,220.0		-
P ratio	1.475		-
Polytropic Eff.	0.8585		-
Mechanical Eff.	0.9700		-
Power (kW)	**4,084.6**	**3,807.5**	**7.28%**

Total Power (kW)	**20,168.3**	**19,212.3**	**4.98%**

PC1	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	572,086		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**296.0**	**290.7**	**1.83%**
CO_2 T_{in} (F)	301.0		-
CO_2 T_{out} (F)	120.0		-
Heat Duty (kBtu/hr)	**112,852**	**109,765**	**2.81%**

Total Heat Duty (kBtu/hr)	**161,791**	**158,258**	**2.23%**

IG 1-B (4 stages)

Stage 1	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,047,487		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**221.8**	**213.5**	**3.87%**
CO_2 P_{in} (psia)	300.0		-
CO_2 P_{out} (psia)	572.4		-
P ratio	1.908		-
Polytropic Eff.	0.8343		-
Mechanical Eff.	0.9700		-
Power (kW)	**6,371.7**	**5,968.3**	**6.76%**

IC1	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	262,657		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**216.8**	**219.4**	**1.19%**
CO_2 T_{in} (F)	211.0		-
CO_2 T_{out} (F)	110.0		-
Heat Duty (kBtu/hr)	**30,676**	**31,393**	**2.28%**

Stage 2	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,047,487		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**193.7**	**190.2**	**1.84%**
CO_2 P_{in} (psia)	567.4		-
CO_2 P_{out} (psia)	950.0		-
P ratio	1.674		-
Polytropic Eff.	0.8846		-
Mechanical Eff.	0.9700		-
Power (kW)	**4,225.6**	**4,011.7**	**5.33%**

Continued on next page

IC2	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	338,243		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**188.7**	**191.6**	**1.49%**
CO_2 T_{in} (F)	193.7		-
CO_2 T_{out} (F)	110.0		-
Heat Duty (kBtu/hr)	**29,955**	**30,953**	**3.22%**

Stage 3	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,047,487		-
CO_2 T_{in} (F)	110.0		-
CO_2 T_{out} (F)	**183.8**	**180.7**	**1.74%**
CO_2 P_{in} (psia)	945.0		-
CO_2 P_{out} (psia)	1,505.3		-
P ratio	1.593		-
Polytropic Eff.	0.8781		-
Mechanical Eff.	0.9700		-
Power (kW)	**3,184.0**	**2,964.7**	**7.40%**

IC3	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	416,059		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**178.8**	**181.4**	**1.44%**
CO_2 T_{in} (F)	183.8		-
CO_2 T_{out} (F)	130.0		-
Heat Duty (kBtu/hr)	**32,720**	**33,844**	**3.32%**

Stage 4	Aspen	REFPROP	% Diff
CO_2 Flow (lb/hr)	1,047,487		-
CO_2 T_{in} (F)	130.0		-
CO_2 T_{out} (F)	**178.7**	**178.0**	**0.38%**
CO_2 P_{in} (psia)	1,500.3		-
CO_2 P_{out} (psia)	2,220.0		-
P ratio	1.480		-
Polytropic Eff.	0.8585		-
Mechanical Eff.	0.9700		-
Power (kW)	**2,211.3**	**1,948.8**	**13.47%**

PC1	Aspen	REFPROP	% Diff
H_2O Mass Flow (lb/hr)	617,858		-
H_2O P_{in} (psia)	300.0		-
H_2O P_{out} (psia)	285.0		-
H_2O T_{in} (F)	100.0		-
H_2O T_{out} (F)	**173.7**	**174.7**	**0.56%**
CO_2 T_{in} (F)	178.7		-
CO_2 T_{out} (F)	120.0		-
Heat Duty (kBtu/hr)	**45,437**	**46,089**	**1.42%**

Total Power (kW)	**15,992.6**	**14,893.5**	**7.38%**

Total Heat Duty (kBtu/hr)	**138,788**	**142,280**	**2.45%**

Appendix C – Plant Integration

C.1 Plant Performance without Thermal Integration

	Ramgen	Inline 4	IG 1-A	IG 1-B	No Capture
Gross Power [kW]	519,266	519,266	519,266	519,266	635,768
Gen Power [kW]	511,477	511,477	511,477	511,477	626,232
STC HR [Btu/kWh]	9,338	9,338	9,338	9,338	7,627
Fan Power [kW]	18,340	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430	3,430
Pump Power [kW]	2,832	2,833	2,851	2,883	1,658
Aux. Power [kW]	15,000	15,000	15,000	15,000	15,000
Pss [kW]	39,602	39,603	39,621	39,653	38,428
Comp. Power [kW]	23,894	22,011	20,168	16,347	--
Net Power [kW]	447,980	449,862	451,687	455,477	587,804
Δ Net Power [kW]	-139,824	-137,942	-136,116	-132,327	--
Unit HR [Btu/kWh]	12,191	12,140	12,091	11,991	9,291
Δ HR [Btu/kWh]	2,900	2,849	2,800	2,699	--
% Unit HR Lost [%]	31.21%	30.66%	30.14%	29.05%	--
Unit Efficiency [%]	27.99%	28.10%	28.22%	28.46%	36.72%
Δ Unit Efficiency [%]	-8.74%	-8.62%	-8.50%	-8.27%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

C.2 Plant Performance with Thermal Integration

C.2.1 Ramgen

Ramgen Plant Performance with FWH Thermal Integration

	FWH-3	FWH-2	FWH-1	No Thermal Integration
Gross Power [kW]	526,792	525,574	523,215	519,266
Gen Power [kW]	518,890	517,690	515,367	511,477
TCHR [Btu/kWh]	9,205	9,226	9,268	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,849	2,824	2,833	2,832
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,619	39,594	39,603	39,603
Comp. Power [kW]	23,894	23,894	23,894	23,894
Net Power [kW]	455,377	454,202	451,870	447,980
Δ Net Power [kW]	7,397	6,222	3,890	--
Unit HR [Btu/kWh]	11,993	12,024	12,087	12,191
Δ Unit HR [Btu/kWh]	-198	-167	-105	--
% Unit HR Improvement	1.62%	1.37%	0.86%	--
Unit Efficiency [%]	28.45%	28.38%	28.23%	27.99%
Δ Unit Efficiency [%]	0.46%	0.39%	0.24%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

Ramgen Plant Performance with Reboiler Thermal Integration

	Reboiler	**No Thermal Integration**
Gross Power [kW]	527,983	519,266
Gen Power [kW]	520,063	511,477
TCHR [Btu/kWh]	9,184	9,338
Fan Power [kW]	18,340	18,340
Pulv. Power [kW]	3,430	3,430
Pump Power [kW]	2,841	2,832
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,611	39,603
Comp. Power [kW]	23,894	23,894
Net Power [kW]	456,558	447,980
Δ Net Power [kW]	8,578	--
Unit HR [Btu/kWh]	11,962	12,191
Δ Unit HR [Btu/kWh]	-229	--
% Unit HR Improvement	1.88%	--
Unit Efficiency [%]	28.52%	27.99%
Δ Unit Efficiency [%]	0.54%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

Ramgen Plant Performance with Coal Drying Thermal Integration

	Coal Drying	**No Thermal Integration**
Coal Moisture Entering Mill [%]	15.03%	28.09%
Wet Coal Flow [lb/hr]	630,216	648,177
% Less Coal Burned	-2.77%	0.00%
CO_2 Flow- Comp [lb/hr]	1,064,769	1,096,118
% Reboiler Extraction Reduction	2.86%	0.00%
Gross Power [kW]	522,598	519,266
Gen Power [kW]	514,759	511,477
TCHR [Btu/kWh]	9,279	9,338
Fan Power [kW]	19,089	18,340
Pulv. Power [kW]	2,823	3,430
Pump Power [kW]	2,848	2,832
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,759	39,603
Comp. Power [kW]	22,519	23,894
Net Power [kW]	452,481	447,980
Δ Net Power [kW]	4,501	--
Unit HR [Btu/kWh]	11,736	12,191
Δ Unit HR [Btu/kWh]	-456	--
% Unit HR Improvement	-3.74%	--
Unit Efficiency [%]	29.07%	27.99%
Δ Unit Efficiency [%]	1.09%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

C.2.2 Inline 4

Inline 4 Plant Performance with FWH Thermal Integration

	FWH-3	FWH-2	FWH-1	No Thermal Integration
Gross Power [kW]	526,617	524,643	523,086	519,266
Gen Power [kW]	518,718	516,774	515,240	511,477
TCHR [Btu/kWh]	9,208	9,242	9,270	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,851	2,824	2,834	2,833
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,621	39,594	39,604	39,604
Comp. Power [kW]	22,011	22,011	22,011	22,011
Net Power [kW]	457,086	455,168	453,625	449,862
Δ Net Power [kW]	7,224	5,307	3,763	--
Unit HR [Btu/kWh]	11,949	11,999	12,040	12,140
Δ Unit HR [Btu/kWh]	-192	-142	-101	--
% Unit HR Improvement	1.58%	1.17%	0.83%	--
Unit Efficiency [%]	28.56%	28.44%	28.34%	28.10%
Δ Unit Efficiency [%]	0.45%	0.33%	0.24%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

Inline 4 Plant Performance with Reboiler Thermal Integration

	Reboiler	**No Thermal Integration**
Gross Power [kW]	527,094	519,266
Gen Power [kW]	519,187	511,477
TCHR [Btu/kWh]	9,199	9,338
Fan Power [kW]	18,340	18,340
Pulv. Power [kW]	3,430	3,430
Pump Power [kW]	2,842	2,833
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,612	39,604
Comp. Power [kW]	22,011	22,011
Net Power [kW]	457,564	449,862
Δ Net Power [kW]	7,702	--
Unit HR [Btu/kWh]	11,936	12,140
Δ Unit HR [Btu/kWh]	-204	--
% Unit HR Improvement	1.68%	--
Unit Efficiency [%]	28.59%	28.10%
Δ Unit Efficiency [%]	0.48%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

Inline 4 Plant Performance with Coal Drying Thermal Integration

	Coal Drying	No Thermal Integration
Coal Moisture Entering Mill [%]	15.03%	28.09%
Wet Coal Flow [lb/hr]	630,216	648,177
% Less Coal Burned	-2.77%	0.00%
CO_2 Flow- Comp [lb/hr]	1,064,769	1,096,118
% Reboiler Extraction Reduction	2.86%	0.00%
Gross Power [kW]	522,598	519,266
Gen Power [kW]	514,759	511,477
TCHR [Btu/kWh]	9,279	9,338
Fan Power [kW]	19,089	18,340
Pulv. Power [kW]	2,823	3,430
Pump Power [kW]	2,848	2,833
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,759	39,604
Comp. Power [kW]	21,382	22,011
Net Power [kW]	453,618	449,862
Δ Net Power [kW]	3,756	--
Unit HR [Btu/kWh]	11,706	12,140
Δ Unit HR [Btu/kWh]	-434	--
% Unit HR Improvement	-3.58%	--
Unit Efficiency [%]	29.15%	28.10%
Δ Unit Efficiency [%]	1.04%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

C.2.3 IG 1-A

IG 1-A Plant Performance with FWH Thermal Integration

	FWH-3	FWH-2	FWH-1	No Thermal Integration
Gross Power [kW]	524,454	522,986	521,825	519,266
Gen Power [kW]	516,588	515,141	513,998	511,477
TCHR [Btu/kWh]	9,246	9,272	9,292	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,868	2,860	2,851	2,851
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,638	39,630	39,621	39,621
Comp. Power [kW]	20,168	20,168	20,168	20,168
Net Power [kW]	456,781	455,342	454,209	451,687
∆ Net Power [kW]	5,094	3,655	2,522	--
Unit HR [Btu/kWh]	11,957	11,994	12,024	12,091
∆ Unit HR [Btu/kWh]	-135	-97	-67	--
% Unit HR Improvement	-1.12%	-0.80%	-0.56%	--
Unit Efficiency [%]	28.54%	28.45%	28.38%	28.22%
∆ Unit Efficiency [%]	0.32%	0.23%	0.16%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

IG 1-A Plant Performance with Reboiler Thermal Integration

	Reboiler	No Thermal Integration
Gross Power [kW]	521,465	519,266
Gen Power [kW]	513,643	511,477
TCHR [Btu/kWh]	9,299	9,338
Fan Power [kW]	18,340	18,340
Pulv. Power [kW]	3,430	3,430
Pump Power [kW]	2,861	2,851
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,631	39,621
Comp. Power [kW]	20,168	20,168
Net Power [kW]	453,844	451,687
Δ Net Power [kW]	2,157	--
Unit HR [Btu/kWh]	12,034	12,091
Δ Unit HR [Btu/kWh]	-57	--
% Unit HR Improvement	0.48%	--
Unit Efficiency [%]	28.35%	28.22%
Δ Unit Efficiency [%]	0.13%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

IG 1-A Plant Performance with Coal Drying Thermal Integration

	Coal Drying Option 1	Coal Drying Option 2	No Thermal Integration
Coal Moisture Entering Mill [%]	17.38%	16.06%	28.09%
Wet Coal Flow [lb/hr]	632,268	631,099	648,177
% Less Coal Burned	-2.45%	-2.63%	0.00%
CO_2 Flow- Comp [lb/hr]	1,069,044	1,066,632	1,096,118
% Reboiler Extraction Reduction	2.47%	2.69%	0.00%
Gross Power [kW]	522,148	522,401	519,266
Gen Power [kW]	514,316	514,565	511,477
TCHR [Btu/kWh]	9,287	9,282	9,338
Fan Power [kW]	19,249	19,664	18,340
Pulv. Power [kW]	2,912	2,861	3,430
Pump Power [kW]	2,858	2,850	2,851
Aux. Power [kW]	15,000	15,000	15,000
Pss [kW]	40,019	40,375	39,621
Comp. Power [kW]	19,671	19,626	20,168
Net Power [kW]	454,626	454,564	451,687
∆ Net Power [kW]	2,939	2,877	--
Unit HR [Btu/kWh]	11,718	11,698	12,091
∆ Unit HR [Btu/kWh]	-373	-393	--
% Unit HR Improvement	-3.09%	-3.25%	--
Unit Efficiency [%]	29.12%	29.17%	28.22%
∆ Unit Efficiency [%]	0.90%	0.95%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

C.2.4 IG 1-B

IG 1-B Plant Performance with FWH Thermal Integration

	FWH-1	No Thermal Integration
Gross Power [kW]	520,887	519,266
Gen Power [kW]	513,074	511,477
TCHR [Btu/kWh]	9,309	9,338
Fan Power [kW]	18,340	18,340
Pulv. Power [kW]	3,430	3,430
Pump Power [kW]	2,883	2,883
Aux. Power [kW]	15,000	15,000
Pss [kW]	39,654	39,653
Comp. Power [kW]	16,347	16,347
Net Power [kW]	457,073	455,477
Δ Net Power [kW]	1,597	--
Unit HR [Btu/kWh]	11,949	11,991
Δ Unit HR [Btu/kWh]	-42	--
% Unit HR Improvement	-0.35%	--
Unit Efficiency [%]	28.55%	28.46%
Δ Unit Efficiency [%]	0.10%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

IG 1-B Plant Performance with Coal Drying Thermal Integration

	Coal Drying	**No Thermal Integration**
Coal Moisture Entering Mill [%]	18.24%	28.09%
Wet Coal Flow [lb/hr]	633,053	648,177
% Less Coal Burned	-2.33%	0.00%
CO_2 Flow- Comp [lb/hr]	1,071,236	1,096,118
% Reboiler Extraction Reduction	2.27%	0.00%
Gross Power [kW]	521,911	519,266
Gen Power [kW]	514,083	511,477
TCHR [Btu/kWh]	9,291	9,338
Fan Power [kW]	19,322	18,340
Pulv. Power [kW]	2,947	3,430
Pump Power [kW]	2,899	2,883
Aux. Power [kW]	15,000	15,000
Pss [kW]	40,168	39,653
Comp. Power [kW]	15,769	16,347
Net Power [kW]	458,146	455,477
Δ Net Power [kW]	2,668	--
Unit HR [Btu/kWh]	11,643	11,991
Δ Unit HR [Btu/kWh]	-348	--
% Unit HR Improvement	-2.90%	--
Unit Efficiency [%]	29.31%	28.46%
Δ Unit Efficiency [%]	0.85%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

C.3 Plant Performance with Stripper Condenser Thermal Integrations

Ramgen	Condenser to FWH-2	Combination 1: Comp. to Reb. + Cond. to FWH-2	Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2	No Thermal Integration
Gross Power [kW]	521,711	530,428	532,455	519,266
Gen Power [kW]	513,885	522,472	524,469	511,477
TCHR [Btu/kWh]	9,294	9,142	9,107	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,863	2,871	2,886	2,832
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,633	39,641	39,656	39,603
Comp. Power [kW]	23,894	23,894	23,894	23,894
Net Power [kW]	450,358	458,936	460,918	447,980
Δ Net Power [kW]	2,378	10,957	12,938	--
Unit HR [Btu/kWh]	12,127	11,900	11,849	12,191
Δ Unit HR [Btu/kWh]	-64	-291	-342	--
% Unit HR Improvement	0.53%	2.39%	2.81%	--
Unit Efficiency [%]	28.14%	28.67%	28.79%	27.99%
Δ Unit Efficiency [%]	0.15%	0.68%	0.81%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

Inline 4	Condenser to FWH-2	Combination 1: Comp. to Reb. + Cond. to FWH-2	Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2	No Thermal Integration
Gross Power [kW]	521,711	529,535	531,667	519,266
Gen Power [kW]	513,885	521,592	523,692	511,477
TCHR [Btu/kWh]	9,294	9,157	9,120	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,864	2,872	2,889	2,833
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,634	39,642	39,659	39,604
Comp. Power [kW]	22,011	22,011	22,011	22,011
Net Power [kW]	452,240	459,939	462,022	449,862
∆ Net Power [kW]	2,379	10,077	12,160	--
Unit HR [Btu/kWh]	12,077	11,874	11,821	12,140
∆ Unit HR [Btu/kWh]	-64	-266	-320	--
% Unit HR Improvement	0.53%	2.19%	2.63%	--
Unit Efficiency [%]	28.25%	28.73%	28.86%	28.10%
∆ Unit Efficiency [%]	0.15%	0.63%	0.76%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia

IG 1-A	Condenser to FWH-2	Combination 1: Comp. to Reb. + Cond. to FWH-2	Combination 2: Comp. to Reb. to FWH-3 + Cond. to FWH-2	No Thermal Integration
Gross Power [kW]	521,711	523,895	526,281	519,266
Gen Power [kW]	513,885	516,036	518,387	511,477
TCHR [Btu/kWh]	9,294	9,256	9,214	9,338
Fan Power [kW]	18,340	18,340	18,340	18,340
Pulv. Power [kW]	3,430	3,430	3,430	3,430
Pump Power [kW]	2,881	2,890	2,909	2,851
Aux. Power [kW]	15,000	15,000	15,000	15,000
Pss [kW]	39,651	39,660	39,679	39,621
Comp. Power [kW]	20,168	20,168	20,168	20,168
Net Power [kW]	454,066	456,208	458,539	451,687
∆ Net Power [kW]	2,379	4,521	6,852	--
Unit HR [Btu/kWh]	12,028	11,972	11,911	12,091
∆ Unit HR [Btu/kWh]	-63	-120	-181	--
% Unit HR Improvement	0.52%	0.99%	1.49%	--
Unit Efficiency [%]	28.37%	28.50%	28.65%	28.22%
∆ Unit Efficiency [%]	0.15%	0.28%	0.43%	--

Supercritical Plant

Coal: PRB

CO_2 Capture: Chilled Ammonia